THE TECHNOLOGY OF ARTIFICIAL LIFT METHODS

Volume 2a

Introduction of Artificial Lift Systems
Beam Pumping: Design and Analysis
Gas Lift

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I dedicate this book to my four children and three grandchildren. They are in the order in which God created them: Stephen Wesley, Sandra Kay, Robert Michael, David Earl, Stephen William, Bryan Scott and Robert Wade.

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Appendix

Preface

For the first time, all the artificial lift methods are presented in one volume. Volume 2 is published in two separate books, Vol. 2A and Vol. 2B, and is complete with sufficient charts, curves, etc. to plan, design, analyze, and compare all artificial lift methods. Volume 1, published in 1977, gives all the preliminary information needed to use Vol. 2. Volume 1 includes (1) the inflow ability of the well, (2) multiphase flow in pipes, and (3) the flowing well. Although not absolutely necessary, Volume 1 should be studied and used in conjunction with Volume 2. Volume 3, which is also available, includes over 2,200 flowing pressure traverse curves for multiphase vertical flow and horizontal flow, gas production, gas injection and water injection curves.

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Volume II offers sufficient flexibility to be used as a text or to be used by the engineer in industry in designing installations. Example problems are worked and numerous class problems are included. Eventually, an answer guide will be available.

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Chapter 1

Artificial lift systems

1.1 INTRODUCTION

This text will discuss the various types of artificial lift systems available today. More and more wells in the world are being placed on artificial lift, and the number will continue to increase. The selection of the most suitable type of artificial lift for a well or group of wells can be difficult or easy, depending upon the conditions.

Generally, more than one method of lift can be used. Each method of lift may be classified from excellent to poor in accomplishing the objective. Depending upon the economic considerations, two types of lift (one used later) may possibly be prescribed for a group of wells.

For example, in a "depletion" type reservoir, high initial production rates may be needed, but decreasing pressures and declining inflow capability may require a low rate in the future. In this case, an initial installation of continuous flow gas lift or electrical submersible pumping may be changed to intermittent gas lift, sucker rod pumping, or hydraulic pumping at a later date, or vice versa.

The following list probably represents the relative standing of lift systems based on the number of installations throughout the world. This differs from field to field, state to state, and country to country.

(1) Sucker rod pumping (beam pumping)
(2) Gas lift
(3) Electrical submersible pumping
(4) Hydraulic pumping
(5) Jet pumping
(6) Plunger (free piston) lift
(7) Other methods

As these methods are discussed, complete design procedures will be given along with numerous example problems.

In addition, other methods are continually being developed and tested. A short discussion on the ball pump and the gas-actuated pump are also given. The ball pump was tried many years ago, and new interest has recently sparked additional development of this lift method. The ball pump uses spherical flexible balls that pass down one tubing string and return through another in order to eliminate the slippage of gas past the liquids. Gas is used as the source of power.

The gas pump has been in the experimental stage for many years. Several field trials have been performed and are installed at the present time. The pump uses gas to actuate a downhole pump and can be used in conjunction with gas lift. In particular, gas lift unloading valves may be used to reach the pump.

Rothrock presented Table 1.1, showing the distribution of 518,867 oil wells based on a 7% sampling from 200 operators with information on 37,100 wells.[1]

TABLE 1.1
CRUDE PRODUCING WELLS (JAN., 1977)

Category	Number	Percent
Rod pumping	409,974	85.21
Gas lift	51,964	10.80
Electrical submersible	9,738	2.02
Hydraulic	9,470	1.97
Total U.S. artificial lift	481,146	100.00%
U.S. flowing oil wells	37,721	
Total U.S. producing oil wells	581,867	

Of the 518,867 wells, 481,146 or 92.7% are being lifted artificially. These are further broken down into 85.2% rod pump, 10.8% gas lift, 2% submersible, and 2% hydraulic. Approximately 383,000 of the artificial wells are strippers (producing less than 10 B/D). Assuming that 100% of the stripper wells are on rod pump, then Table 1.2 shows a re-distribution of the remaining wells.

TABLE 1.2
ARTIFICIAL LIFT WELLS (LESS STRIPPER WELLS)

Category	Number	Percent
Rod pumping	26,974	27.48
Gas lift	51,964	52.95
Electrical submersible	9,738	9.92
Hydraulic	9,470	9.65
Total wells	98,146	100.00%

Table 1.2 shows that the largest percentage of the wells are on gas lift after eliminating stripper wells. Gas lift predominates on offshore wells but, according to Rothrock, is not keeping pace with other lift methods in areas other than offshore.

Submersible pump use is increasing rapidly in West Texas and in some Rocky Mountain areas. Rothrock noted that hydraulic pumping is not growing at the rate of other methods. However, jet pumping is now taking 50% of the hydraulic market, and its popularity will probably continue to grow.

Table 1.3 breaks down total maintenance cost into failures, failure rates, cost per failure, total cost, and percent of each failure spent for well servicing. Also included are costs and rates for well workover.

Two main types of downhole equipment failures are pumps and rods. Each pumping well has a 57% chance of pump failure and a 44% chance of rod failure each year. These rates are additive so each well will average 1.01 failures/year.

Costs to maintain these wells total $346,000,000, including $230,000,000 for well servicing. Remaining expenditures translate to approximately 34,000,000 feet of rods and 31,000 bottomhole pumps needed for replacement of worn out equipment plus an additional repair of 180,000 pumps.

Submersible pump failure rates apparently are decreasing based on the 7% sample. Repairs to submersibles by manufacturers are running considerably less than in previous years.

The survey indicates that failure incidence in hydraulic pumps is increasing. Again, this may be due to the small number of hydraulic pumps contained in the survey. In any event, the growth of the hydraulic piston pump market appears to be declining and jet pumping is on the increase.

Gas lift failure rate is the lowest of any form of artificial lift (21%) but costs of individual failures are high compared to the average of all failures. A high portion of these costs, however, is attributable to high cost of service units, crews, and related equipment.

Various lift methods are compared in Chapter 9. Comparisons are quite difficult, but some choices of lift methods are obvious. One example is high volume wells where either electrical submersible pumping or continuous flow gas lift should be considered. Very deep lift must look at hydraulic pumping with installations presently pumping from 15,000 to 18,000 feet, with rates of 300 to 500 B/D. Although these pumps are set at these depths, the effective lift depth may be less. Effective lift refers to that depth to which the flowing bottom hole pressure will support the producing fluids. For example, if the flowing bottom hole pressure is 700 psi and the average fluid gradient is 0.30 psi/ft, the 700 psi will support the fluid 700/.30 = 2333 ft. Therefore, if the pump is set at a total depth of 15,000 ft, it is really only lifting from 15,000-2,333 = 12,667 ft and theoretically could be set at that depth and produce the same amount of fluids (neglects tubing well-head back pressure).

Availability of certain power sources will influence the decision on type of lift. All these factors are discussed in detail in Chapter 9.

1.11 Purpose of artificial lift

The purpose of artificial lift is to maintain a reduced producing bottom hole pressure so the formation can give up the desired reservoir fluids. A well may be capable of performing this task under its own power. In its latter stages of flowing life, a well is capable of producing only a portion of the desired fluids. During this stage of a well's flowing life and particularly after the well dies, a suitable means of artificial lift must be installed so the required flowing bottom hole pressure can be maintained.

Maintaining the required flowing bottom hole pressure is the basis for the design of any artificial lift installation; if a predetermined drawdown in pressure can be maintained, the well will produce the desired fluids. This is true regardless of the type of lift installed.

In gas lift operations, a well may be placed on continuous or intermittent lift. In continuous flow, the flowing bottom hole pressure will remain constant for a particular set of conditions, while in intermittent flow the flowing bottom hole pressure will vary with the particular operation time of one cycle of production. In this latter case, a weighted average flowing bottom hole pressure must be determined for one cycle and, hence, for a day's production. Economics enters into the design of any lift installation.

Many types of artificial lift methods are available: beam-type sucker rod pumps, piston-type sucker rod pumps, hydraulic oil well pumps, electrical submergible centrifugal pumps, rotating rod pumps, plunger lift, gas lift, and others. The advantages and disadvan-

TABLE 1.3
DOWNHOLE COSTS TO MAINTAIN U.S. PRODUCING OIL WELLS (YEAR 1977)

	Failure rate	Number of failures	Average cost, $	Total cost, $	Percent well servicing
Subsurface rod pumps	.57	210,277	1,078	226,657,000	60
Sucker rods	.44	164,118	729	119,665,000	79
Submersible pumps	.35	3,390	7,679	26,030,000	15
Hydraulic pumps	1.86	16,397	2,445	41,411,000	40
Gas lift	.21	11,490	4,153	47,713,000	78
Tubing	.12	62,623	1,837	115,027,000	73
Casing	.021	11,043	16,005	176,742,000	51
Total failures	.92	479,878	1,570	753,245,000	61
Workovers	.20	105,145	10,686	1,123,532,000	58
Total maintenance cost				1,876,777,000	59

tages of all types of artificial lift are discussed in Chapter 9.

1.12 Utilization of multiphase flow correlations for artificial lift systems

1.121 Introduction

All lift systems can be designed better if the proper use of multiphase flow correlations is utilized. Numerous references, including Volume 1 of this series "Inflow Performance, Multiphase Flow in Pipes, and the Flowing Well," give excellent discussions on vertical, horizontal, and inclined multiphase flow correlations. The best correlations for vertical multiphase flow are those of Ros,[2] Hagedorn,[3] Orkiszewski,[4] and Beggs and Brill.[5] Some of these correlations are better in some ranges of flow conditions than others, as was discussed in a paper by Lawson and Brill.[6] The best horizontal flow correlations are those of Dukler,[7] Eaton,[8] Lockhart and Martinelli,[9] Baker,[10] and Beggs and Brill.[5] The Beggs and Brill correlation has proven to be excellent for horizontal flow, and some designers are using the Dukler procedure with the Eaton hold-up correlation. For inclined flow applicable to the directional well problem, the best solution appears to be the use of a vertical multiphase flow correlation by calculating frictional loss over the measured tubing length and calculating the static density loss over the vertical distance only. The kinetic energy term is calculated over the total tubing length, also.

For the surface hilly terrain problem the Beggs and Brill correlation is widely used. The Flanigan solution as well as modifications to the Flanigan solution are also used.[11] For example, use of the Flanigan procedure to determine pressure loss over the hills in conjunction with either the Dukler or Eaton procedure for frictional loss is commonly used.

For a complete discussion on multiphase flow, refer to Volume 1 of this series.[12] Also, the productive ability of wells is discussed in Volume 1, along with a complete analysis of the flowing well.

1.122 Use of multiphase flow correlations

The use of multiphase flow correlations is very important in the design of artificial lift installations. This subject is discussed in detail in Volume 1. Refer to it for discussions on the various correlations, their range of applicability, and those that are best for vertical, horizontal, inclined, and directional flow.

The publication of gradient curves (working curves) has made this technology readily available for the field man who does not always have a computer available for immediate calculations. Most energy companies and artificial lift companies have a computer program available for the calculation of pressure losses for multiphase flow in pipes. Refer to Volume 3 of this series for a complete set of curves that have been prepared for both the Hagedorn and Beggs correlations. Others are available, such as the curves used by the Shell companies, that were prepared from the Duns and Ros correlation.

In the future it will be necessary to lift more heavy and viscous crude oils. Present-day correlations show a great deal of discrepancy in handling viscous effects. Hagedorn and Ros were two investigators who varied the viscosities in their experimental work, yet their predictions of pressure loss for viscous crudes do not coincide.

1.1221 Viscosity

Increases in viscosity are expected to cause increases in pressure drop since frictional pressure loss is directly proportional to viscosity. The pressure traverse is expected to move to the right for increasing values of oil viscosity. A plot of well bottom hole pressure vs. viscosity will show increasing bottom hole pressure with increasing viscosity.

During two-phase flow in a well where both gas and liquid are present, the resultant frictional pressure drop depends on both the gas and liquid viscosities. Different multiphase flow correlations handle this combination of gas and liquid viscosities differently, e.g. Beggs and Brill and Hagedorn and Brown define a mixture viscosity as follows:

$$\mu_m = \mu_L \lambda_L + \mu_g (1 - \lambda_L) \quad \text{Beggs and Brill}$$

$$\mu_m = \mu_L^{H_L} \times \mu_g^{(1-H_L)} \quad \text{Hagedorn and Brown}$$

where:

μ_m = mixture viscosity
μ_L = liquid viscosity
H_L = liquid holdup
λ_L = no slip holdup or input liquid content

Duns and Ros use only the liquid viscosity in the slug and bubble flow regimes and correct the Moody friction factor thus obtained by means of dimensionless factors. These factors are functions of the Moody friction factor, in situ gas and liquid velocity, and pipe diameter number. Gas viscosity only is used in the mist flow regime. Orkiszewski also uses liquid viscosity in bubble and slug flow and the gas viscosity in mist flow.

To study how each correlation handles increasing viscosity, a typical well was chosen and, with all other data held constant, the pressure traverses were calculated for increasing values of viscosity. Since it was necessary to hold the oil API constant, the Chew and Connally correlation was used to predict oil viscosity.[13] Dead oil viscosities (μ_{oD}) were chosen and the actual live oil viscosities (μ_o) were calculated as follows:

$$\mu_o = A[\mu_{oD}(T)]^B$$

where:

μ_{oD} = deal oil viscosity (stock tank oil)
μ_o = live oil viscosity (oil containing dissolved gas)
A and B are factors given by Chew and Connally.[13]

$\mu_{oD}(T)$ is the dead oil viscosity at the required temperature and is found using an interpolation scheme given by Brill and Beggs as follows:[5]

$$\mu_{oD}(T) = \mu_{oD}(100°F.)\left[\frac{T(°F.)}{100}\right]^X$$

$$\text{where } X = 1.345 \ln\left[\frac{\mu_{oD}(210°F.)}{\mu_{oD}(100°F.)}\right]$$

For calculating the viscosity above the bubble point pressure, it was necessary to interpolate the plot of rate

of increase of viscosity above the bubble point pressure given by Beal.[14] Gas viscosity was calculated using the Lee correlation. [15]

The following well data were used in calculating the pressure traverses:

Well depth	8000 ft
Tubing diameter	1.995 in. (I.D.)
Gas-liquid ratio	500 scf/bbl
Water-cut	0
Wellhead pressure	100 psia
Bottom hole temperature	200°F.
Wellhead temperature	150°F.

Refer to Figures 1.1, 1.2, and 1.3 for comparisons of the Hagedorn, Ros, Orkizewski, and Beggs and Brill correlation for oil viscosities of 10.0, 100.0, and 1000.0 cp. at 100°F.

Figure 1.4 shows plots of bottom hole pressure vs. viscosity for each correlation tested. The results show the following:

(1) For all correlations tested, pressure drop increases with increasing viscosity.

(2) Increases in viscosity within the range of 5 cp to 100 cp (at 100°F.) cause rapid changes in pressure drop for the Hagedorn, Beggs, and Ros correlations. Beyond 100 cp (at 100°F.) the rate of increase in pressure drop decreases until approximately 500 cp, after which increases in viscosity have negligible effects on pressure drop.

The Orkiszewski correlation shows an initial decrease in pressure drop as viscosity increases from 5 cp to 10 cp (at 100°F.). Beyond 10 cp, the pressure drop increases rapidly until 100 cp as with the other correlations. Beyond 100 cp, pressure drop increases with increased viscosity. However, the rate of increase decreases with increasing viscosity.

The differences in bottom hole pressures predicted

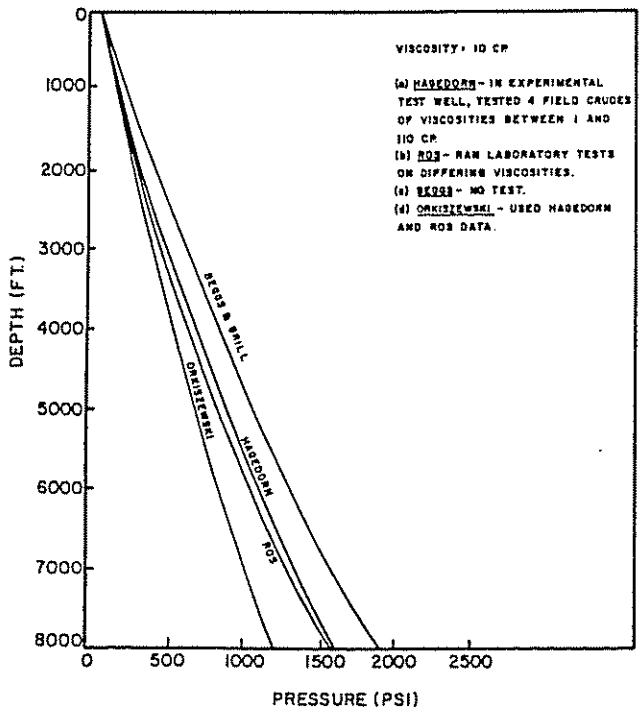


Fig. 1.1 Comparison of Correlations for Viscosity Effects

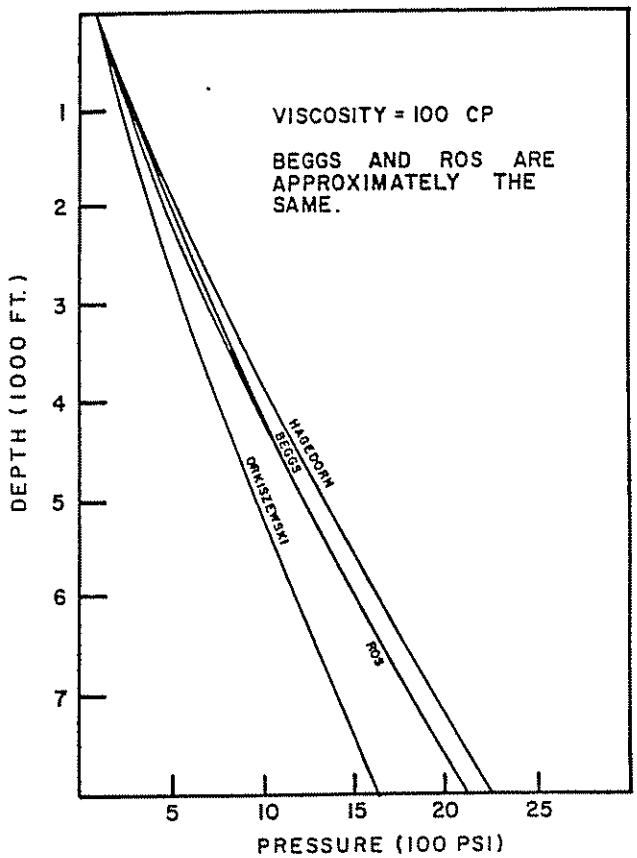


Fig. 1.2 Comparison of Correlations for Viscosity Effects

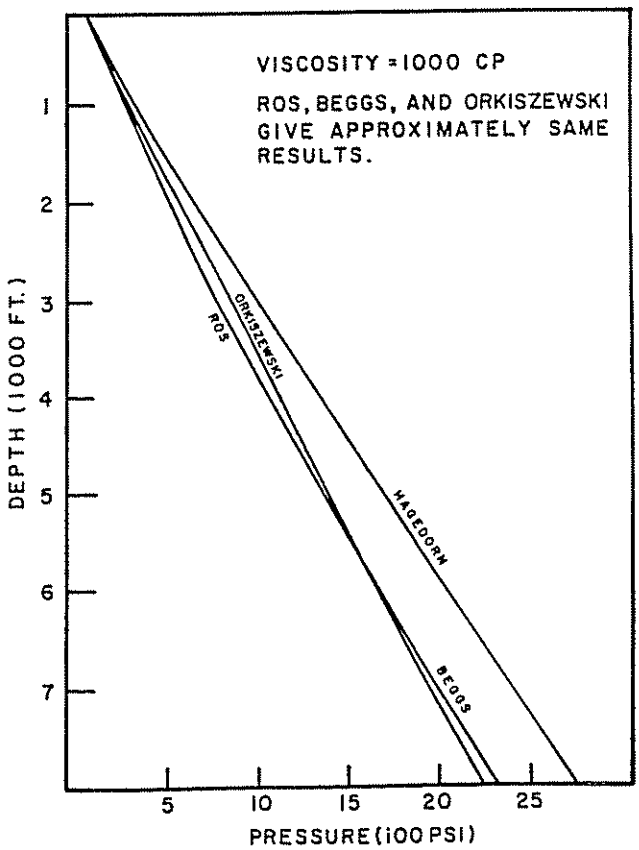


Fig. 1.3 Comparison of Correlations for Viscosity Effects

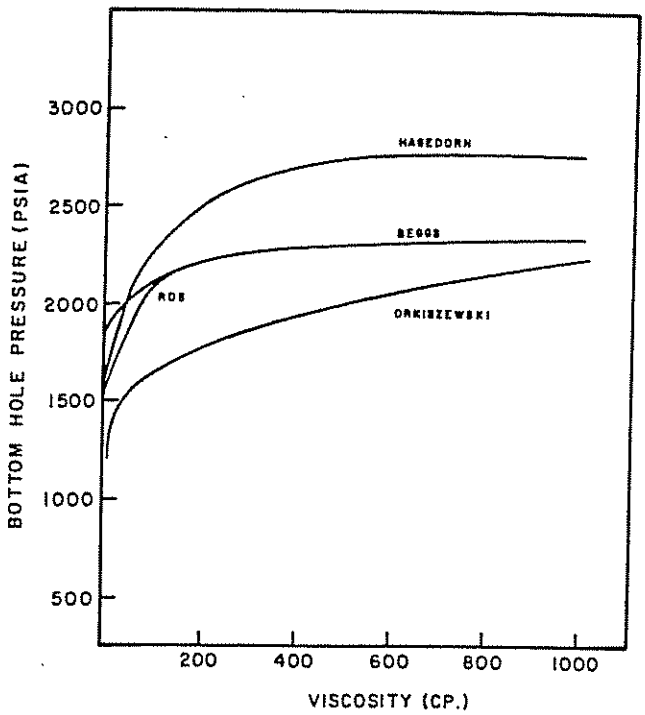


Fig. 1.4 Comparison of Correlations for Viscosity Effects

for each correlation between 5 cp and 1000 cp (at 100°F.) are as follows:

Hagedorn	1,204 psia
Ros	702 psia
Orkiszewski	1,000 psia
Beggs	487 psia

The Hagedorn correlation has the largest spread in pressure drop between the highest and lowest viscosities tested, indicating that this correlation is the most sensitive to viscosity changes, followed by the Orkiszewski, Ros, and Beggs.

This indicates that all the correlations predict changes in pressure drop differently for changes in viscosity. However, they all show rapid increases for viscosity increases between 5 cp and 100 cp (at 100°F.) This information shows the need for additional research in the area of viscous and heavy crudes. The effect of emulsions is an entirely different problem and is being investigated presently at The University of Tulsa.

1.1222 Effect of slippage or fall-back

Another area where correlations tend to differ considerably is in the case of flow rate for a particular flow configuration size where the fall-back of liquids becomes excessive. Field experience has shown that in some cases the oil production rate may be increased by replacing the tubing with tubing of a smaller diameter.

In many cases, a dying well can be brought back to flowing conditions by reducing the tubing diameter, and the rate can be increased on artificial lift wells. Low flow velocities up the tubing string result in increased fall-back of liquids and high pressure losses. The flow velocity is increased by reducing the pipe diameter, hence fall-back and pressure loss is reduced.

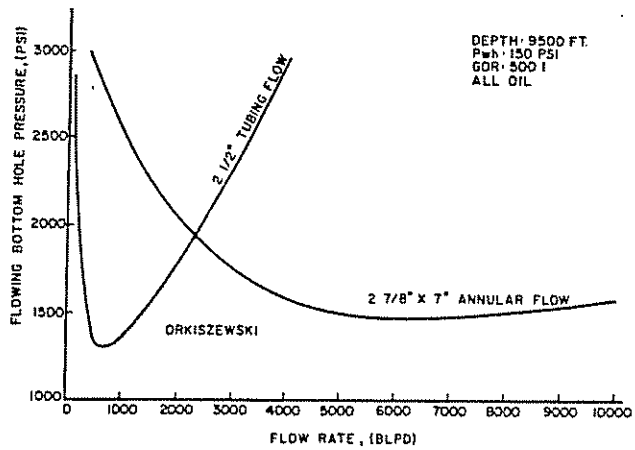


Fig. 1.5 Pressure Flow Rate Diagram Showing Rate for Minimum Flowing Pressure

Figure 1.5 shows the expected performance from a flowing well. For a given flow configuration, the bottom hole pressure decreases as flow rate decreases down to a point below which any further decrease in flow rate increases bottom hole pressure. Reducing the tubing size reduces the flow rate required before this limiting flow rate is reached.

During multiphase flow in a vertical or inclined well, density differences cause the heavier liquids to fall back and the lighter gas to slip by up the well. Hence, in a given section of pipe during upward flow, the fractional volume of liquid present is normally greater than the input fractional volume of liquid. The lower the flow rate, the greater the effect of fall-back.

One fallacy with present multiphase flow correlations is that they fail to correctly predict fall-back. A study was conducted to show how this phenomena of fall-back is handled by each of the following correlations:

(1) Hagedorn and Brown
(2) Duns and Ros
(3) Orkiszewski
(4) Beggs and Brill

For this study a typical well was chosen and the flowing bottom hole pressures were calculated by each of the four correlations for decreasing flow rates. Both tubing and annulus flow were considered. The well data are as follows:

Total depth	9500 ft
Flowing gas-oil ratio	500 scf/bbl
Wellhead pressure	150 psia
API gravity oil	35°
Specific gravity of gas	0.65
Water-cut	0.
Tubing I.D. (in.—for tubing flow)	1.995, 2.441, 2.952, 3.548
Annulus (in.—for annulus flow)	7—2⅞, 7½—3½, 5½—2⅜, 5½—2⅞
Bottom hole temperature	200°F.
Wellhead temperature	150°F.

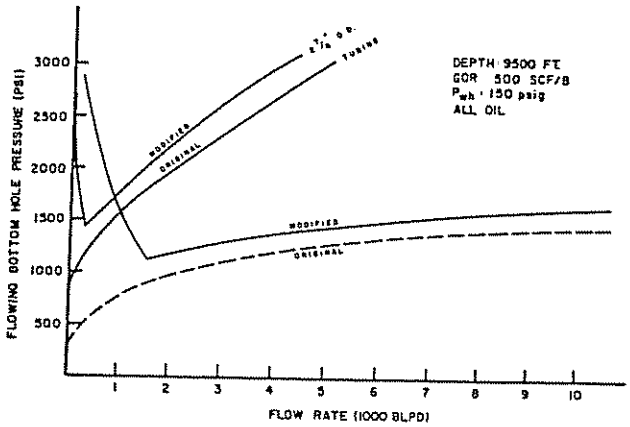


Fig. 1.6 Effects of Slippage or Fallback

All of the correlations show the expected reduction of bottom hole pressures with decreasing flow rates up to a point where the flowing pressure increases. Figure 1.6 shows the results predicted by the original and modified Hagedorn and Brown correlation for tubing flow. The bottom hole pressure reduces for decreasing flow rates. The original correlation failed to predict the upturn of increased flowing pressure with the lower rates. The present modified correlation does make this prediction.

This difference is due to the inaccuracies contained in the original Hagedorn and Brown correlation for predicting pressure drop in the bubble flow regime. The original Hagedorn and Brown correlation was modified to account for this limitation by including the Griffith correlation for bubble flow.[16]

A comparison of results predicted by the different correlations shows a few similarities and many differences (see Figure 1.7 and Table 1.4). The differences are:

(1) For a given tubing size the limiting flow rates below which the bottom hole pressure increases with decreasing flow rate are different for each correlation. This is illustrated in Table 1.1.
(2) The Orkiszewski correlation shows the bottom hole pressures at the limiting flow rates decrease with reducing tubing diameter. All of the others show the opposite trend.
(3) The Hagedorn and Brown and the Beggs and Brill correlations show rapid increase of bottom hole pressure from the limiting flow rate. The Orkiszewski and the Ros correlations show a more gradual rise.

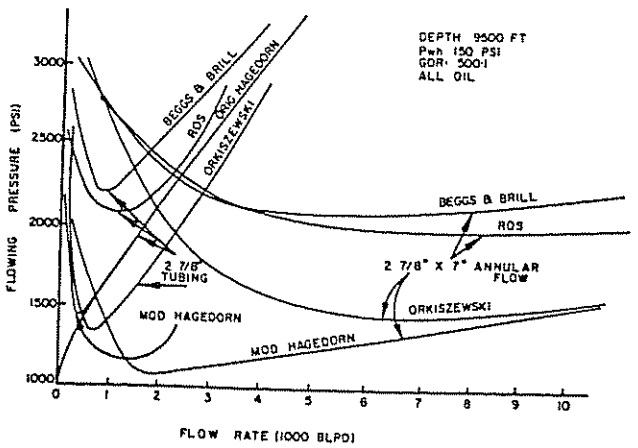


Fig. 1.7 Comparisons of Fallback of Liquids for Different Correlations

TABLE 1.4
LIMITING FLOW RATES (B/D) FOR THE VARIOUS CORRELATIONS AND PIPE SIZES TESTED

Tubing/annulus	Hagedorn	Ros	Orkiszewski	Beggs
2	150	450	500	400
2½	250	500	900	500
3½	400	1500	1000	900
4	450	2000	1900	1500
7 × 2⅞	1500	9000	6000	5000
5½ × 2⅜	900	4000	3000	3000
5½ × 2⅞	1000	1500	3000	2000
7½ × 3½	2000	1000	6000	6000

Points of similarity are as follows:

(1) All the correlations predict the expected limiting flow rates below which the bottom hole pressures increase with decreasing flow rates.
(2) All of the correlations indicate limiting flow rates decrease as tubing size decreases.

Apparently, the predicted rates for fall-back effect are too low for the Hagedorn correlation and possibly too high for the Ros. Additional research is needed to determine the correct values.

The previous discussions concerning comparison of correlations for predicting effects of viscosity and fall-back are also discussed to a limited extent in Volume 1.

1.123 Summary of multiphase flow

The multiphase flow correlations should be used for the design of all types of artificial lift. In particular, gas lift, electrical pumping, and jet pumping are probably the most critical, and industry is readily using the available correlations for these three methods of lift. Piston hydraulic pumping and sucker rod pumping are less critical but could also benefit from the use of such correlations in the future.

Therefore, two areas of flow exist where multiphase flow correlations may give questionable results: (1) low flow rates whereby slippage becomes excessive and (2) heavy viscous crudes where the viscosity is excessive.

Unless specifically stated, the working curves available should not be used for crude viscosities above 10 cp.

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Chapter 2

Beam pumping: design and analysis

by John J. Day and J. P. Byrd

2.1 INTRODUCTION

80—90% of all artificial lift wells are being produced by sucker rod pumping; the most common is the beam pumping system. Although the beam sucker rod system is mechanically simple and has proven to be long-lived and economical in operation, many factors must be considered in the design of a proper system. The design engineer must be thoroughly familiar with the function and complicating features of each part of the overall system if optimum performance is to be expected. Although it appears simple, in field practice the behavior of the beam and sucker rod system is surprisingly complex.

There are basic formulas for calculating the various factors affecting the selection of a suitable system, and this chapter will present those calculations and explore some of the problems involved in the design and analysis of a beam pumping installation. Note, however, that the formulas are inexact, and any analysis or design must be tempered with local experience.

2.2 GENERAL CONSIDERATIONS

Oil well pumping methods can be divided into two main groups:[1]

(1) *Rod systems.* Those in which the motion of the subsurface pumping equipment originates at the surface and is transmitted to the pump by means of a rod string
(2) *Rod-less systems.* Those in which the pumping motion of the subsurface pump is produced by means other than sucker rods.

Of these two groups, the first is represented by the beam pumping system and the second is represented by hydraulic and centrifugal pumping systems.

The beam pumping system consists essentially of five parts:

(1) The subsurface sucker rod-driven pump
(2) The sucker rod string which transmits the surface pumping motion and power to the subsurface pump. Also included is the necessary string of tubing and/or casing within which the sucker rods operate and which conducts the pumped fluid from the pump to the surface
(3) The surface pumping equipment which changes the rotating motion of the prime mover into oscillating linear pumping motion
(4) The power transmission unit or speed reducer
(5) The prime mover which furnishes the necessary power to the system

Figure 2.1 illustrates the various components of a complete beam pumping system. Items 3 and 4 are designated the beam pumping unit.

All beam type pumping unit geometrics fall into two distinct classes: (1) the Class I lever system which has its speed (gear) reducer rear-mounted with the fulcrum at mid beam, represented by the conventional unit and (2) the Class III lever system, a push-up geometry with its speed reducer front-mounted, represented by the air balance and Lufkin Mark II units, where the fulcrum is located at the rear of the beam. Figure 2.2 illustrates schematically these different pumping unit types. Figure 2.3 shows the arrangement of the surface equipment for a typical conventional unit.

The rotary motion of the crank arm is converted into oscillating motion by means of the walking beam. The crank arm is connected to the walking beam by means of the Pitman arm, and the walking beam is supported by the Sampson post and the saddle bearing.

The horse's head and the bridle—or the hanger cable arrangement—are used to ensure that the pull on the sucker rod string is vertical at all times so that no bending moment is applied to that part of the sucker rod string above the stuffing box. The polished rod and stuffing box combination is used to maintain a good liquid seal at the surface.

Such beam pumping units are available in a wide range of sizes. Stroke lengths vary from 12 to 240 inches. The stroke length for any particular unit can be varied with three or more different lengths being pos-

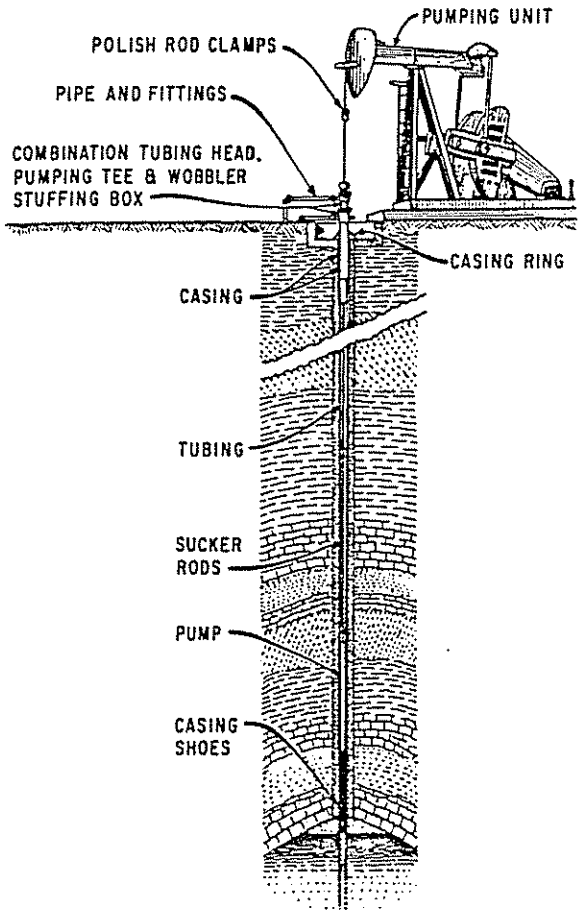


Fig. 2.1 Beam Pumping System

sible. These different strokes can be achieved by varying the position of the Pitman connection to the crank. Rod and structural ratings are expressed in terms of maximum allowable polished rod loads, which can vary from about 3,000 to over 42,000 lbs.

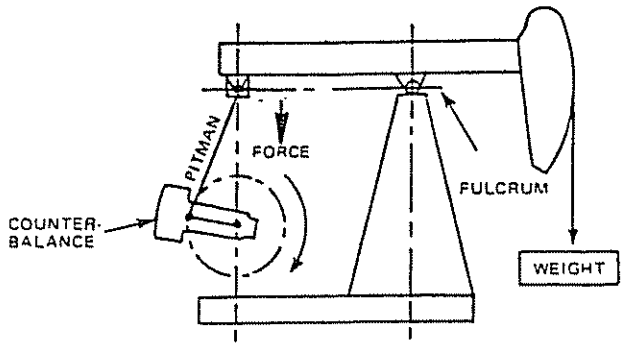
One of the most important aspects of pumping-installation design is the selection of counterbalance to reduce the size and torque requirements of the prime mover and gear reducer. For conventional beam units, counterbalance is accomplished by placing weights directly on the walking beam (this is usually done for the smaller units), or by attaching counterweights to the rotating crank arm, or by a combination of the two methods in the larger pumping units. In more recent designs, the rotary counterbalance can be adjusted by shifting the position of the counterweights on the crank arms.

Another important consideration in a pumping installation is the prime mover; two basic types are in widespread use today: electric motors and internal combustion engines.

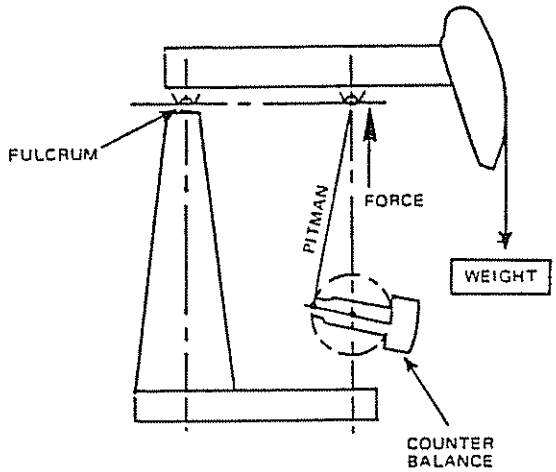
The main advantages of electric motors over gas engines are their lower initial cost and lower maintenance costs. Electric motors also provide dependable all-weather service and can be more easily fitted into an automatic system.

On the other hand, gas engines have the advantage of more flexible speed control and can operate over a wider range of load conditions. Fuel costs for gas engines may be lower than comparable energy costs for electric motors, although as fuel power costs increase this condition may be reversed.

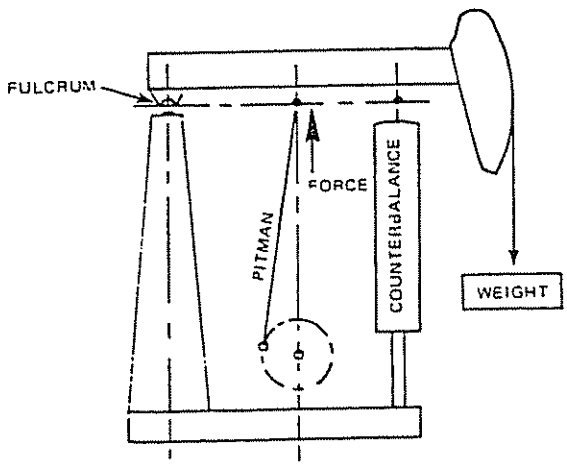
The selection of one type of prime mover over another may well depend upon local availability, fuel supply, local conditions, availability of maintenance, and personal experience or preference.



CLASS I LEVER SYSTEM - CONVENTIONAL UNIT.



CLASS III LEVER SYSTEM - LUFKIN MARK II.



CLASS III LEVER SYSTEM - AIR BALANCED SYSTEM.

Fig 2.2 (after Ref. 22)

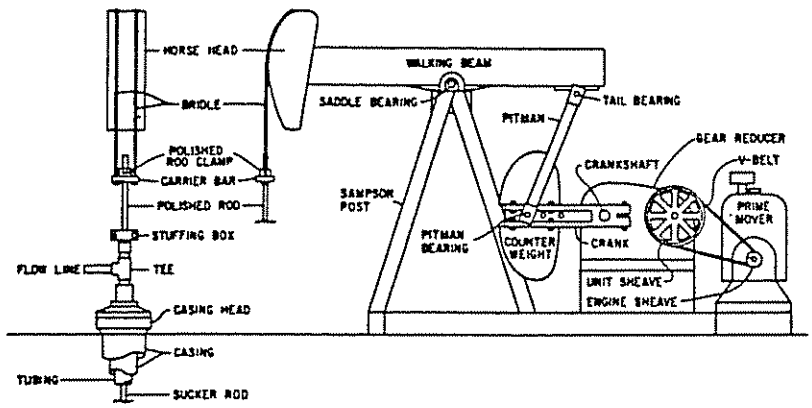


Fig. 2.3 *The Surface Equipment of a Beam Pumping Installation (after Ref. 5)*

Another important aspect of the surface transmission equipment is reducing the speed of the prime mover to a suitable pumping speed. This speed reduction is accomplished by means of the gear (or chain) reducer.

Any complete installation design must consider the behavior of all elements of the system. However, there are still some aspects of the engineering analysis of this method of pumping which have not been fully solved. In spite of this, there is an even greater need today for a better understanding of the pumping behavior because of the trend towards greater pumping depths.

The minimum amount of information which must be known, or assumed, to determine even approximate loads and pump displacements for sucker rod pumping unit installation design must include:[3]

(1) Fluid level (net lift, ft)
(2) Pump depth, ft
(3) Pumping speed, strokes per minute
(4) Length of surface stroke, in.
(5) Pump plunger diameter, in.
(6) Specific gravity of the fluid
(7) The nominal tubing diameter and whether it is anchored or unanchored
(8) Sucker rod size and design
(9) Unit geometry

With these factors, the designer should be able to calculate, with some degree of reliability, the following:

(1) Plunger stroke, in.
(2) Pump displacement, (B/D)
(3) Peak polished rod load, lb
(4) Minimum polished rod load, lb
(5) Peak (crank) torque, in.-lb or ft-lb, (when the unit's torque factor schedule is known)
(6) Polished rod horsepower
(7) Counterweight required, lb

The final solution to the design problem is reached, to a great extent, through trial and error methods.[3] Generally, three steps are required in designing an installation.

(1) A preliminary selection of components for the installation must be made.
(2) The operating characteristics of the preliminary selection are calculated by use of the basic formulas, tables, and figures to be presented.
(3) The calculated pump displacement and loads are compared with the volumes, load ratings, stresses, and other limitations of the preliminary selection.

It may then be necessary to make other selections and calculations to bring the limitations of the various components of the installation into agreement. Often, more than one selection of equipment and calculation of operating conditions is necessary before the optimum selection is made.

It is important to observe each of the elements of a pumping installation and tie them together into an operable system.

2.3 SUBSURFACE PUMPS

When reservoir pressure is too low to permit a well to flow by its own energy, some artificial means of supplementing that energy is required to lift the fluid to the surface. This can be accomplished through the use of subsurface pumps, which can be divided into four designs:[4]

(1) Rod-drawn pumps
(2) Hydraulic subsurface pumps
(3) Submerged centrifugal pumps
(4) Sonic pumps

Rod-drawn pumps can be divided into three basic types:

(1) Tubing pumps
(2) Insert (rod) pumps
(3) Casing pumps (a larger version of insert pumps)

All of these pumps are actuated by a sucker rod string and a surface pumping unit.

Any rod-drawn pump consists of four essential elements:

(1) A working barrel
(2) A plunger
(3) An intake valve (standing valve)
(4) An exhaust valve (traveling valve)

The basic difference between a tubing pump and an insert pump is the manner in which the working barrel is installed. With tubing pumps, the working barrel is connected to the bottom of the tubing and is run into the well as an integral part of the tubing string. With

insert pumps, the working barrel is an integral part of the entire subsurface pump assembly and is run as a unit on the sucker rod string inside of the tubing (or casing) string.

2.31 Tubing pumps

One basic advantage of tubing pumps is that they have a greater displacement than insert types because larger plunger diameters can be used within the larger working barrels of tubing pumps. However, tubing pumps have the disadvantage that the entire tubing string must be pulled from the well in order to service the working barrel. For this reason, tubing pumps are generally used only when the production desired cannot be obtained with an insert pump at the available stroke and speed combinations on the pumping unit selected. Obviously, the entire unit selected depends on economics and operating efficiency. Figure 2.4 is a schematic of a typical tubing pump.

The different types of tubing pumps can be classified:

(1) In relation to the type of working barrel used
(2) In relation to the standing valve arrangement
(3) In relation to the type of plunger used

Obviously, a large number of different types of these three classifications is possible. For example, a tubing pump with a particular type of working barrel could have any one of several different standing valve arrangements or types of plungers.

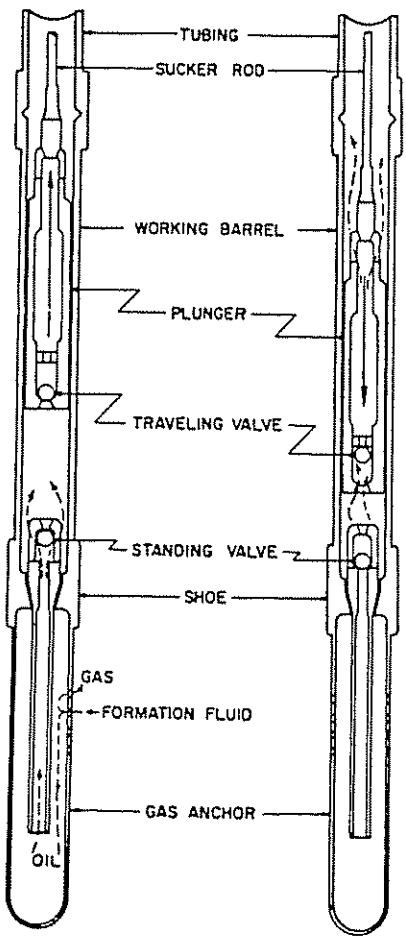


Fig. 2.4 The Subsurface Tubing Pump (after Ref. 3)

2.311 Tubing pumps classified according to type of working barrel

From the standpoint of the type of working barrel used, the tubing pumps can be subdivided into three groups:

(1) *Common working barrel pumps.* The barrel is made of cold-drawn seamless steel, cast iron, or corrosion-resisting alloys. With this type of pump, a length of cold-drawn tubing is polished on the inside to smooth the walls. The maximum diameter of the barrel is usually about ¼ inch less than the inside diameter of the tubing string to allow enough clearance for the plunger when lowered through the tubing string. The working barrel is connected to the bottom of the tubing string. These types of barrels are designated by the API as heavy-wall barrel design.

(2) *Full-liner working barrel pump.* This type of pump consists of a single hardened-steel tube (or corrosion-resistant alloy) which is machined and honed in one piece (not API).

(3) *Sectional liner pump.* The barrel of this type of pump consists of an outside steel jacket with short machined and accurately honed liners assembled end to end inside of the jacket. The various sections are made of hard steel, cast iron, or alloy steel and are held in position by the compressive force exerted by clamping collars at the ends of the jacket (not API).

2.312 Tubing pumps classified according to type of standing valve

From the standpoint of standing valve arrangement, the tubing pumps may have:

(1) A fixed standing valve
(2) A removable standing valve

The *fixed-type* standing valve is attached to the bottom of the tubing; in order to service the standing valve the tubing must be pulled from the well. This type of valve can be made oversize and can be quite effective in wells of low fluid levels, with viscous fluids, or when the working barrel does not fill completely.

The *removable-type* standing valve is placed in the working barrel before the barrel is run into the well or when the valve is dropped down the tubing from the surface and is pushed into place by the plunger. These valves are held in place by any one of several types of anchoring arrangements which employ special seating cups and frictional fit.

2.313 Tubing pumps classified according to type of plunger

From the standpoint of the type of plunger used, the most convenient method of classification is to divide the tubing pumps into groups according to the type of plunger seal used:

(1) *Pumps with cup-equipped plungers (soft-packed plungers).* This type is the oldest form of plunger seal for oil well sucker rod pumps (see Fig. 2.5). The cups are usually made of leather or rubber-impregnated canvas. However, new types of synthetic materials for cups and cup rings are becoming available each year, and many of them have excellent corrosion resistance. On the upstroke, pressure exerted by the fluid column forces the cups to expand and form a seal between the lip of the cup and the wall of the barrel. On the downstroke,

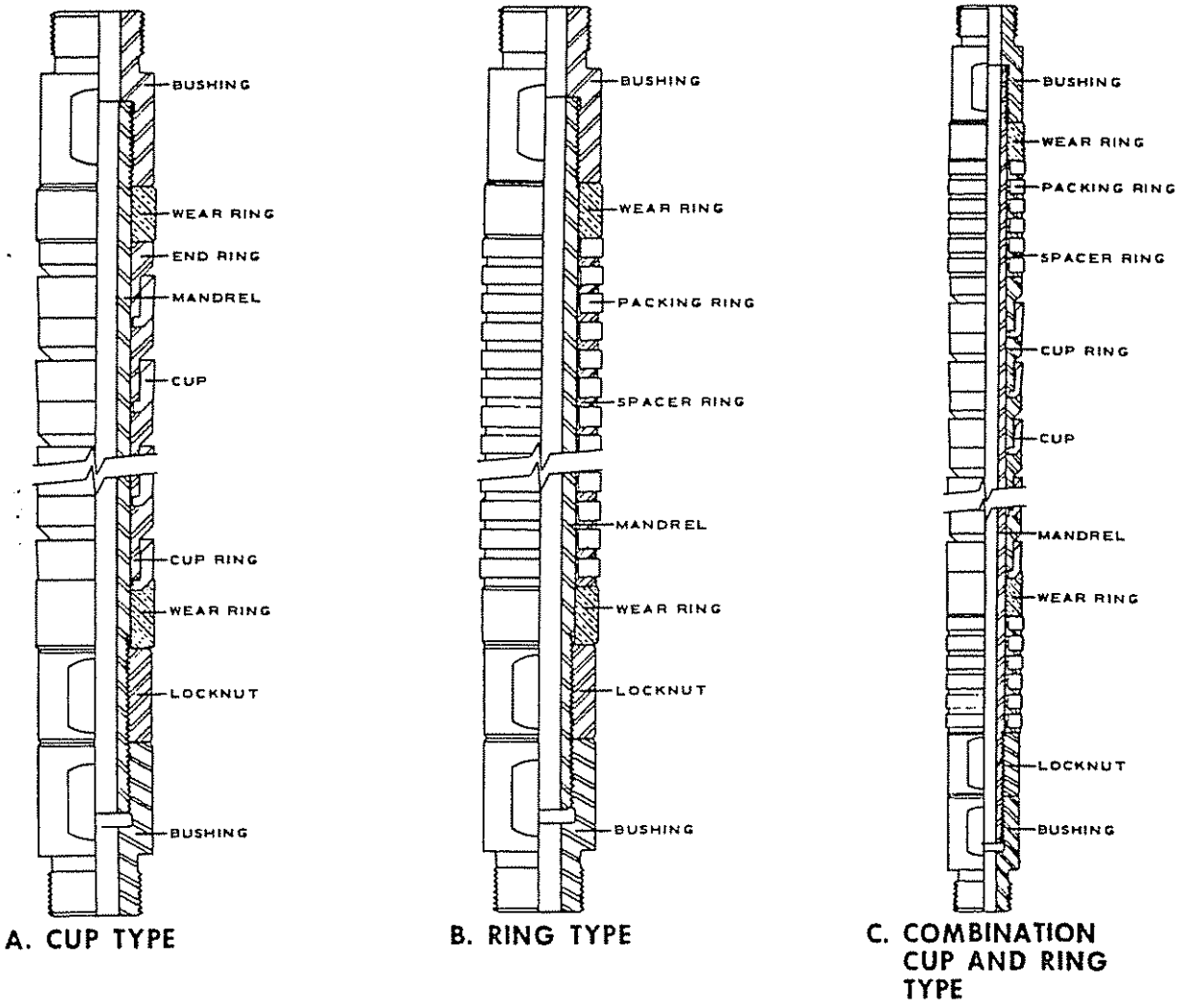


Fig. 2.5 Soft-Packed Plungers (after Ref. 16)

pressure is equalized on both sides of the cup and the cup collapses inward, allowing the plunger to fall freely. Soft-packed plungers can be used with all types of rod and tubing pumps but normally not below 5000 feet under most circumstances.[16]

(2) *Pumps with metal plungers.* Metal plungers may be made of cast iron or steel and are made with a smooth sealing surface called a "plain" metal plunger or with a grooved surface called a "grooved" metal plunger (see Fig. 2.6). The metal-to-metal plungers depend upon an extremely close clearance (depending upon fluid viscosity) to provide the fluid seal. These plungers can be made from a single piece of tubing, or short sections of tubing can be assembled on a core tube to the desired length. These plungers generally wear better than cup types and are used in deeper wells (greater than about 7,000 ft).

(3) *Pumps with concentric tubes.* This group includes pumps which depend upon both the fluid seal and an unusually long distance that the fluid would have to travel in order to slip past the plunger. This type utilizes three tubes, as illustrated in Figure 2.7. One advantage of this arrangement is that the great length of the fluid seal eliminates the necessity of a very close plunger fit.

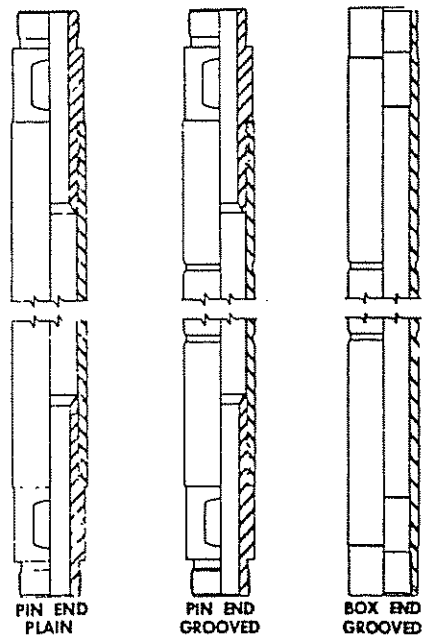


Fig. 2.6 Plain & Grooved Metal Plungers (after Ref. 16)

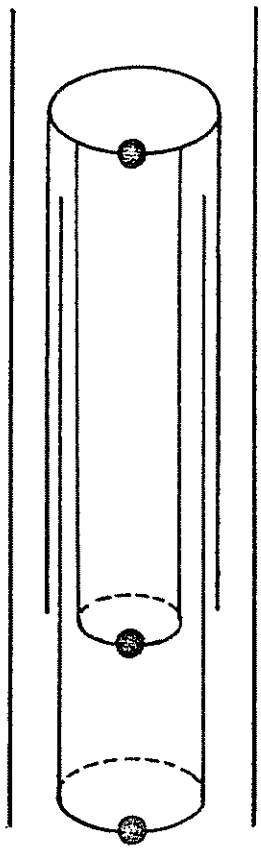


Fig. 2.7 Schematic of Concentric Type of Plunger

2.32 Insert pumps

The advantage of insert pumps is that they connect to the sucker rod string, and the entire assembly can be removed from the well merely by pulling the rod string. With this type of pump, the working barrel is lowered on rods; consequently, some means must be provided to secure the barrel into the bottom of the tubing in order to provide fluid packoff and to facilitate the relative motion of the working barrel and plunger. Several arrangements are used for this purpose. Seating cups can be provided on the working barrel, or a special seating housing may be provided on the bottom of the tubing. Hold-down anchors can also be used at the top or bottom of the barrel.

From the standpoint of operation, insert pumps can be divided into two groups:

(1) *Inverted pump (traveling pumps).* In this type of pump, the plunger assembly is stationary, and the working barrel is the moving part. The advantage of this arrangement is that the moving barrel gives sand less chance to settle between the working barrel and the tubing. However, frictional wear can be considerable.

(2) *Stationary insert pump.* In this type of pump the working barrel is stationary and the plunger is the moving part.

From the standpoint of working barrel and plunger used, most of the considerations already discussed for tubing pumps also apply to insert pumps.

Advantages and disadvantages of thin and heavy-wall (one-piece) barrels and liner barrel construction are (liner barrel construction is no longer API):[16]

(1) With the same length barrel, a thin wall barrel pump will produce a greater volume of fluid than a liner barrel pump of the same outside diameter because it can be made with a larger inside diameter.
(2) Because of its simpler construction, the one-piece barrel is lower in price than a liner barrel of the same length and outside diameter.
(3) Maintenance costs are lower with a one-piece barrel pump than with a liner pump due to fewer parts and easier servicing.
(4) Where a top-anchored rod pump is desirable, a pump equipped with a one-piece barrel can be used in deeper wells than is considered safe with a liner pump. The one-piece barrel can stretch under the load of the fluid column without hurting the pump; the jacket used in a liner pump stretches under the fluid-column load in deep wells, and the liners become loose and misaligned. A "locked-liner" construction overcomes this disadvantage and can be used with a top anchor in deep wells.
(5) Liner and one-piece heavy wall barrels are not limited in length as are one-piece thin wall barrels, and they can be used with center connecting collars.
(6) Sections, from a sectional-liner barrel (often of cast material), can be rebored and reused to reduce replacement cost.
(7) A closer tolerance between the barrel and the plunger can be achieved with a sectional-liner barrel than with a one-piece barrel or liner. This is an important consideration in wells with high bottom-hole pressures where an accurate fit is necessary to reduce the plunger-slippage rate. In addition, the possibilities of sand pockets are lessened.
(8) Special corrosive and abrasive metals that cannot stand the stresses imposed upon them in long, solid one-piece barrel construction, or those that become distorted when formed in great lengths, can be made into short liners and used in sectional-liner construction.

2.33 Casing pumps

This group of pumps includes all pumps which use casing instead of tubing through which fluid is pumped to the surface. A casing pump is run into the well on sucker rods, and a packer, either on the top or bottom of the working barrel, provides the fluid packoff between the working barrel and the casing. No tubing is used in this type of installation.

Generally, the casing pump is just a larger version of the insert pump and is set and operated in essentially the same manner. The casing pump is a large-volume, shallow-depth pump and is particularly suitable for installations where high production rates are required.

2.34 The pumping cycle

Figure 2.8 is a schematic of the various stages in a pumping cycle. The cycle can be applicable to tubing, insert, or casing pumps.

In Figure 2.8 (a) the plunger is moving down near the bottom of the stroke.[c] Fluid is moving up through the

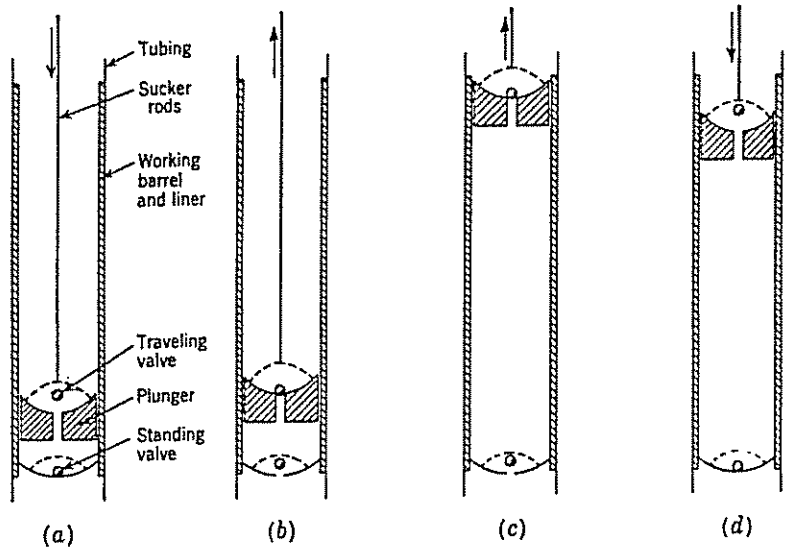


Fig. 2.8 *The pumping cycle: (a) plunger moving down, near bottom of stroke; (b) plunger moving up, near bottom of stroke; (c) plunger moving up, near top of stroke; (d) plunger moving down, near top of stroke. (after Ref. 6)*

open traveling valve, while the weight of the fluid column in the tubing is supported by the standing valve, which is consequently closed.

In (b) the plunger is moving up near the bottom of the stroke. The traveling valve is now closed; consequently, the load due to the fluid column has been transferred from the tubing to the rod string.

In (c) the plunger is moving up, near the top of the stroke. The traveling valve is closed and the standing valve is open if there is any production at all from the well. As the fluid column is lifted, the pump barrel is evacuated, permitting the inflow of new fluid.

In (d) the plunger is moving down near the top of the stroke. The standing valve is closed by the increased pressure resulting from the compression of the fluids in the volume between the standing and traveling valves. The traveling valve is open.

After the plunger reaches the bottom of the stroke, the cycle is repeated.

2.35 API pump classifications

The American Petroleum Institute has adopted a classification system for subsurface pumps. These classifications, taken from API Recommended Practice 11AR, are shown in Figure 2.9.

The types of pumps are as follows:

(1) Tubing type with or without extension shoe and nipple
(2) Rod type, stationary barrel, with top hold-down
(3) Rod type, stationary barrel, with bottom hold-down
(4) Rod type, traveling barrel

Complete pump designations are given in Figure 2.10.

2.36 Pump size selection

For a given pumping depth and volume of fluid to be produced, there is an optimum size of pump bore which will result in effective pump plunger travel and maintain moderate speed of operation. If the plunger is too large, unnecessarily high loads will be imposed upon the equipment and plunger undertravel can result in inefficient operations. On the other hand, if the plunger is too small, pumping speeds become too high and the increased acceleration (inertial) effects can result in increased peak loads on the equipment. The basic factor in the selection of a suitable pump size is the volume of fluid displaced by the pump per inch of each stroke. This volume displacement will depend upon the diameter of the pump bore.

The total theoretical pump displacement can be determined by:

$$PD = A_p(\text{in.}^2) \times S_p(\text{in./stroke}) \times N(\text{strokes/min.}) \times \frac{1440 \text{ min/day}}{9702 \text{ in}^3/\text{bbl}} = 0.1484\, A_p S_p N \quad (2.1)$$

where:

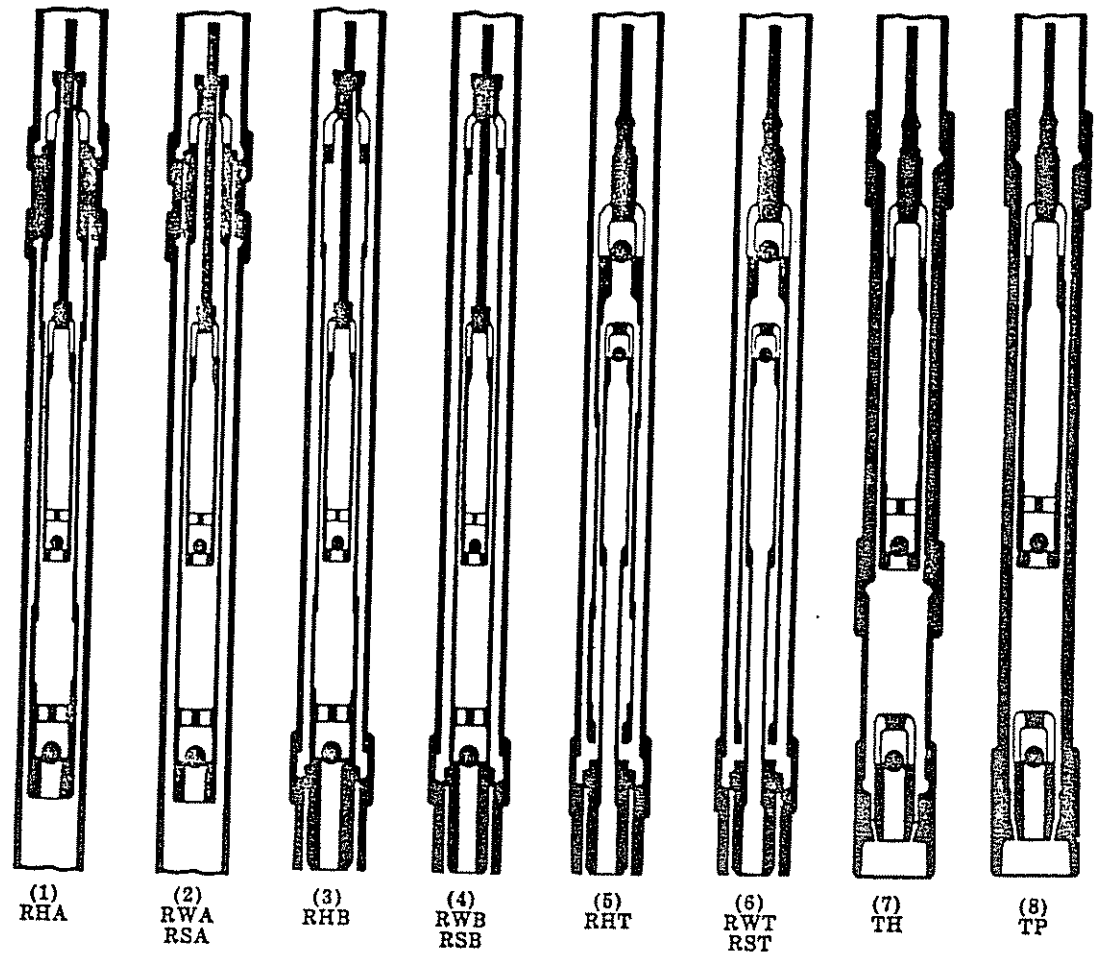
PD = Total pump displacement, B/D
A_p = The cross-sectional area of the pump plunger in square inches
S_p = The effective plunger stroke, in.
N = The pumping speed in number of strokes per minute

A pump constant, K, for any given plunger size is determined from:

$$K = 0.1484\, A_p \quad (2.2)$$

Thus, the pump displacement for a given plunger size and for a given combination of pumping speed and stroke can be determined from:

$$PD = KS_pN \quad (2.3)$$



(1) RHA: Rod, Stationary Heavy Wall Barrel, Top Anchor Pump
(2) RWA: Rod, Stationary Thin Wall Barrel, Top Anchor Pump
RSA: Rod, Stationary Thin Wall Barrel, Top Anchor, Soft-Packed Plunger Pump
(3) RHB: Rod, Stationary Heavy Wall Barrel, Bottom Anchor Pump
(4) RWB: Rod, Stationary Thin Wall Barrel, Bottom Anchor Pump
RSB: Rod, Stationary Thin Wall Barrel, Bottom Anchor, Soft-Packed Plunger Pump
(5) RHT: Rod, Traveling Heavy Wall Barrel, Bottom Anchor Pump
(6) RWT: Rod, Traveling Thin Wall Barrel, Bottom Anchor Pump
RST: Rod, Traveling Thin Wall Barrel, Bottom Anchor, Soft-Packed Plunger Pump
(7) TH: Tubing, Heavy Wall Barrel Pump
(8) TP: Tubing, Heavy Wall Barrel Soft-Packed Plunger Pump

Fig. 2.9 API Pump Classification (after Ref. 15)

Table 2.1 shows plunger areas and pump constants for various pump sizes.

EXAMPLE PROBLEM #1

Calculate the pump constant, K, for a 1½ in. plunger.

Solution

$$K = 0.1484\ A_p$$
$$A_p = (\pi/4)(1.5)^2 = 1.767\ \text{in.}^2$$

Thus

$$K = 0.1484\ (1.767)$$
$$= 0.262\ \text{B/D/in./spm}$$

TABLE 2.1 (after Ref. 2)
TABLE OF PUMP CONSTANTS

Plunger diam in.	Area of plunger sq in. (A_p)	Constant (K)	Plunger diam in.	Area of plunger sq in. (A_p)	Constant (K)
5/8	0.307	0.046	*1 3/4	2.405	0.357
3/4	0.442	0.066	*1 25/32	2.488	0.370
15/16	0.690	0.102	*2	3.142	0.466
*1	0.785	0.117	*2 1/4	3.976	0.590
*1 1/16	0.886	0.132	*2 1/2	4.909	0.728
1 1/8	0.994	0.148	*2 3/4	5.940	0.881
*1 1/4	1.227	0.182	*3 3/4	11.045	1.640
*1 1/2	1.767	0.262	*4 3/4	17.721	2.630

*API sizes

The basic types of pumps and letter designation covered by this specification are as follows:

Type of Pump	Letter Designation: Metal Plunger Pumps, Heavy-Wall Barrel	Letter Designation: Metal Plunger Pumps, Thin-Wall Barrel	Letter Designation: Soft-packed Plunger Pumps, Heavy-Wall Barrel	Letter Designation: Soft-packed Plunger Pumps, Thin-Wall Barrel
Rod Pumps				
Stationary Barrel, Top Anchor	RHA	RWA	.....	RSA
Stationary Barrel, Bottom Anchor	RHB	RWB	.....	RSB
Traveling Barrel, Bottom Anchor	RHT	RWT	.....	RST
Tubing Pumps	TH	.....	TP	.....

Complete pump designations include: (1) nominal tubing size, (2) basic bore diameter, (3) type of pump, including type of barrel and location and type of seating assembly, (4) barrel length, (5) plunger length, and (6) total length of extensions when used, as follows:

XX - XXX X X X X X - X - X

- Total length of extensions, whole feet.
- Nominal plunger length, feet.
- Barrel length, feet.
- Type seating assembly; C—Cup type; M—Mechanical type
- Location of seating assembly; A—Top; B—Bottom; T—Bottom, traveling barrel
- Type barrel; H—Heavy-wall, W—Thin-wall (For metal plunger pumps); S—Thin-wall, P—Heavy-wall (For soft-packed plunger pumps)
- Type pump; R—Rod; T—Tubing
- Pump bore (basic): 125—1¼ in. *(31.8 mm)*; 150—1½ in. *(38.1 mm)*; 175—1¾ in. *(44.5 mm)*; 178—1 25/32 in. *(45.2 mm)*; 200—2 in. *(50.8 mm)*; 225—2¼ in. *(57.2 mm)*; 250—2½ in. *(63.5 mm)*; 275—2¾ in. *(69.9 mm)*
- Tubing size; 15—1.900 in. OD. *(48.3 mm)*; 20—2⅜ in. OD. *(60.3 mm)*; 25—2⅞ in. OD. *(73.0 mm)*; 30—3½ in. OD. *(88.9 mm)*

Example: A 1¼ in. *(31.8 mm)* bore rod type pump with a 10 ft. *(3.048 m)* heavy wall barrel and 2 ft. *(0.610 m)* of extensions, a 4 ft. *(1.219 m)* plunger, and a bottom cup type seating assembly for operation in 2⅜ in. *(60.3 mm)* tubing, would be designated as follows:

20-125 RHBC 10-4-2

In addition to the pump designation described above, it is necessary for the purchaser to provide the following additional information:

a. Barrel material
b. Plunger material
c. Plunger clearance (fit)
d. Valve material
e. Length of each extension

Fig. 2.10 Pump Designation (after Ref. 8)

The actual production rate at the surface, Q, may be less than the total theoretical pump displacement because of volumetric efficiency, E_v, of the pump.

$$E_v = Q/PD \text{ or } Q = E_vPD \qquad (2.4)$$

Volumetric efficiencies can vary over a wide range but are commonly 70-80%. Volumetric efficiencies are affected by pump slippage and fluid properties such as gas content and foaming characteristics of the fluid ("spongy" fluids). These are difficult properties to determine. An important consideration in many beam pumped wells is fluid shrinkage between the bottomhole pump and the wellhead. As the fluid is elevated and gas breaks out of solution, there is a significant difference between the volumetric displacement of the bottomhole pump and the volume of the fluid delivered to the surface. This effect is denoted by a shrinkage factor greater than 1.0—indicating that the bottomhole pump must displace more fluid by some additional percentage than the volume delivered to the surface. For example a shrinkage factor of 1.15 means that the bottomhole pump must displace 115 barrels of fluid to deliver 100 barrels of fluid to the surface. Local experience will generally give the best estimate of volumetric pump efficiency.

EXAMPLE PROBLEM #2

A well is equipped with a 1½ in. plunger. The pumping speed is 20 spm and the effective plunger stroke

is 55 in. Production at the surface is 210 B/D of fluid of specific gravity 0.85.

Calculate the total pump displacement and the volumetric efficiency of the pump.

Solution

PD = Total pump displacement, B/D
$= KS_pN$

From Table 2.1, K = 0.262 for 1½ inch plunger

Thus,

PD = (0.262) (55) (20)
= 288 B/D at the pump

Volumetric efficiency, E_v, is:

$E_v = (Q/PD)\,100 = (210/288)\,100$
$= 72.9\%$

EXAMPLE PROBLEM #3

The theoretical pump displacement is 250 B/D and pump efficiency is 75%. Calculate the surface production rate.

Solution

$E_v = Q/PD$

or

$Q = PDE_v = (250)\,(0.75)$
$= 187.5$ B/D

Merely considering a pump size, speed, and stroke combination will not necessarily ensure that the proper size pump has been selected for a given pumping installation. The selection of the optimum size plunger for a desired production rate from some given depth is important in obtaining high efficiencies and in preventing unnecessarily high loads on the rod string and the surface equipment.

As an example, Table 2.2 can be used *as a guide* in determining pump plunger sizes for "optimum" conditions when surface stroke is less than 74 inches. Such a table should be used only for preliminary selections.

Table 2.3 gives other pump data, and Table 2.4 gives tubing data.

TABLE 2.2 (after Ref. 2)
PUMP PLUNGER SIZES RECOMMENDED FOR "OPTIMUM" CONDITIONS

Net lift of fluid ft	Fluid production—Barrels per day—80 pct efficiency									
	100	200	300	400	500	600	700	800	900	1000
2000	1½ 1¼	1¾ 1½	2 1¾	2¼ 2	2½ 2¼	2¾ 2½	2¾	2¾	2¾	2¾
3000	1½ 1¼	1¾ 1½	2 1¾	2¼ 2	2½ 2¼	2½ 2¼	2¾ 2½	2¾	2¾	2¾
4000	1¼	1¾ 1½	2 1¾	2¼ 2	2¼ 2	2¼	2¼	2¼		
5000	1¼	1¾ 1½	2 1¾	2 1¾	2¼ 2	2¼				
6000	1¼	1½ 1¼	1¾ 1½	1¾						
7000	1¼ 1⅛	1½ 1¼			In this tabulation surface pumping strokes up to 74 in. only are considered.					
8000	1¼ 1⅛									

TABLE 2.3 (after Ref. 3)
PUMP CONSTANTS

1	2	3	4
Plunger diameter, in. D_p	Plgr. diam. squared sq. in. D_p^2	Fluid load factor* lb per ft $(0.340 \times D_p^2)$	Pump factor $(0.1166 \times D_p^2)$
1 1/16	1.1289	0.384	0.132
1¼	1.5625	0.531	0.182
1½	2.2500	0.765	0.262
1¾	3.0625	1.041	0.357
2	4.0000	1.360	0.466
2¼	5.0625	1.721	0.590
2½	6.2500	2.125	0.728
2¾	7.5625	2.571	0.881
3¾	14.0625	4.781	1.640
4¾	22.5625	7.671	2.630

*For fluids with specific gravity of 1.00.

TABLE 2.4 (after Ref. 3)
TUBING DATA

1	2	3	4	5
Tubing size	Outside diameter, in.	Inside diameter, in.	Metal area, sq. in.	Elastic constant, in. per lb ft E_t
1.900	1.900	1.610	0.800	0.500×10^{-6}
2⅜	2.375	1.995	1.304	0.307×10^{-6}
2⅞	2.875	2.441	1.812	0.221×10^{-6}
3½	3.500	2.992	2.590	0.154×10^{-6}
4	4.000	3.476	3.077	0.130×10^{-6}
4½	4.500	3.958	3.601	0.111×10^{-6}

EXAMPLE PROBLEM #4

A pump is to be set in a well at the working fluid level of 4000 feet. It is desired to produce 400 B/D of fluid at the surface. If local experience indicates that pump efficiency is 80% what size pump would you recommend for this well?

Solution

The net lift for this well is 4000 ft. Thus, from Table 2.2 the suggested pump size to be installed is 2 in. or 2¼ in. The exact size will depend upon other factors and other considerations to be discussed.

For example, pumping speed and effective plunger stroke will be important determining factors. More precise methods will be dealt with later.

One could also make an initial selection of the pump plunger diameter from Equation 2.3, which can be written in the form:

$$PD = 0.1484\ A_p\ S_p\ N = K\ S_p\ N$$

$$= (0.1484)\left(\frac{\pi}{4}\right)(D_p^2)\ S_p N$$

$$= 0.1166\ D_p^2\ S_p\ N \qquad (2.5)$$

The effective plunger stroke should be about 80% or more of the surface stroke ($S_p/S = 0.80$).[29] Thus, Equation 2.5 could be written:

D_p^2 = plunger diameter squared, sq in.

$$D_p^2 = \frac{(PD)}{(0.1166)(0.8\,S)\,N} = \frac{10.72\ (PD)}{SN} \qquad (2.6)$$

where:

S = surface stroke, in.

Once the square of the plunger diameter has been calculated from Equation 2.6, find the plunger diameter from Table 2.3 which has a value of D_p^2 just larger than the calculated value. That would be the indicated required pump size.

After the pump size and type have been determined, be certain that the maximum size for the given tubing size is not exceeded. Maximum pump size and type are given in Table 2.5.

TABLE 2.5 (after ref. 22)
MAXIMUM PUMP SIZE AND TYPE

Pump type	Tubing size, in.			
	1.900	2⅜	2⅞	3½
Tubing one-piece, thin-wall barrel (TW)	1½	1¾	2¼	2¾
Tubing one-piece, heavy-wall barrel (TH)	1½	1¾	2¼	2¾
Tubing liner barrel (TL)	—	1¾	2¼	2¾
Rod one-piece, thin-wall barrel (RW)	1¼	1½	2	2½
Rod one-piece, heavy wall barrel (RH)	1 1/16	1¼	1¾	2¼
Rod liner barrel (RL)	—	1¼	1¾	2¼

Figures 2.11, 2.12, and 2.13 can be used to estimate the maximum practical pumping speed for conventional, air balanced, and Mark II units, respectively.[22] In most cases maximum free-fall of the rod string is primarily controlled by forces of retardation in the well rather than by pumping unit geometry. In some wells maximum free-fall pumping speeds may be considerably greater than these curves would indicate, while in others the maximum pumping speeds may be less than shown. For instance, on a shallow well in Venezuela the maximum pumping speed for a 120-in. stroke Mark II unit predicted 13.1 spm. In actual practice, however, the Mark II unit operated satisfactorily for years at 20 spm, nearly 50% faster than predicted.

CLASS PROBLEMS

(1) Re-work Example Problem #4 for a conventional unit with a surface stroke of 64 in.
(2) Do the same for an air balanced unit.
(3) Do the same for a Mark II unit.
(4) Repeat these problems for all three units for a surface stroke of 54 in.
(5) Repeat for all three units for a surface stroke of 74 in.
(6) Repeat for all three units for a surface stroke of 240 in.

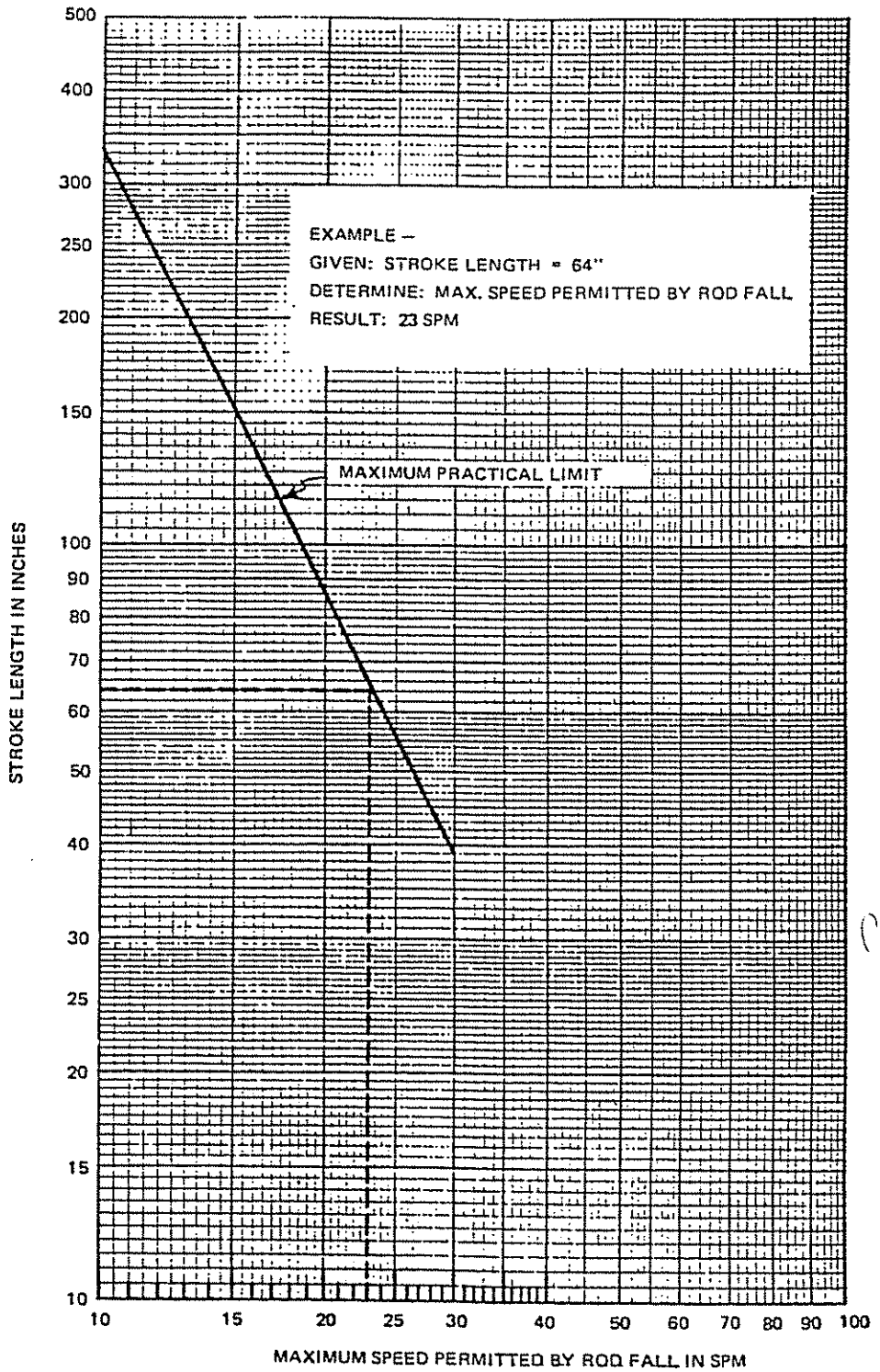


Fig. 2.11 Maximum Practical Pumping Speed, Conventional Unit (after Ref. 22)

2.4 THE SUCKER ROD STRING—GENERAL CONSIDERATIONS

The sucker rod string is a complex vibratory system which transmits energy from the surface equipment to the subsurface pump. The design of a pumping installation, including the surface and subsurface equipment, requires the selection of a suitable sucker rod string. Before this subject is discussed, let us review some general aspects of sucker rods.

The main constituent of all sucker rods is iron. Most steel rods contain over 90% iron in the composition of the rods. Pure iron, however, is soft and weak, and other elements must be added to the composition to provide the necessary strength (and other properties).

The various materials which may be added to make

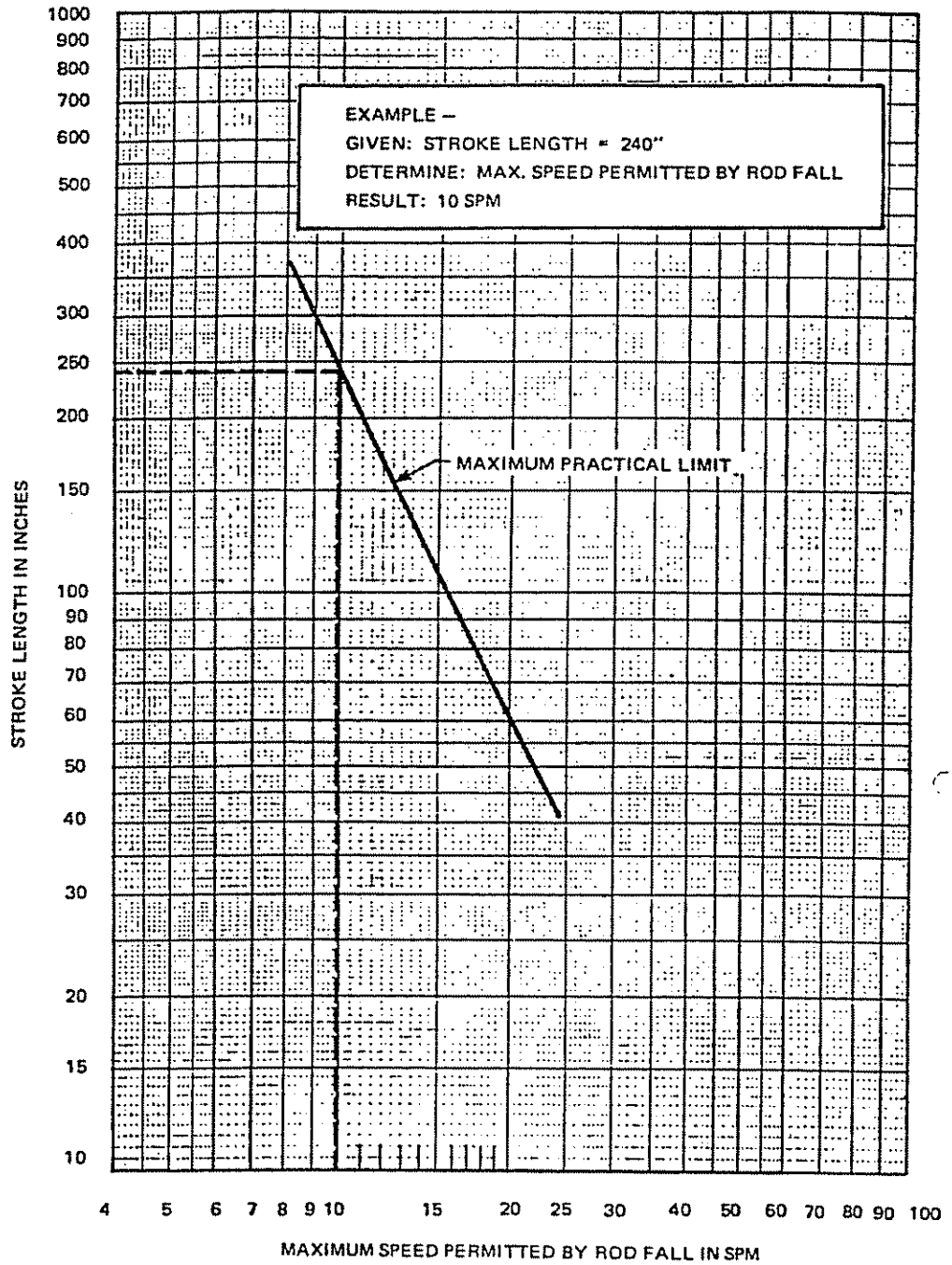


Fig. 2.12 Maximum Practical Pumping Speed, Air Balanced Unit (after Ref. 22)

up these alloy steels and the effects of the various additivies are summarized as follows:[2]

Carbon is an essential ingredient of all steels and is added (or increased in content) to increase strength, hardness, and susceptibility to heat-treatment. However, as the carbon content increases, the corrosion-resistance, the ductility, and the impact-resistance tend to decrease, although the magnitude of these effects can be controlled somewhat by heat-treatment. The main advantage of other additives is to permit the carbon content to be reduced without reducing the strength of the steel.

Manganese combines with sulphur to make the steel less brittle, and it also acts as a deoxidizer to reduce the formation of iron oxides which tend to weaken the alloy. Some sucker rods containing over 1% manganese have higher strength than can be obtained with the same carbon content alone.

Silicon, much like manganese, is useful as a deoxidizer in refining high-grade steels. Most sucker rods contain about 0.15 to 0.35% of this element.

Nickel is added to combat corrosive conditions encountered in oil wells resulting from hydrogen sul-

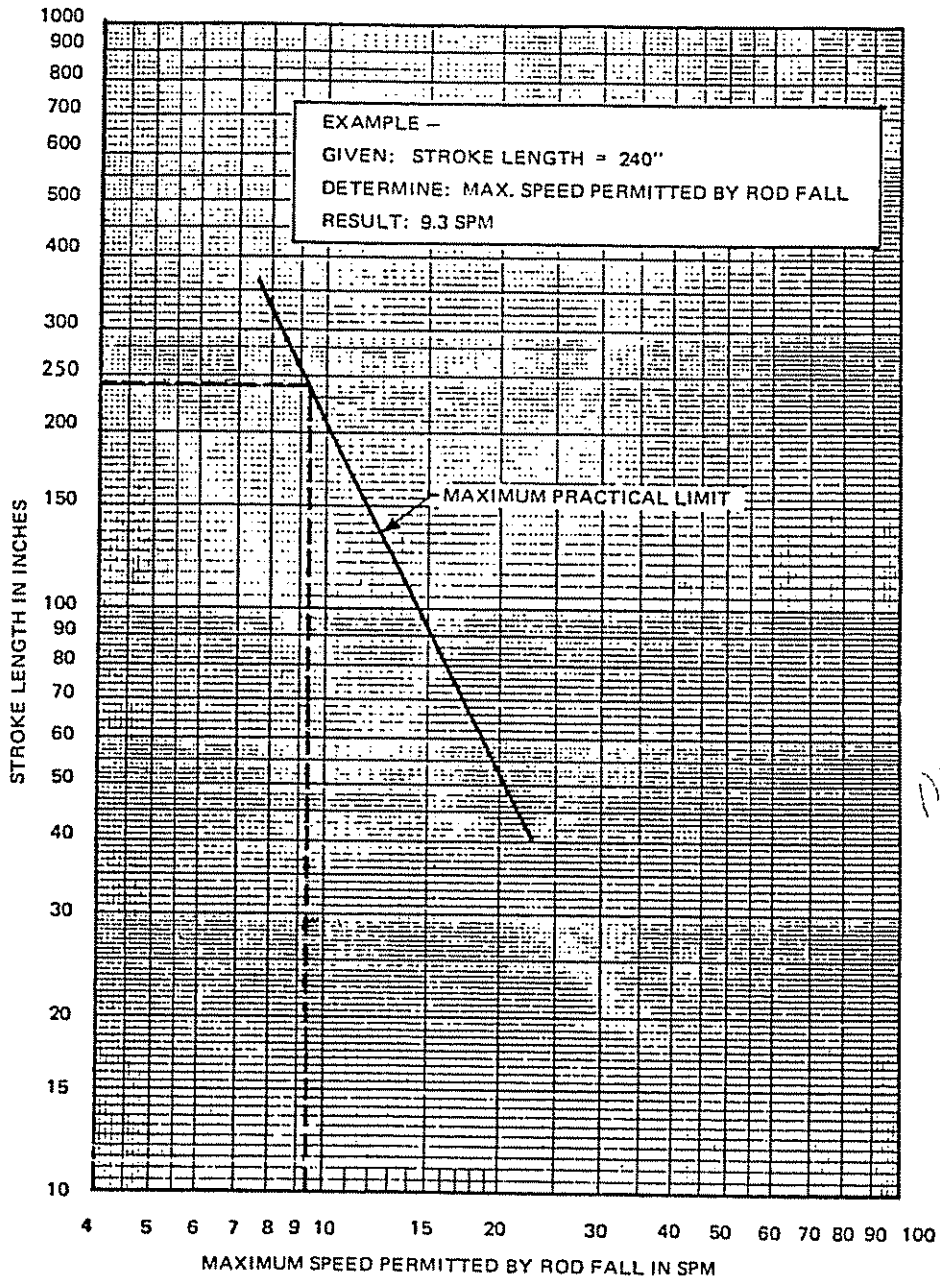


Fig. 2.13 Maximum Practical Pumping Speed, Mark II Unit (after Ref. 22)

fide or other corrosive gases. Nickel also has a hardening effect on steel by dissolving in ferrite.

Vanadium increases the hardenability of steel even when present in small amounts. It promotes a fine-grained structure and retards softening during tempering.

Copper is generally added for resistance to atmospheric and other corrosive environments. Steels containing more than 0.6% copper have a pronounced tendency toward precipitation-hardening.

Boron is used in alloy steel for one purpose—to increase hardenability. It is powerful, and only a few thousandths of 1% is ordinarily added.

Chromium forms a stable carbide and contributes considerably to the hardenability of steels. It also improves the corrosion-resistance of steels in air and other environments. Chromium is found in high percentages in all stainless type steels. For oil well service, chromium appears to be less effective than nickel in resisting hydrogen sulfide corrosion.

Molybdenum is one of the most powerful hardening agents among the alloying elements, although it is not as effective as carbon. It strengthens sucker rods by improving their response to heat-treatment.

In general, maximum allowable working stresses should not be higher than about 30,000 to 40,000 psi —although some sucker rods, such as the newer "Elec-

tra" series manufactured by Oilwell Division of United States Steel, are rated at 40,000 to 50,000 psi maximum. In corrosive fluids, the maximum allowable stress must be decreased accordingly.

Sucker rods are available in several standard sizes: ½, ⅝, ¾, ⅞, 1.0, and 1⅛-inch diameters. Table 2.6 gives typical sucker rod data.

TABLE 2.6 (After Ref. 3)
SUCKER ROD DATA

1	2	3	4
Rod size	Metal area, sq in.	Rod weight in air, lb per ft	Elastic constant, in. per lb ft E_r
½	0.196	0.72	1.990×10^{-6}
⅝	0.307	1.13	1.270×10^{-6}
¾	0.442	1.63	0.883×10^{-6}
⅞	0.601	2.22	0.649×10^{-6}
1	0.785	2.90	0.497×10^{-6}
1⅛	0.994	3.67	0.393×10^{-6}

Complete API specifications for sucker rods can be found in API Standard 11-B, "API Sucker Rod Specifications."

2.41 Design of the sucker rod string

For depths greater than about 3500 feet, it is usually desirable to use a tapered rod string which consists of different lengths of rods of different sizes. The smallest rods would be placed at the bottom of the string; the largest rods would be placed at the top of the string where the load on the string is the greatest. The maximum and minimum loads expected during the pumping cycle for any given rod string must be determined as accurately as possible to design or select suitable surface equipment to handle these loads.

There are two general methods of designing a tapered sucker rod string:[1]

(1) Assign to each of the graduated sections of the string its maximum stress. In other words, a point in the string is determined at which the stress in the rod equals the arbitrarily selected maximum safe working stress; from this point up a larger size rod is used.

(2) Design the string so the unit stresses are equal in the top rod of each of the different-sized sections of the string.

The second method is usually safer, since it provides a greater safety margin as far as corrosion pitting is concerned. However, some people prefer the first method, in which the maximum allowable stress is placed in the top rod of the smallest (lowest) size. With this method, rod breaks would occur in the smallest rod and would prevent any potential buckling of rods that could occur from breaks farther up the hole.

The percentage of each size rod in a tapered string can be determined from Table 2.7 which is reproduced from API RP 11L.[3]

As mentioned previously, the sucker rod string is a complex system for transmitting the power from the surface equipment to the downhole pump. In the design of an "optimum" sucker rod string, the entire pumping system, including surface and subsurface equipment, must be considered.

To save unnecessary weight and to distribute the loading to better advantage on long rod strings, tapered rod strings are often used. Several sizes of rods may be used in the total string, depending upon the well depth and operating conditions. The problem is to determine the length of each rod size which should be used in the total string.

As a final step in designing a tapered rod string, the maximum anticipated stress must be checked to certify that it does not exceed the safe allowable working stress. The maximum stress at the top of the entire rod string will be the peak polished rod load (to be discussed in a following section) divided by the cross-sectional area of the top section of rods.

$$\text{Stress at the top} = \frac{W_{max}}{A_{top}} \tag{2.7}$$

The stress calculated from Equation 2.7 must never be greater than the allowable working stress (usually 30,000 psi). If it is, the rod string must be re-designed.

EXAMPLE PROBLEM # 5

A pump with a 2-in. plunger is to be set at 6050 ft using a three-way taper consisting of ¾-in., ⅞-in., and 1-in. rods. Sucker rods are available in 25 ft lengths. Determine the length of each section of the tapered rod string.

Solution
From Table 2.7 (rod no. 86);

R_1 = 32.8% of 1-in. rods
R_2 = 33.2% of ⅞-in. rods
R_3 = 33.9% of ¾-in. rods

Then

$L_1 = 6050\ (0.328) = 1984.4$ ft
$L_2 = 6050\ (0.332) = 2008.6$ ft
$L_3 = 6050\ (0.339) = 2051$ ft

Thus, in 25-ft increments,

$L_1 = 2000$ ft
$L_2 = 2000$ ft
$L_3 = 2050$ ft

Note: These values may have to be adjusted. Check maximum anticipated stress against allowable working stress for the particular string of rods used. This will involve the peak polished rod load, to be discussed later.

CLASS PROBLEM

A pump with a 1¼-in. plunger is set at 5050 ft using a three-way taper (API Rod No. 75) rod string. Fluid level is 4975 ft and fluid specific gravity is 0.976. Pumping speed is 21 spm and stroke length is 54 in. The tubing size (anchored) is 2.375 in. Obtain a pump displacement of 206 bf/d. If the plunger stroke is 94% of surface stroke and pump volumetric efficiency is 80%, determine the following:

(1) Percentage of each size rod in the string
(2) Length of each size rod, if rods are available in 25-ft lengths

Table 2.7 (After Ref. 3)
ROD AND PUMP DATA

1	2	3	4	5	6	7	8	9	10	11
Rod* no.	Plunger diam., inches	Rod weight lb per ft	Elastic constant, in. per lb ft E_r	Frequency Factor, F_c	Rod string, % of each size					
					1⅛	1	⅞	¾	⅝	½
44	All	0.726	1.990×10^{-6}	1.000	—	—	—	—	—	100.0
54	1.06	0.908	1.668×10^{-6}	1.138	—	—	—	—	44.6	55.4
54	1.25	0.929	1.633×10^{-6}	1.140	—	—	—	—	49.5	50.5
54	1.50	0.957	1.584×10^{-6}	1.137	—	—	—	—	56.4	43.6
54	1.75	0.990	1.525×10^{-6}	1.122	—	—	—	—	64.6	35.4
54	2.00	1.027	1.460×10^{-6}	1.095	—	—	—	—	73.7	26.3
54	2.25	1.067	1.391×10^{-6}	1.061	—	—	—	—	83.4	16.6
54	2.50	1.108	1.318×10^{-6}	1.023	—	—	—	—	93.5	6.5
55	All	1.135	1.270×10^{-6}	1.000	—	—	—	—	100.0	—
64	1.06	1.164	1.382×10^{-6}	1.229	—	—	—	33.3	33.1	33.5
64	1.25	1.211	1.319×10^{-6}	1.215	—	—	—	37.2	35.9	26.9
64	1.50	1.275	1.232×10^{-6}	1.184	—	—	—	42.3	40.4	17.3
64	1.75	1.341	1.141×10^{-6}	1.145	—	—	—	47.4	45.2	7.4
65	1.06	1.307	1.138×10^{-6}	1.098	—	—	—	34.4	65.6	—
65	1.25	1.321	1.127×10^{-6}	1.104	—	—	—	37.3	62.7	—
65	1.50	1.343	1.110×10^{-6}	1.110	—	—	—	41.8	58.2	—
65	1.75	1.369	1.090×10^{-6}	1.114	—	—	—	46.9	53.1	—
65	2.00	1.394	1.070×10^{-6}	1.114	—	—	—	52.0	48.0	—
65	2.25	1.426	1.045×10^{-6}	1.110	—	—	—	58.4	41.6	—
65	2.50	1.460	1.018×10^{-6}	1.099	—	—	—	65.2	34.8	—
65	2.75	1.497	0.990×10^{-6}	1.082	—	—	—	72.5	27.5	—
65	3.25	1.574	0.930×10^{-6}	1.037	—	—	—	88.1	11.9	—
66	All	1.634	0.883×10^{-6}	1.000	—	—	—	100.0	—	—
75	1.06	1.566	0.997×10^{-6}	1.191	—	—	27.0	27.4	45.6	—
75	1.25	1.604	0.973×10^{-6}	1.193	—	—	29.4	29.8	40.8	—
75	1.50	1.664	0.935×10^{-6}	1.189	—	—	33.3	33.3	33.3	—
75	1.75	1.732	0.892×10^{-6}	1.174	—	—	37.8	37.0	25.1	—
75	2.00	1.803	0.847×10^{-6}	1.151	—	—	42.4	41.3	16.3	—
75	2.25	1.875	0.801×10^{-6}	1.121	—	—	46.9	45.8	7.2	—
76	1.06	1.802	0.816×10^{-6}	1.072	—	—	28.5	71.5	—	—
76	1.25	1.814	0.812×10^{-6}	1.077	—	—	30.6	69.4	—	—
76	1.50	1.833	0.804×10^{-6}	1.082	—	—	33.8	66.2	—	—
76	1.75	1.855	0.795×10^{-6}	1.088	—	—	37.5	62.5	—	—
76	2.00	1.880	0.785×10^{-6}	1.093	—	—	41.7	58.3	—	—
76	2.25	1.908	0.774×10^{-6}	1.096	—	—	46.5	53.5	—	—
76	2.50	1.934	0.764×10^{-6}	1.097	—	—	50.8	49.2	—	—
76	2.75	1.967	0.751×10^{-6}	1.094	—	—	56.5	43.5	—	—
76	3.75	2.039	0.722×10^{-6}	1.078	—	—	68.7	31.3	—	—
76	3.75	2.119	0.690×10^{-6}	1.047	—	—	82.3	17.7	—	—
77	All	2.224	0.649×10^{-6}	1.000	—	—	100.0	—	—	—
85	1.06	1.883	0.873×10^{-6}	1.261	—	22.2	22.4	22.4	33.0	—
85	1.25	1.943	0.841×10^{-6}	1.253	—	23.9	24.2	24.3	27.6	—
85	1.50	2.039	0.791×10^{-6}	1.232	—	26.7	27.4	26.8	19.2	—
85	1.75	2.138	0.738×10^{-6}	1.201	—	29.6	30.4	29.5	10.5	—
86	1.06	2.058	0.742×10^{-6}	1.151	—	22.6	23.0	54.3	—	—
86	1.25	2.087	0.732×10^{-6}	1.156	—	24.3	24.5	51.2	—	—
86	1.50	2.133	0.717×10^{-6}	1.162	—	26.8	27.0	46.3	—	—
86	1.75	2.185	0.699×10^{-6}	1.164	—	29.4	30.0	40.6	—	—
86	2.00	2.247	0.679×10^{-6}	1.161	—	32.8	33.2	33.9	—	—
86	2.25	2.315	0.656×10^{-6}	1.153	—	36.9	36.0	27.1	—	—
86	2.50	2.385	0.633×10^{-6}	1.138	—	40.6	39.7	19.7	—	—
86	2.75	2.455	0.610×10^{-6}	1.119	—	44.5	43.3	12.2	—	—

Table 2.7 (After Ref. 3) (Cont.)
ROD AND PUMP DATA

1	2	3	4	5	6	7	8	9	10	11
Rod* no.	Plunger diam., inches	Rod weight lb per ft	Elastic constant, in. per lb ft E_r	Frequency Factor, F_c	Rod string, % of each size					
					1⅛	1	⅞	¾	⅝	½
87	1.06	2.390	0.612×10^{-6}	1.055	—	24.3	75.7	—	—	—
87	1.25	2.399	0.610×10^{-6}	1.058	—	25.7	74.3	—	—	—
87	1.50	2.413	0.607×10^{-6}	1.062	—	27.7	72.3	—	—	—
87	1.75	2.430	0.603×10^{-6}	1.066	—	30.3	69.7	—	—	—
87	2.00	2.450	0.598×10^{-6}	1.071	—	33.2	66.8	—	—	—
87	2.25	2.472	0.594×10^{-6}	1.075	—	36.4	63.6	—	—	—
87	2.50	2.496	0.588×10^{-6}	1.079	—	39.9	60.1	—	—	—
87	2.75	2.523	0.582×10^{-6}	1.082	—	43.9	56.1	—	—	—
87	3.25	2.575	0.570×10^{-6}	1.084	—	51.6	48.4	—	—	—
87	3.75	2.641	0.556×10^{-6}	1.078	—	61.2	38.8	—	—	—
87	4.75	2.793	0.522×10^{-6}	1.038	—	83.6	16.4	—	—	—
88	All	2.904	0.497×10^{-6}	1.000	—	100.0	—	—	—	—
96	1.06	2.382	0.670×10^{-6}	1.222	19.1	19.2	19.5	42.3	—	—
96	1.25	2.435	0.655×10^{-6}	1.224	20.5	20.5	20.7	38.3	—	—
96	1.50	2.511	0.633×10^{-6}	1.223	22.4	22.5	22.8	32.3	—	—
96	1.75	2.607	0.606×10^{-6}	1.213	24.8	25.1	25.1	25.1	—	—
96	2.00	2.703	0.578×10^{-6}	1.196	27.1	27.9	27.4	17.6	—	—
96	2.25	2.806	0.549×10^{-6}	1.172	29.6	30.7	29.8	9.8	—	—
97	1.06	2.645	0.568×10^{-6}	1.120	19.6	20.0	60.3	—	—	—
97	1.25	2.670	0.563×10^{-6}	1.124	20.8	21.2	58.0	—	—	—
97	1.50	2.707	0.556×10^{-6}	1.131	22.5	23.0	54.5	—	—	—
97	1.75	2.751	0.548×10^{-6}	1.137	24.5	25.0	50.4	—	—	—
97	2.00	2.801	0.538×10^{-6}	1.141	26.8	27.4	45.7	—	—	—
97	2.25	2.856	0.528×10^{-6}	1.143	29.4	30.2	40.4	—	—	—
97	2.50	2.921	0.515×10^{-6}	1.141	32.5	33.1	34.4	—	—	—
97	2.75	2.989	0.503×10^{-6}	1.135	36.1	35.3	28.6	—	—	—
97	3.25	3.132	0.475×10^{-6}	1.111	42.9	41.9	15.2	—	—	—
98	1.06	3.068	0.475×10^{-6}	1.043	21.2	78.8	—	—	—	—
98	1.25	3.076	0.474×10^{-6}	1.045	22.2	77.8	—	—	—	—
98	1.50	3.089	0.472×10^{-6}	1.048	23.8	76.2	—	—	—	—
98	1.75	3.103	0.470×10^{-6}	1.051	25.7	74.3	—	—	—	—
98	2.00	3.118	0.468×10^{-6}	1.055	27.7	72.3	—	—	—	—
98	2.25	3.137	0.465×10^{-6}	1.058	30.1	69.9	—	—	—	—
98	2.50	3.157	0.463×10^{-6}	1.062	32.7	67.3	—	—	—	—
98	2.75	3.180	0.460×10^{-6}	1.066	35.6	64.4	—	—	—	—
98	3.25	3.231	0.453×10^{-6}	1.071	42.2	57.8	—	—	—	—
98	3.75	3.289	0.445×10^{-6}	1.074	49.7	50.3	—	—	—	—
98	4.75	3.412	0.428×10^{-6}	1.064	65.7	34.3	—	—	—	—
99	All	3.676	0.393×10^{-6}	1.000	100.0	—	—	—	—	—
					1¼	1⅛	1	⅞	¾	⅝
107	1.06	2.977	0.524×10^{-6}	1.184	16.9	16.8	17.1	49.1	—	—
107	1.25	3.019	0.517×10^{-6}	1.189	17.9	17.8	18.0	46.3	—	—
107	1.50	3.085	0.506×10^{-6}	1.195	19.4	19.2	19.5	41.9	—	—
107	1.75	3.158	0.494×10^{-6}	1.197	21.0	21.0	21.2	36.9	—	—
107	2.00	3.238	0.480×10^{-6}	1.195	22.7	22.8	23.1	31.4	—	—
107	2.25	3.336	0.464×10^{-6}	1.187	25.0	25.0	25.0	25.0	—	—
107	2.50	3.435	0.447×10^{-6}	1.174	26.9	27.7	27.1	18.2	—	—
107	2.75	3.537	0.430×10^{-6}	1.156	29.1	30.2	29.3	11.3	—	—
108	1.06	3.325	0.447×10^{-6}	1.097	17.3	17.8	64.9	—	—	—
108	1.25	3.345	0.445×10^{-6}	1.101	18.1	18.6	63.2	—	—	—
108	1.50	3.376	0.441×10^{-6}	1.106	19.4	19.9	60.7	—	—	—
108	1.75	3.411	0.437×10^{-6}	1.111	20.9	21.4	57.7	—	—	—
108	2.00	3.452	0.432×10^{-6}	1.117	22.6	23.0	54.3	—	—	—
108	2.25	3.498	0.427×10^{-6}	1.121	24.5	25.0	50.5	—	—	—
108	2.50	3.548	0.421×10^{-6}	1.124	26.5	27.2	46.3	—	—	—
108	2.75	3.603	0.415×10^{-6}	1.126	28.7	29.6	41.6	—	—	—
108	3.25	3.731	0.400×10^{-6}	1.123	34.6	33.9	31.6	—	—	—
108	3.75	3.873	0.383×10^{-6}	1.108	40.6	39.5	19.9	—	—	—

Table 2.7 (After Ref. 3) (Cont.)
ROD AND PUMP DATA

1	2	3	4	5	6	7	8	9	10	11
Rod* no.	Plunger diam., inches	Rod weight lb per ft	Elastic constant, in. per lb ft E_r	Frequency Factor, F_c	Rod string, % of each size					
					1¼	1⅛	1	⅞	¾	⅝
109	1.06	3.839	0.378×10^{-6}	1.035	18.9	81.1	—	—	—	—
109	1.25	3.845	0.378×10^{-6}	1.036	19.6	80.4	—	—	—	—
109	1.50	3.855	0.377×10^{-6}	1.038	20.7	79.3	—	—	—	—
109	1.75	3.867	0.376×10^{-6}	1.040	22.1	77.9	—	—	—	—
109	2.00	3.880	0.375×10^{-6}	1.043	23.7	76.3	—	—	—	—
109	2.25	3.896	0.374×10^{-6}	1.046	25.4	74.6	—	—	—	—
109	2.50	3.911	0.372×10^{-6}	1.048	27.2	72.8	—	—	—	—
109	2.75	3.930	0.371×10^{-6}	1.051	29.4	70.6	—	—	—	—
109	3.25	3.971	0.367×10^{-6}	1.057	34.2	65.8	—	—	—	—
109	3.75	4.020	0.363×10^{-6}	1.063	39.9	60.1	—	—	—	—
109	4.75	4.120	0.354×10^{-6}	1.066	51.5	48.5	—	—	—	—

*Rod No. shown in first column refers to the largest and smallest rod size in eighths of an inch. For example, Rod No. 76 is a two-way taper of 7/8 and 6/8 rods. Rod No. 85 is a four-way taper of 8/8, 7/8, 6/8, and 5/8 rods. Rod No. 109 is a two-way taper of 1¼ and 1⅛ rods. Rod No. 77 is a straight string of 7/8 rods, etc.

(3) Effective plunger stroke, inches
(4) If this pumping combination will produce the desired pump displacement
(5) Surface production rate

CLASS PROBLEM

Repeat the previous class problem for a 1½-in. plunger and pumping speed of 14.6 spm. Plunger stroke is 75% of surface stroke.

CLASS PROBLEM

Repeat the previous class problem for a 1½-in. plunger and pumping speed of 19 spm. Plunger stroke is 81% of surface stroke.

Which of these three situations would you recommend?

2.42 Modified Goodman diagram

Equation 2.7 gives the actual maximum stress in the top rod, based upon the peak polished rod load and cross-sectional area of the top rod. This stress must never exceed the maximum *allowable* stress for the particular grade of rod being used, nor should the allowable *stress range* be exceeded. The API RP 11BR (5th edition, March 1969) discusses the modified Goodman diagram and its construction. Figure 2.14 illustrates its components. Figures 2.15 and 2.16 are diagrams for API Grade C rods (90,000 psi minimum tensile strength) and Grade D rods (115,000 psi minimum tensile strength), respectively.[22]

For API Grade C rods, the maximum allowable stress is given by the relationship:

$$S_A = \left(\frac{T}{4} + M\,S_{min}\right) \times S.F.$$
$$= \left(\frac{90{,}000}{4} + 0.5625\,S_{min}\right) \times S.F.$$

or,

$$S_A = (22{,}500 + 0.5625\,S_{min}) \times S.F. \quad (2.8)$$

For API Grade D rods, the relationship is:

$$S_A = \left(\frac{T}{4} + M\,S_{min}\right) \times S.F.$$
$$= \left(\frac{115{,}000}{4} + 0.5625\,S_{min}\right) \times S.F.$$

or,

$$S_A = (28{,}750 + 0.5625\,S_{min}) \times S.F. \quad (2.9)$$

where:

S_A = maximum allowable stress, psi
S_{min} = minimum rod stress (either calculated or measured), psi
S.F. = service factor (see table 2.8)

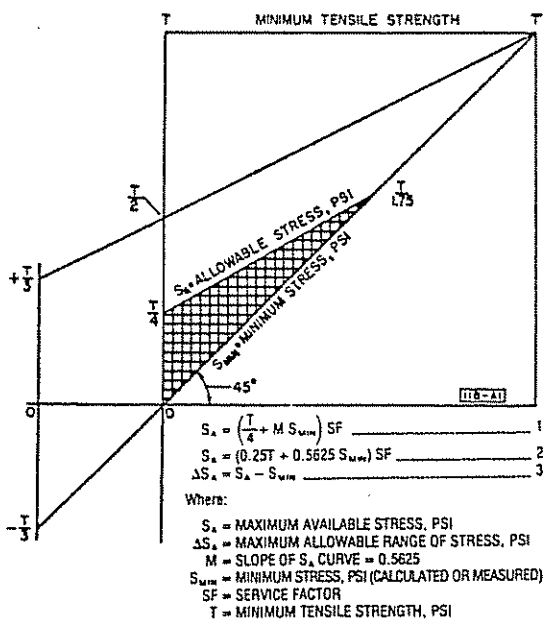
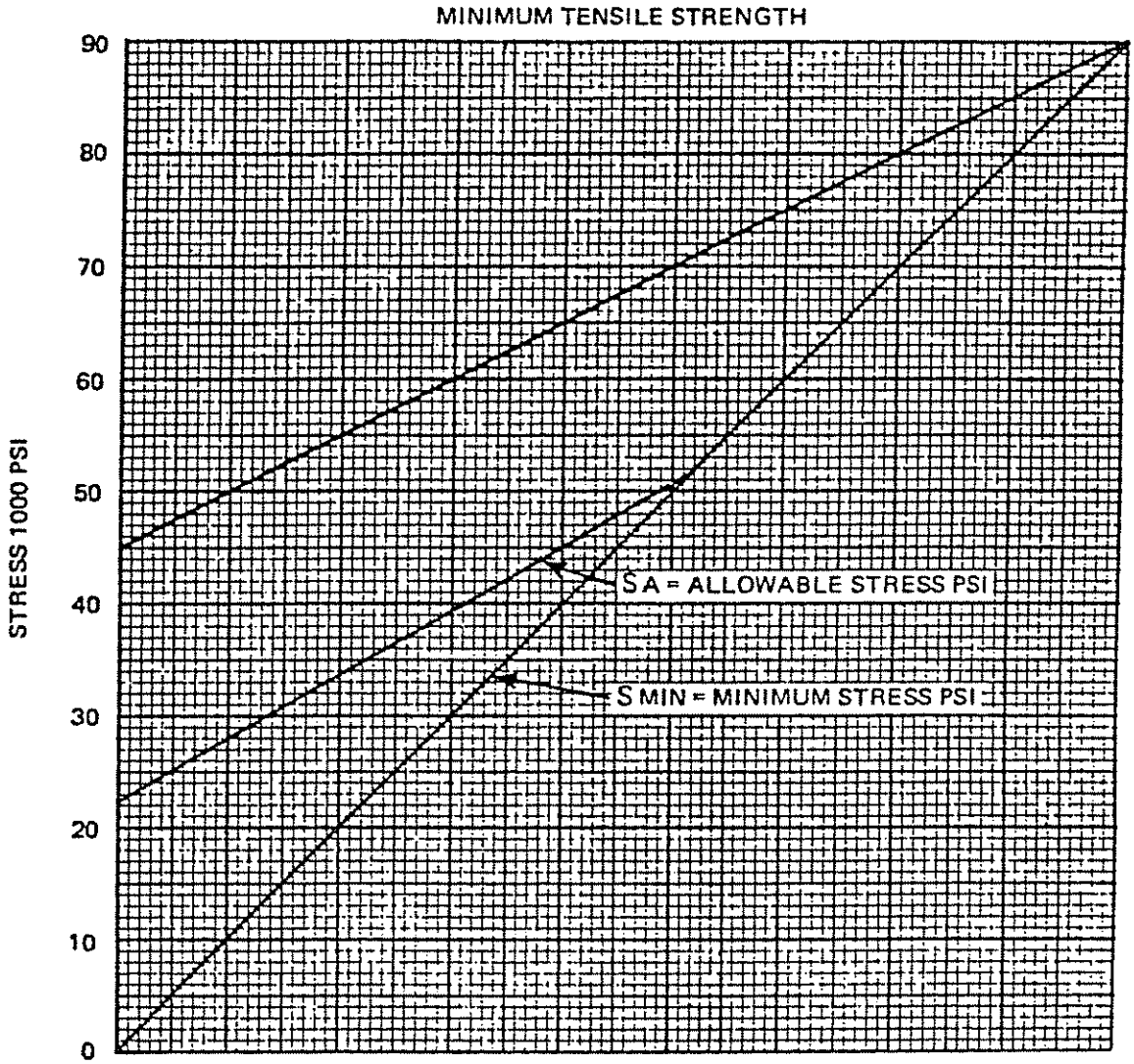


Fig. 2.14 Modified Goodman Diagram (from API RP11BR)



$S_A = (22{,}500 + 0.5625\ S_{MIN})\ SF$

WHERE: S_A = MAXIMUM ALLOWABLE STRESS, PSI

S_{MIN} = MINIMUM STRESS, PSI (CALCULATED OR MEASURED) = $\frac{\text{MPRL}}{\text{ROD AREA}}$

S_F = SERVICE FACTOR

MODIFIED GOODMAN DIAGRAM FOR ALLOWABLE STRESS AND RANGE OF STRESS FOR API GRADE C SUCKER RODS. SERVICE FACTOR = 1.0

Fig. 2.15 Modified Goodman Diagram, Grade C Sucker Rods (after Ref. 22)

TABLE 2.8
SERVICE FACTORS

Service	API C	API D
Non-corrosive	1.00	1.00
Salt water	0.65	0.90
Hydrogen sulfide	0.50	0.70

Grade D rods are susceptible to hydrogen embrittlement and normally should not be used in H_2S service.

The maximum allowable range of stress must also be determined for any rod string design according to the relationship:

$$\Delta S_A = S_A - S_{min} \tag{2.10}$$

where:

ΔS_A = maximum allowable range of stress, psi

S_{min} = calculated or measured MPRL divided by cross-sectional area of top rod, psi.

2.5 PUMPING MOTION

Beam and sucker rod pumping motion can be understood by visualizing two basic theoretical motions: (1) simple harmonic motion and (2) crank and pitman motion. Pumping motion is important because it controls polished rod velocities and accelerations which, in turn, influence rod and structural loads, load range, and bottomhole pump displacement.

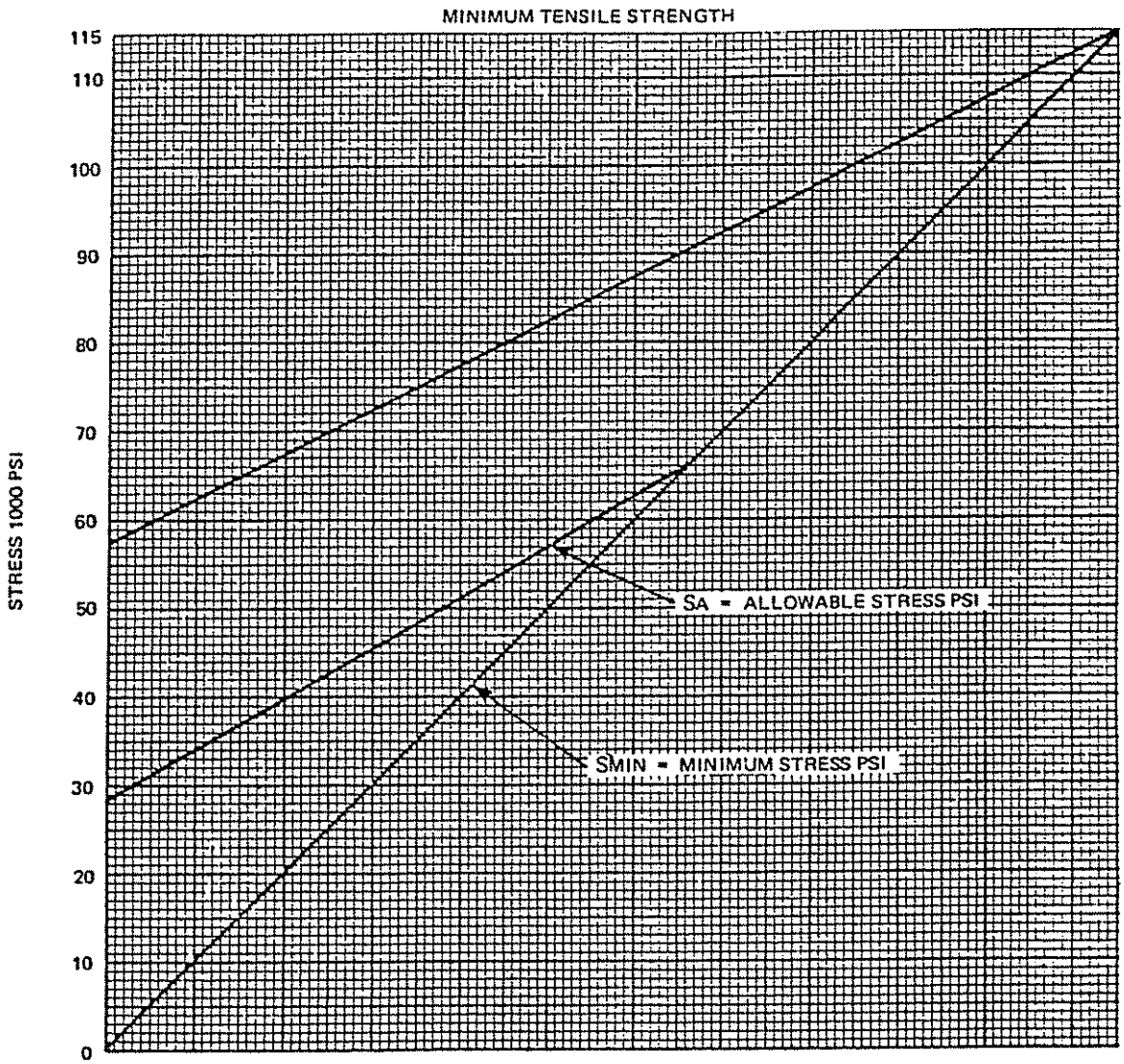


Fig. 2.16 Modified Goodman Diagram, Grade D Sucker Rods (after Ref. 22)

2.51 Simple harmonic motion (SHM)

Simple harmonic motion is a uniform, periodic motion, illustrated by the crank circle of Figure 2.17.

Suppose a perpendicular projection P' -P is drawn from the vertical axis D-B to the circumference, where it terminates at point P. If P travels around the crank circle with constant angular velocity (assumed as a reference), then point P' moves up and down the vertical axis D-B periodically with simple harmonic motion (SHM).

With this type of motion, P' starts downward off-top and upward off-bottom with the same value of maximum acceleration. The acceleration of P' as it reaches position 0 at the mid-point of the vertical axis is zero because it is changing from acceleration to deceleration.

The velocity of P' is zero at both the top and bottom of its travel (points B and D), but at mid-point 0 it is maximum at the same position where its acceleration is zero.

In this basic, theoretical system, if the top of a sucker rod string were attached to point P', it would be reciprocated up and down with simple harmonic motion.

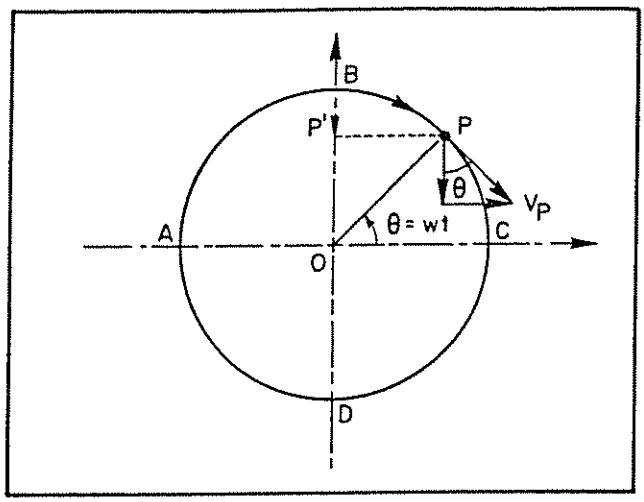


Fig. 2.17 Simple Harmonic Motion

2.52 Crank and pitman motion

In practice, however, the crank is connected by means of a crank pin (Fig. 2.18) to a rigid member called the pitman, which in turn drives the walking beam to effect the pumping motion. For convenience, the upper pitman connection (V), called the cross yoke or equalizer, is assumed to move vertically up and down a straight line; it actually travels along an arc whose center is the fulcrum, or Sampson post bearing, of the walking beam. The motion of the upper pitman connection, or cross yoke, is not simple harmonic motion, even though the crank is rotated with constant angular velocity.

As point P continues to move uniformly around its crank circle, the cross yoke (V) describes a reciprocating motion that may differ substantially from the simple harmonic motion of point P'. This is due to the angularity of the pitman-crank mechanism. The smaller the angle between crank and pitman when the crank is horizontal, the greater the divergence of the motion of the cross yoke (V) from that of the simple harmonic motion described by point P'. An infinitely long pitman would develop true simple harmonic motion at its cross yoke (V).

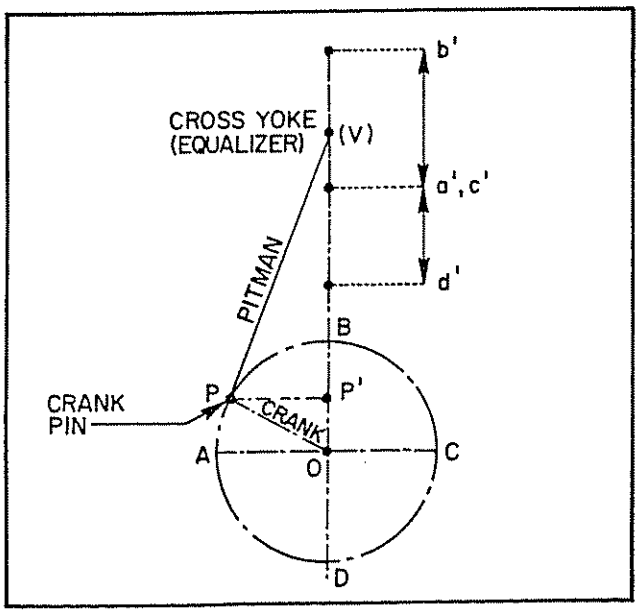


Fig. 2.18 Crank-Pitman Motion

In Figure 2.18, when the crankpin (P) moves around the circle from A to B to C, the cross yoke (V) moves vertically from a' to b' and back to c'. This distance is greater than the distance from c' to d' and back to a', which corresponds to the travel of the crankpin (P) from C to D and back to A. This is because the cross yoke (V) travel is a function of the vertical components of the crank (OP) and the pitman (PV). In the two upper quadrants of the crank circle, the vertical components of crank and pitman travel are additive; in the two lower quadrants they are subtractive.

This crank-pitman modification of simple harmonic motion can have a dramatic effect upon the velocity and acceleration characteristics of the cross yoke (V) driving the walking beam.

Because point P travels the circle with constant angular velocity, the time required to sweep through each of the four quadrants is the same. Since the distance the cross yoke (V) traverses from b' to a' is considerably longer than from a' to d', and since the over-bottom and over-top velocities are zero, the acceleration is substantially greater from b' to a' than from d' to a'.

This kinematic anomaly has important consequences. As the crankpin (P) sweeps across the two upper quadrants of the crank circle (A to B to C), the cross yoke (V) moves into the top of its stroke with a relatively high maximum deceleration and starts downward off-top with an equally high relative maximum acceleration. But as the crankpin (P) sweeps across the two lower quadrants (C to D to A), the cross yoke (V) decelerates into the bottom of the stroke with a relatively lower maximum value while likewise accelerating upward with a similar, reduced maximum acceleration. This differential cross yoke acceleration pattern plays a major role in determining rod, structural, and rod load range values in different pumping geometries.

2.53 The influence of pumping motion on the rod and structural loading of a beam-type unit*

In a beam and sucker rod pumping system, to lift the load the polished rod must exert an upward force greater than the "dead weight" of rods and fluid. This force is known as the *peak polished rod load* (PPRL) and consists of two components: (1) the static weight of rods and fluid, and (2) an additional force component. This additional force is often expressed as a percentage or fraction of the static weight of rods and fluid and is referred to as the "impulse" or "acceleration" factor—more precisely, the inertial component. For a given static weight of rods and fluid, the greater the inertial component, the faster the rod and fluid load is accelerated upward; the lower the inertial component, the slower the load is lifted.

The greatest force (peak polished rod load) is required near the beginning of the upstroke when the

*Momentarily assuming a frictionless, inelastic, rod, and fluid load.

maximum static load (rods plus fluid) is being elevated with maximum acceleration.

The *minimum polished rod load* (MPRL) results from the maximum downward inertial force component being subtracted from the static weight of the rods. Minimum load occurs near the top of the stroke just as the rods are starting down, because this is the point of maximum downward acceleration. The greater the downward acceleration, the greater the inertial force component and, when subtracted from the static weight of the rods, results in the minimum polished rod load.

Lifting maximum mass (weight of rods and fluid) with high acceleration results in a greater rod and structural load, producing larger rod stresses. Lifting the maximum load of rods and fluid off-bottom with reduced acceleration results in lower structural loads and rod stresses. In this basic, inelastic system the bottom reversal loads the rod string while the top reversal unloads it.

Table 2.9 shows the basic equations for determining maximum accelerations, inertial forces, peak and minimum polished rod loads in these two, frictionless, inelastic systems: (1) simple harmonic motion and (2) crank and pitman motion.

The relationships in Table 2.9 attempt to account for the fact that both rods and fluid must be accelerated. They are based upon the assumption that the rods and fluid can be represented as a concentrated mass and are accelerated simultaneously. In practice this does not always happen, but no other relationship presented heretofore has dealt satisfactorily with the problem of fluid acceleration. In fact, most traditional approaches have also neglected such factors as frictional and harmonious forces, damping, etc., in the derivation of equations for calculating polished rod loads. These loads must, of course, be predicted in the selection of equipment for a pumping installation and for designing and selecting a suitable sucker rod string.

The most commonly used method for calculating peak and minimum polished rod loads is that of Mills, which (incorrectly) assumes simple harmonic motion of the rod string and which neglects fluid acceleration.[12] Many industry "calculation sheets" actually use the Mills method, probably because it presupposes that in an elastic rod string, the rods move before the fluid does. Actually, the method does give fairly reasonable results, and many satisfactory pumping installations have been designed using this approach.

Mills' method is applicable only for simple harmonic motion and does not account for the effect of the crank-pitman ratio (c/p)—even for conventional pumping units—upon the acceleration factor. The crank-pitman ratio (c/p) can alter simple harmonic acceleration as much as 30% and perhaps even more. A summary of the (c/p) influence on polished rod acceleration is shown in Table 2.10. The Mills acceleration factor is given by:

$$\alpha = \frac{SN^2}{70{,}500} \tag{2.11}$$

where:

S = length of stroke, inches
N = pumping speed, strokes per minute (spm)

The term $(1 + \alpha)$ is sometimes referred to as the "impulse factor." Mills applies this factor only to the static weight of the rods and not to the fluid, as was done with the equations in Table 2.9. Note that the factor applies only for simple harmonic motion and must be modified by the crank-pitman ratio for either Class I or Class III lever systems.

For comparison with the relationships given in Table 2.9, Mills' equations for peak polished rod load (PPRL) and minimum polished rod load (MPRL) are given by the following:

1- For conventional units:

$$\begin{aligned} PPRL &= W_f + W_r + W_r\alpha \\ &= W_f + W_r(1 + \alpha) \end{aligned} \tag{2.12a}$$

$$MPRL = W_r(1 - \alpha) - \left[\frac{62.4GW_r}{490}\right] \tag{2.13}$$

or,

$$MPRL = W_r(1 - \alpha - 0.127G) \tag{2.14a}$$

where:

G = fluid specific gravity
W_r = static weight of rods in air, lb (see Table 2.6 or 2.7)
490 = density of steel rods, lb/ft³
W_f = weight of fluid, lb

2- For air balance units:

$$PPRL = W_f + W_r(1 + 0.7\alpha) \tag{2.12b}$$

The 0.7 factor simply considers that the air balance unit makes its bottom polished rod reversal with only 70% of the acceleration of a comparable conventional unit.[28]

$$MPRL = W_r(1 - 1.3\alpha - 0.127G) \tag{2.14b}$$

The air balance unit makes its bottom reversal approximately 30% slower than the comparable conventional unit, but its top reversal is about 30% faster; hence, 30% must be added to the acceleration factor to obtain a more accurate minimum load.[28]

3- For Mark II units:

$$PPRL = W_f + W_r(1 + 0.6\alpha) \tag{2.12c}$$

The 0.6 factor considers that the Mark II makes its bottom polished rod reversal with but 60% of the acceleration of a comparable conventional unit.[28]

$$MPRL = W_r(1 - 1.4\alpha - 0.127G) \tag{2.14c}$$

The Mark II makes its bottom reversal 40% slower than the comparable conventional unit, but its top reversal is 40% faster; hence, 40% must be added to the acceleration factor to obtain an accurate minimum load.[28]

The fluid load, W_f, is the weight of the fluid column supported by the *net* plunger area:

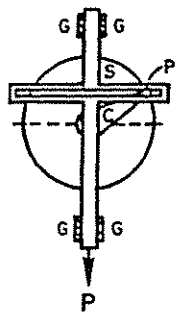
$$W_f = 0.433\ GL(A_p - A_r) \tag{2.15}$$

where:

L = length of rod string, ft
A_p = gross plunger area, sq. in.
A_r = rod average cross-sectional area, sq. in.

TABLE 2.9

FORCES AND ACCELERATIONS FOR THREE TYPES OF PUMPING MOTION
(INELASTIC, CONCENTRATED MASS SYSTEMS)

	SIMPLE HARMONIC MOTION	CRANK AND PITMAN MOTION (CLASS I LEVER)	CRANK AND PITMAN MOTION (CLASS III LEVER)
		 Crank/Pitman Ratios 64" -- .32 74" -- .33 86" -- .33 100" -- .33 120" -- .27 144" -- .26 168" -- .22	  Crank/Pitman Ratios 64" -- .22 64" -- .12 74" -- .26 74" -- .13 86" -- .23 86" -- .15 100" -- .22 100" -- .15 120" -- .26 120" -- .15 144" -- .29 144" -- .16 168" -- .33 168" -- .15 192" -- .32 192" -- .19 216" -- .34 216" -- .17 240" -- .18
Maximum Off-Bottom Acceleration (Ft/Sec^2)	$A_{max} = \frac{SN^2}{2189}$	$A_{max_1} = \frac{SN^2 (1 + \frac{C}{P})}{2189}$	$A_{max_1} = \frac{SN^2 (1 - \frac{C}{P})}{2189}$
Maximum Off-Top Acceleration (Ft/Sec^2)	$A_{max} = \frac{SN^2}{2189}$	$A_{max_2} = \frac{SN^2 (1 - \frac{C}{P})}{2189}$	$A_{max_2} = \frac{SN^2 (1 + \frac{C}{P})}{2189}$
Inertial Polished Rod Force Component (Off-Bottom) (Lbs.)	$I_F = \frac{W_1 \times SN^2}{70,500}$	$I_{F_1} = \frac{W_1 \times SN^2 (1 + \frac{C}{P})}{70,500}$	$I_{F_1} = \frac{W_1 \times SN^2 (1 - \frac{C}{P})}{70,500}$
Inertial Polished Rod Force Component (Off-Top)	$I_F = \frac{W_2 \times SN^2}{70,500}$	$I_{F_2} = \frac{W_2 \times SN^2 \times (1 - \frac{C}{P})}{70,500}$	$I_{F_2} = \frac{W_2 \times SN^2 \times (1 + \frac{C}{P})}{70,500}$
Peak Polished Rod Load	$PPRL = W_1 \left(1 + \frac{SN^2}{70,500}\right)$	$PPRL = \frac{W_1 + W_1 \times SN^2 \times (1 + \frac{C}{P})}{70,500}$	$PPRL = \frac{W_1 + W_1 \times SN^2 \times (1 - \frac{C}{P})}{70,500}$
Minimum Polished Rod Load	$MPRL = W_2 \left(1 - \frac{SN^2}{70,500}\right)$	$MPRL = \frac{W_2 - W_2 \times SN^2 \times (1 - \frac{C}{P})}{70,500}$	$MPRL = \frac{W_2 - W_2 \times SN^2 \times (1 + \frac{C}{P})}{70,500}$

WHERE:

S = Stroke Length (In.)
N = Strokes Per Minute (SPM)
$\frac{C}{P}$ = Crank-Pitman Ratio
W_1 = Weight of Rods and Fluid = $W_r + W_f$
W_2 = Weight of Buoyant Rods = $W_r (1 - 0.127G)$

TABLE 2.10
(From Lufkin by permission)
MILLS ACCELERATION FACTOR*

Strokes Per Minute	Length of polished rod stroke—in.																
	16	24	30	36	42	48	54	64	74	86	100	120	144	168	192	216	240
1	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.001	1.001	1.001	1.001	1.002	1.002	1.002	1.003	1.003	1.003
2	1.001	1.001	1.002	1.002	1.002	1.003	1.003	1.004	1.004	1.005	1.006	1.007	1.008	1.010	1.011	1.012	1.014
3	1.002	1.003	1.004	1.005	1.005	1.006	1.007	1.008	1.009	1.011	1.013	1.015	1.018	1.021	1.025	1.028	1.031
4	1.004	1.005	1.007	1.008	1.010	1.011	1.012	1.015	1.017	1.020	1.023	1.027	1.033	1.038	1.044	1.049	1.054
5	1.006	1.009	1.011	1.013	1.015	1.017	1.019	1.023	1.026	1.031	1.035	1.043	1.051	1.060	1.068	1.077	1.085
6	1.008	1.012	1.015	1.018	1.021	1.025	1.028	1.033	1.038	1.044	1.051	1.061	1.074	1.086	1.098	1.110	1.128
7	1.011	1.017	1.021	1.025	1.029	1.033	1.038	1.044	1.051	1.060	1.070	1.083	1.100	1.117	1.133	1.150	1.167
8	1.015	1.022	1.027	1.033	1.038	1.044	1.049	1.058	1.067	1.078	1.091	1.109	1.131	1.153	1.174	1.196	1.218
9	1.018	1.028	1.035	1.041	1.048	1.055	1.062	1.074	1.085	1.099	1.115	1.138	1.165	1.193	1.221	1.248	1.276
10	1.023	1.034	1.043	1.051	1.060	1.068	1.077	1.091	1.105	1.122	1.142	1.170	1.204	1.238	1.272	1.306	1.340
11	1.028	1.041	1.052	1.062	1.072	1.082	1.903	1.110	1.127	1.148	1.172	1.206	1.247	1.288	1.329	1.371	1.412
12	1.033	1.049	1.061	1.074	1.086	1.098	1.110	1.131	1.151	1.176	1.204	1.245	1.294	1.343	1.392	1.441	
13	1.038	1.058	1.072	1.086	1.101	1.115	1.129	1.153	1.177	1.206	1.240	1.288	1.345	1.403	1.460		
14	1.045	1.067	1.083	1.100	1.117	1.134	1.151	1.179	1.207	1.239	1.278	1.335	1.402	1.469	1.533		
15	1.051	1.077	1.096	1.115	1.134	1.154	1.173	1.205	1.237	1.275	1.319	1.385	1.462				
16	1.058	1.087	1.109	1:131	1.153	1.174	1.196	1.233	1.269	1.312	1.363	1.436	1.523				
17	1.066	1.098	1.123	1.148	1.172	1.197	1.221	1.262	1.303	1.353	1.410	1.492					
18	1.074	1.110	1.138	1.165	1.193	1.221	1.248	1.294	1.340	1.395	1.460	1.552					
19	1.082	1.123	1.154	1.184	1.215	1.246	1.277	1.328	1.379	1.440	1.512						
20	1.091	1.136	1.170	1.204	1.239	1.272	1.306	1.363	1.420	1.488	1.567						
21	1.100	1.150	1.188	1.225	1.263	1.300	1.338	1.400	1.463	1.538	1.625						
22	1.110	1.165	1.206	1.247	1.288	1.330	1.371	1.440	1.508	1.590							
23	1.120	1.180	1.225	1.270	1.315	1.360	1.405	1.480	1.555	1.645							
24	1.131	1.196	1.245	1.294	1.343	1.392	1.441	1.523	1.605	1.703							

*Mills acceleration factor $= 1 + \frac{SN^2}{70{,}500}$

Where S = length of stroke in inches
N = pumping speed in strokes per minute

The weight of the fluid could be expressed as:

$$W_f = 0.433G(LA_p - 0.294W_r) \qquad (2.16)$$

EXAMPLE PROBLEM #6

The lengths of each section of a three-way taper were determined in Example Problem #5. Check this string of rods to determine whether or not the maximum stress is within the allowable working stress (assumed to be 30,000 psi). Assume fluid specific gravity of 0.89, pumping speed of 18 spm, and stroke length of 54 in.

Solution:

From Table 2.6, $A_{wp} = 0.785$ sq in. (1-in. rods)
W_{r1}(1-in. rods) = 2.90 lb/ft
W_{r2}(⅞-in. rods) = 2.22 lb/ft
W_{r3}(¾-in. rods) = 1.63 lb/ft

$$W_r = W_{r1}L_1 + W_{r2}L_2 + W_{r3}L_3$$
$$= (2.90)(2000) + (2.22)(2000) + (1.63)(2050)$$
$$= 13{,}582 \text{ lb}$$

$$W_f = 0.433G(LA_p - 0.294\,W_r) \qquad (2.16)$$
$$= 0.433\,(0.89)[6050\,(3.142) - 0.294\,(13{,}582)]$$
$$= 5787 \text{ lb}$$

$$W_{max} = PPRL = W_f + W_r(1 + \alpha) \qquad (2.12)$$

$$\alpha = \frac{SN^2}{70{,}500} = \frac{54(18)^2}{70{,}500} = 0.248 \qquad (2.11)$$
Table 2.10

Thus,

$$W_{max} = 5787 + 13{,}582\,(1 + 0.248)$$
$$= 22{,}737 \text{ lb.}$$

$$\text{Stress} = \frac{W_{max}}{A_{top}} = \frac{22{,}737}{0.785} = 28{,}964 \text{ psi}$$

CLASS PROBLEMS

(1) Use the modified Goodman diagram relationships to determine the maximum allowable stress and stress range for both API Grade C and D rods for Example Problem #6. Assume a service factor of 1.0. Are the stress relationships satisfactory for this system?
(2) Repeat Problem 1 for salt water service.
(3) Repeat Problem 1 for hydrogen sulfide service.

2.6 THE EFFECT OF CRANK-TO-PITMAN RATIO ON CLASS 1 AND CLASS III GEOMETRIES

In practice, crank-to-pitman ratios (crank length divided by pitman length) run from about 0.10 to nearly 0.50 (Table 2.11). The two factors that significantly affect off-bottom and off-top acceleration are (1) the

TABLE 2.11
TYPICAL CRANK-TO-PITMAN RATIOS
(c/p) (APPROXIMATE)
(From Lufkin, by Permission)

Conventional unit	Stroke length	Mark II unit
(0.32)	64"	(0.22)
(0.33)	74"	(0.26)
(0.33)	86"	(0.23)
(0.33)	100"	(0.22)
(0.27)	120"	(0.26)
(0.26)	144"	(0.29)
(0.22)	168"	(0.33)
--	192"	(0.32)
--	216"	(0.34)

crank-pitman ratio and (2) the type pumping unit geometry—whether Class I [conventional (pull-down)] or Class III [Mark II or air balance (push-up)] (see Fig. 2.19).

In a conventional (Class I) lever system, the lower the crank-to-pitman ratio, the slower the bottom polished rod reversal and the faster the top reversal relative to simple harmonic motion. The converse is also true: the higher the crank-to-pitman ratio in a conventional unit, the higher the off-bottom acceleration, and the lower the off-top acceleration. As the c/p ratio becomes smaller, the conventional unit approaches simple harmonic motion.

On the other hand, a push-up system (Class III) behaves in the opposite fashion. The higher the c/p ratio, the slower the unit makes its bottom reversal and the faster it makes its top reversal relative to simple harmonic motion. The converse is also true on a Class III system: the lower the c/p ratio, the faster the off-bottom acceleration and the lower the off-top acceleration. As the crank-to-pitman ratio approaches zero on a Class III lever system, the off-bottom and off-top accelerations become the same and the motion approaches simple harmonic motion, like the Class I (conventional) system.

In the real world, sucker rod pumping does not behave in this convenient fashion, because neither the sucker rod string nor the fluid column act as a concentrated, inelastic, frictionless mass.

The elasticity of the sucker rod system with its harmonic stress waves running up and down the rod string modifies this simple concentrated mass system in a complex fashion; no easy alteration to the above simple relationships can predict accurate load and pump displacement values in field conditions.

In shallow to medium depth pumping, with reasonably stiff rod strings, the beam and sucker rod system may approach these basic concentrated mass relationships of theoretical mechanics, but in deeper wells with longer, more flexible rod strings and heavier fluid loads, the involved nature of the elastic rod string does not result in such simple load relationships. In such cases, the classic solution of the wave equation accounting for the actual surface geometry of the pumping unit must be applied in order to construct a mathematical model that will accurately simulate actual load and pump displacement values in field performance.

2.7 EFFECTIVE PLUNGER STROKE

A considerable difference can exist between the length of the stroke of the polished rod and the actual length of travel of the pump plunger. The point of interest is the distance that the plunger travels relative to the working barrel. This relative motion between the plunger and the working barrel results in the net or effective plunger stroke, which differs from the motion of the polished rod because of several factors:

(1) Rod and tubing stretching and contraction resulting from the imposition and release of loads during the pumping cycle
(2) Plunger overtravel because of the dynamic loads and the elastic nature of the rods
(3) Rod vibration effects
(4) Subsurface friction effects
(5) Gas in the pump barrel.

Most formulas commonly used for determining effective plunger stroke consider only the rod and tubing stretch and the plunger overtravel. The effect of rod and tubing stretch *decreases* the plunger stroke, and the effect of plunger overtravel *increases* the plunger stroke. A simple *approximation* for effective plunger stroke is given by:

$$S_p = S + e_p - (e_t + e_r) \qquad (2.17)$$

S_p = effective plunger stroke, inches
S = polished rod stroke, inches
e_p = plunger overtravel, inches
e_t = tubing stretch, inches
e_r = rod stretch, inches

2.71 Rod and tubing stretch—single rod size

Rod and tubing stretch, or elongation, is caused by the cyclic transfer of fluid load from the standing valve to the traveling valve. Rod stretch is the result of the rod weight and the imposition of the weight of the fluid column in the tubing onto the rod string when the traveling valve closes at the bottom of the stroke.

At the top of the stroke, the traveling valve opens and the weight of the fluid column is shifted back to the standing valve (thus back to the tubing). The elongation of the rod string and tubing can be determined from charts furnished by some of the sucker rod manufacturers, or the stretch can be calculated from formulas.

Rod stretch is given by: $$e_r = \frac{12W_fL}{A_rE} \qquad (2.18)$$

where:

e_r = rod stretch, in.
W_f = fluid load on the plunger, lbs.
L = length of sucker rod string, ft
A_r = average cross-sectional area of rods, sq. in.
E = modulus of elasticity for steel (approximately 30×10^6 psi)

The fluid load for Equation 2.18 can be determined from Equation 2.15 or Equation 2.16. The result is the equation for *rod stretch* presented by Marsh:[10]

$$e_r = \frac{5.20\ GL^2(A_p - A_r)}{A_rE} \qquad (2.19)$$

Equation 2.19 assumes that the pump is set at the casing fluid level, i.e., the well is "pumped off."

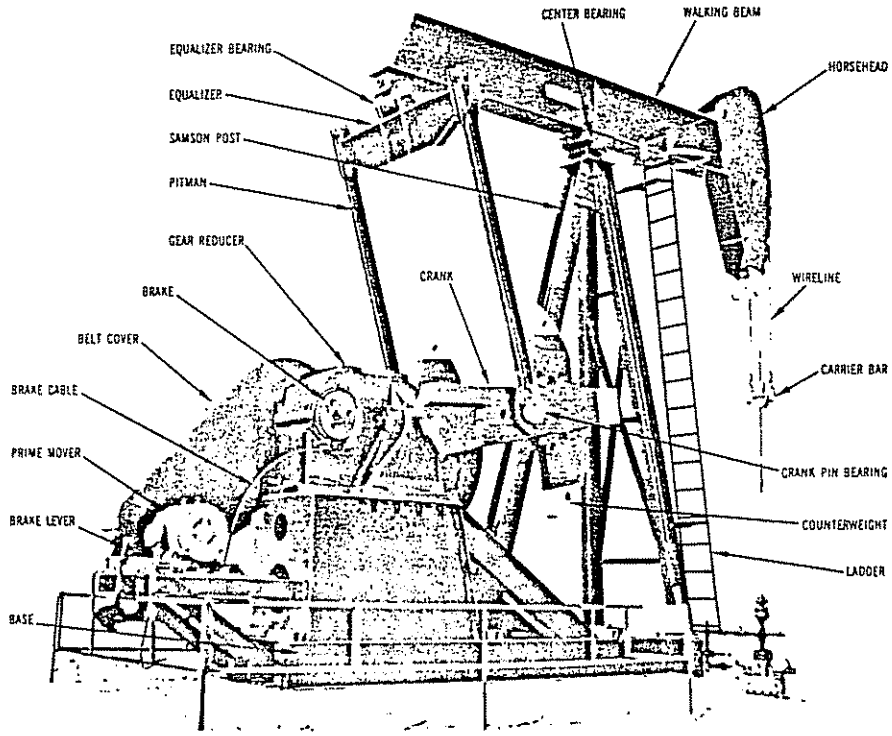
Another approach is to assume that the fluid weight on the plunger acts on the gross plunger area and to attempt to account for the upthrust on the plunger when the casing fluid level is higher than the pump setting depth. This is the approach taken by the API and can be expressed as follows:

$$W_f = \frac{62.4GDA_p}{144} = 0.433DGA_p \qquad (2.20)$$

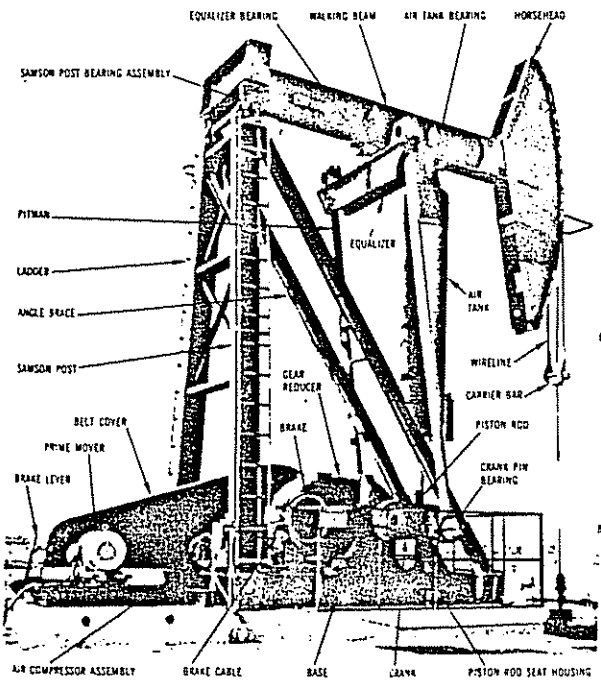
where:

D = depth of working fluid level in the casing annulus, (net lift), feet

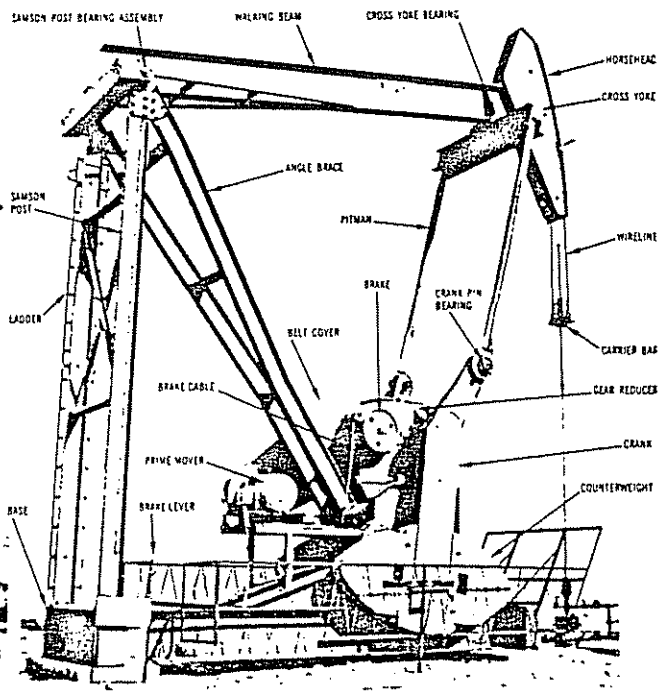
PUMPING UNITS NOMENCLATURE



Conventional Pumping Unit



Air Balanced Pumping Unit



Mark II Pumping Unit

Fig. 2.19 Pumping Unit (Courtesy Lufkin)

Fluid specific gravity, G, can be determined from Figure 2.20. The equation for rod stretch can then be written in the form:

$$e_r = \frac{12 \times 0.433 G D A_p L}{A_r E}$$

or,

$$e_r = \frac{5.20 G D A_p L}{A_r E} \quad (2.21)$$

This equation is general and can be used for the case when the pump setting depth is lower than the fluid level in the casing annulus.

Tubing stretch can be determined in a similar manner, if the cross-sectional area of the metal in the tubing, A_t, is substituted for A_r in the above equations. The equation for tubing stretch then becomes:

$$e_t = \frac{5.20 G D A_p L}{A_t E} \quad (2.22)$$

or according to Equation 2.19:

$$e_t = \frac{5.20 G (A_p - A_r) L^2}{A_t E} \quad (2.23)$$

In shallow wells, the tubing stretch may be small compared to the rod stretch and is frequently ignored. In particular, if the tubing is properly anchored, then the tubing stretch is zero ($e_t = 0$). Use tubing anchors to prevent unnecessary wear of the tubing and casing at points of contact with the casing.

2.72 Rod stretch—tapered strings

For the case of a tapered rod string, i.e., a string which contains several sections of different size rods, Equation 2.18 can be applied to each section of rods as follows:

$$e_1 = \frac{12 W_f L_1}{A_1 E}, \; e_2 = \frac{12 W_f L_2}{A_2 E}, \cdots e_n = \frac{12 W_f L_n}{A_n E}$$

where $L_1, L_2, \ldots, L_n$ refer to the length of each section which has rod cross-sectional area of $A_1, A_2, \ldots, A_n$, respectively. The total rod stretch of the tapered string is then written:

$$e_r = e_1 + e_2 + \cdots + e_n = \frac{12 W_f}{E}\left[\frac{L_1}{A_1} + \frac{L_2}{A_2} + \cdots + \frac{L_n}{A_n}\right] \quad (2.24)$$

Note that the tapered rod string must be selected prior to calculating the rod stretch.

2.73 Plunger overtravel

The equations given in the two previous sections are used to calculate the elongation of the rods and tubing due to the fluid load. One must also consider the elongation due to the rod load in a dynamic system. This additional elongation results from the dead weight of the rods plus the load due to acceleration of the rod string and is termed *plunger overtravel*, e_p.

Many formulas have been presented in literature, and each of them give varying results because of the complexity of the pumping system, particularly in the deeper wells.

2.731 Coberley's method

The first mathematical relationship for plunger overtravel was presented by C. J. Coberly[11] in a discussion of a paper by H. N. Marsh.[10] This method is sometimes referred to as the Marsh-Coberly Method, or more commonly as the Coberly Method. The method assumes simple harmonic motion, no reflected stresses, and no time lag in transmission of stresses through the rod string.

The Coberly Method is:

$$e_p = 1.93 \times 10^{-11} (LN)^2 S \quad (2.25)$$

where:

e_p = plunger overtravel, in.
S = length of polished rod stroke, in.
N = pumping speed, strokes per minute
L = length of rod string, ft

Equation 2.25 is valid for a single size rod string. See Reference 11 for a derivation of Equation 2.25.

To see the effect of acceleration more clearly, Equation 2.25 can be expressed as:

$$e_p = 1.36 \times 10^{-6} L^2 \left(\frac{SN^2}{70{,}500}\right) \quad (2.26)$$

where the last term is the acceleration factor presented earlier:

$$\alpha = \frac{SN^2}{70{,}500}.$$

Thus, we could write

$$e_p = 1.36 \times 10^{-6} L^2 \alpha \quad (2.27)$$

or

$$e_p = \frac{40.8 L^2 \alpha}{E} \quad (2.28)$$

Equation 2.28 is the formula presented by Craft, Holden, and Graves[5] which is simply Coberly's formula when $E = 30 \times 10^6$ psi.

In the development of the above equation, no allowance was made for the effect of unit geometry, and the possibility of a tapered string was ignored. When these factors are considered, the actual plunger travel could be quite different from that calculated by these formulas. However, because of these additional complexities no entirely satisfactory relationship has been obtained. Nevertheless, the following formula has been suggested for general use, including tapered strings:[2]

$$e_p = 1.55\left[\frac{L}{1000}\right]^2\left[\frac{SN^2}{70{,}500}\right] \quad (2.29)$$

or

$$e_p = \frac{46.5 L^2 \alpha}{E} \quad (2.30)$$

or

$$e_p = 2.2 \times 10^{-11} S(LN)^2 \quad (2.31)$$

Use equations 2.29, 2.30, or 2.31 for calculating plunger overtravel for tapered strings; some authorities even suggest their use for single rod strings to attempt to account for unit geometry. Exercise caution, however, because the factor of 2.2 in Equation 2.31 (or even the

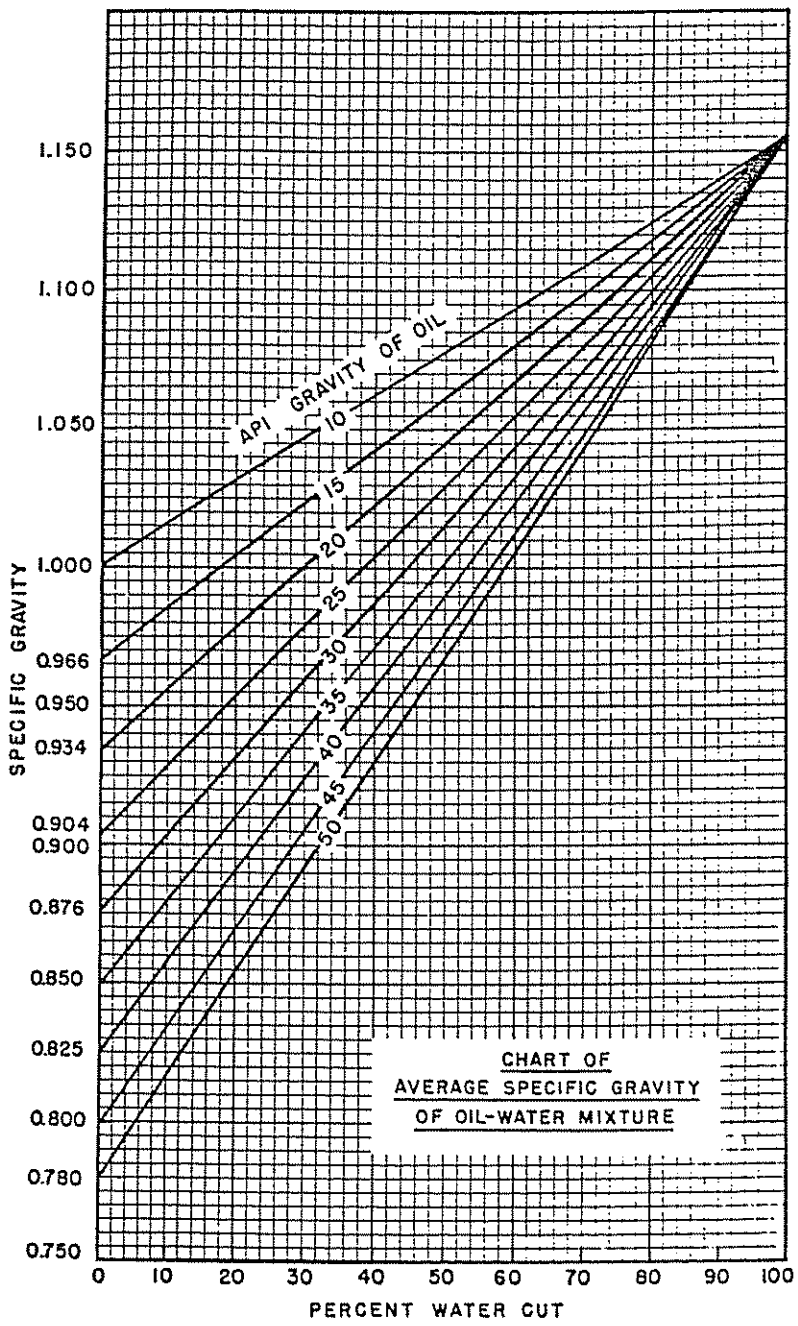


Fig. 2.20 FLUID SPECIFIC GRAVITY

factor 1.93 in Equation 2.25) can vary from well to well.

EXAMPLE PROBLEM #7

A pumping installation consists of a 2¼-in. pump set at 7080 ft in 2⅞-in. tubing (2.441-in. I.D., and 2.875-in. O.D.). Oil having a specific gravity of 0.81 is at a level of 5800 ft in the casing annulus. The unit utilizes a rod string consisting of ¾-in. and ⅞-in. rods and operates at 16.8 spm. Pump efficiency is 75% and 55 B/D are being produced.

Determine:
(a) Effective plunger stroke
(b) Tubing stretch
(c) Tapered rod stretch
(d) Polished rod stroke
(e) Overtravel

Solution:

a) $PD = 0.1484\ A_p S_p N$ (2.1)

$Q = PDE_v = 0.1484\ A_p S_p N E_v$

$$S_p = \frac{Q}{0.1484\ A_p N E_v}$$

From Table 2.1, $A_p = 3.976$ sq in.

$$S_p = \frac{55}{(0.1484)(3.976)(16.8)(0.75)}$$
$$= 7.4 \text{ in.}$$

b) $$e_t = \frac{5.20GDA_pL}{EA_t} \qquad (2.22)$$

From Table 2.4, $A_t = 1.812$ sq in.

$$e_t = \frac{(5.20)(0.81)(5800)(3.976)(7080)}{(30 \times 10^6)(1.812)}$$
$$= 12.65 \text{ in.}$$

We could also calculate tubing stretch as follows:

From Table 2.4, the elastic constant is $E_t = 0.221 \times 10^{-6}$ in./lb/ft.

Thus:

$$e_t = (0.221 \times 10^{-6})\, W_t L$$
$$= (0.221 \times 10^{-6})[0.433GDA_p]\,(7080)$$
$$= (0.221 \times 10^{-6})[0.433(0.81)(5800)(3.976)]\,(7080)$$
$$= (0.221 \times 10^{-6})(8088)(7080)$$
$$e_t = 12.65 \text{ in.}$$

c) $$e_r = \left(\frac{5.20GDA_p}{E}\right)\left(\frac{L_1}{A_1} + \frac{L_2}{A_2}\right) \qquad (2.24)$$

For illustration, assume that $L_1 = 3788$ ft and $L_2 = 3292$ ft.

$$e_r = \frac{(5.20)(0.81)(5800)(3.976)}{30 \times 10^6}\left(\frac{3788}{0.442} + \frac{3292}{0.601}\right)$$
$$= 45.48 \text{ in.}$$

We could also calculate the tapered rod stretch as follows:

From Table 2.7, the elastic constant for this string (Rod No. 76) is

$$E_r = 0.774 \times 10^{-6} \text{ in./lb/ft}$$

Thus,

$$e_r = (0.774 \times 10^{-6})W_t L$$
$$= (0.774 \times 10^{-6})(8088)(7080)$$
$$e_r = 44.32 \text{ in.}$$

d) $$S_p = S + e_p - e_t - e_r$$

or,

$$S = S_p - e_p + e_t + e_r$$
$$e_p = 1.55\left(\frac{L}{1000}\right)^2 \alpha = 1.55\left(\frac{L}{1000}\right)^2 \left(\frac{SN^2}{70{,}500}\right)$$
$$S = 7.4 + 12.65 + 45.48 - e_p$$
$$= 65.33 - e_p$$
$$S = 65.53 - 1.55\left(\frac{7080}{1000}\right)^2\left[\frac{S(16.8)^2}{70{,}500}\right] =$$
$$65.53 - 0.311\,S$$

$$S = \frac{65.53}{1.311} = 50.0 \text{ in.}$$

e) $$e_p = 1.55\left(\frac{L}{1000}\right)^2 \alpha$$
$$= 1.55\left(\frac{7080}{1000}\right)^2\left[\frac{(50.0)(16.8)^2}{70{,}500}\right]$$
$$= 15.6 \text{ in.}$$

Note that this example problem illustrates an impractical solution ($S = 50$ in.; $S_p = 7.4$) in. which can result when basic relationships are misapplied or when the entire system is improperly optimized.

CLASS PROBLEMS

(1) A Mark II unit is pumping 750 B/D of a 35° API (S.G. = 0.85) oil. Total well depth is 3286 ft. The 2½-in. pump is set at 3037 ft in unanchored 3½-in. tubing. Pump submergence is 1,146 ft. The rod string is API No. 76. Pumping speed is 11.5 spm with a surface stroke of 104 in. Determine:
   (a) rod stretch
   (b) tubing stretch
   (c) plunger overtravel
   (d) effective plunger stroke
   (e) Is this a satisfactory stroke ratio (S_p/S)?

(2) An air balanced unit is pumping 738 B/D of total fluid (544 B/D of oil and 194 B/D of water). The oil gravity is 50° API. Well depth is 6440 ft. Pump depth is 5560 ft. The pump is 2½ in. set in 3½-in. anchored tubing. The rod string is API No. 97. Pumping speed is 8 spm with a surface stroke of 192 in. Determine:
   (a) rod stretch
   (b) tubing stretch
   (c) plunger overtravel
   (d) effective plunger stroke
   (e) Is this a satisfactory stroke ratio (S_p/S)?

2.8 CALCULATIONS FOR SURFACE EQUIPMENT

Now that the basic calculations and design factors in down-hole equipment have been discussed, it is important to look at the calculations and considerations required in the design and selection of surface equipment. There are several factors which must be considered:

(1) Counterbalance
(2) Torque on the gear reducer
(3) Horsepower of prime mover
(4) Speed reduction and engine sheave size

2.81 Counterbalance

In a beam pumping system, the amount of polished rod work needed to lift the fluid column is required only during the first half of the crank cycle, i.e., during the upstroke. If a beam pumping unit were not counterbalanced, the total work required of the prime mover would be performed during the upstroke when the maximum load of rods and fluid are being lifted. During the last half of the cycle (the downstroke) the prime mover would be "coasting" while the force of gravity pulls the rods and subsurface pump back down to their starting position.

Consequently, if the beam pumping unit were not counterbalanced, the prime mover would have to do all of its useful work during the upstroke portion of the cycle. This would require a relatively large prime mover and speed (gear) reducer and would produce fluid in a most ineffective, inefficient manner. Furthermore, the torque exerted on the gear reducer (which is con-

sidered the "heart" of the pumping unit) would be determined primarily by the uncounterbalanced load.

To reduce the size of the prime mover and gear reducer and to load the gear box more uniformly, the system is furnished with counterweights approximately equal to the buoyant weight of the rods plus half the weight of the fluid. This means that the "ideal" counterbalance required can be approximated by the average load throughout the cycle. This can be illustrated as follows:

C_i = "ideal" counterbalance effect

$$= \frac{\text{PPRL} + \text{MPRL}}{2} \qquad (2.32)$$

The PPRL given by the Mills equation actually neglects buoyancy and friction because they approximately cancelled each other. However, for the purpose of deriving an equation for effective counterbalance, all factors will be considered.

$$\text{Maximum load} = W_f + W_r(1+\alpha) - (\text{buoyancy}) + (\text{friction})$$

$$\text{Minimum load} = W_r(1-\alpha) - (\text{buoyancy}) - (\text{friction})$$

$$\text{Mean load} = \tfrac{1}{2}(\text{max. load} + \text{min. load})$$

$$= \tfrac{1}{2}[W_f + W_r(1+\alpha) + W_r(1-\alpha) - 2(\text{buoyancy})]$$

$$= \tfrac{1}{2}[W_f + W_r + W_r\alpha + W_r - W_r\alpha - 2(\text{buoyancy})]$$

$$= \tfrac{1}{2}[W_f + 2W_r - 2(\text{buoyancy})]$$

or,

$$C_i = \frac{W_f}{2} + W_r - \text{buoyancy}$$

The approximate ideal counterbalance effect is equal to half the weight of the fluid plus the buoyant weight of the rods.

Thus,

$$C_i = \frac{W_f}{2} + W_r - 0.127GW_r$$

$$= 0.5W_f + W_r(1 - 0.127G) \qquad (2.33)$$

Equation 2.33 can be written in the form:

$$C_i = \frac{W_f}{2} + W_2 \qquad (2.34)$$

where:

W_2 = buoyant weight of rods, pounds

On the upstroke, then, the prime mover is assisted by this counterbalance effect in lifting the combined weight of buoyant rods and fluid, resulting in an unbalanced force equivalent to only half the fluid weight. Thus, during the upstroke the prime mover needs only to lift an unbalanced force equivalent to half the fluid weight:

$$\text{(upstroke load)} - \text{(counterbalance)} = \text{unbalanced force}$$

$$(W_2 + W_f) - \left(W_2 + \frac{W_f}{2}\right) = \frac{W_f}{2}$$

On the downstroke, the return of the buoyant rods is opposed by the counterbalance, leaving an unbalanced force again equivalent to half the fluid load.

Thus,

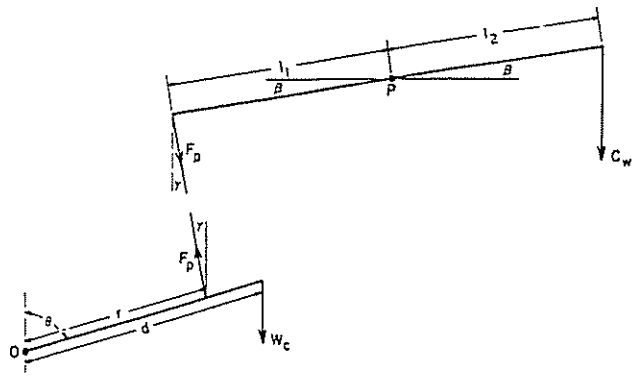


Fig. 2.21 Counterbalance effect of the counterweight (after Ref. 5)

$$\text{(counterbalance)} - \text{(downstroke load)} = \text{unbalanced force}$$

$$\left(W_2 + \frac{W_f}{2}\right) - W_2 = \frac{W_f}{2}$$

When properly counterbalanced, the upstroke and downstroke work requirements of the prime mover will be approximately equal. Likewise, the torque exerted on the gear reducer will be approximately equal during the upstroke and downstroke portions of the pumping cycle. By properly counterbalancing a beam pumping unit, the instantaneous torque requirement of the gear reducer and prime mover may be considerably less than half the torque required for an uncounterbalanced system. Consequently, proper counterbalancing of the pumping unit is one of the principal ways in which prime mover and gear reducer size and peak torque may be reduced markedly. Example 2.8 will illustrate this condition.

EXAMPLE PROBLEM #8

Consider a pumping system as follows:

Buoyant weight of rods (W_2) = 10,000 lb

Weight of fluid (W_f) = 4,000 lb

Then,

$$\text{counterbalance} = \frac{W_f}{2} + W_2$$

$$= \frac{4{,}000 \text{ lbs.}}{2} + 10{,}000 \text{ lb}$$

$$= 12{,}000 \text{ lb}$$

Determine unbalanced force during both the upstroke and downstroke for: (a) counterbalanced and (b) uncounterbalanced beam pumping units (disregarding friction and dynamic effects).

(a) Solution for properly counterbalanced unit

Upstroke:

$$\text{Upstroke load} - \text{counterbalance} = \text{unbalanced force}$$

$$(10{,}000 \text{ lb} + 4000 \text{ lb}) - 12{,}000 \text{ lb} = 2000 \text{ lb}$$

Downstroke:

$$\text{Counterbalance} - \text{downstroke load} = \text{unbalanced force}$$

$$12{,}000 \text{ lb} - 10{,}000 \text{ lb} = 2{,}000 \text{ lb}$$

Thus, for a properly counterbalanced unit the unbalanced force is equal during the upstroke and downstroke and is equivalent to one-half the fluid weight.

(b) Solution for an uncounterbalanced unit:
Upstroke:

$$\text{Upstroke load} - \text{counterbalance} = \text{unbalanced force}$$
$$(10{,}000 \text{ lb} + 4{,}000 \text{ lb}) - 0 = 14{,}000 \text{ lb}$$

Downstroke:

$$\text{Counterbalance} - \text{downstroke load} = \text{unbalanced force}$$
$$0 - 10{,}000 \text{ lb} = -10{,}000 \text{ lb}$$

In an uncounterbalanced system, the gear reducer and prime mover must handle (during the upstroke) a polished rod load equivalent to the weight of the rods plus the weight of the fluid. Furthermore, the upstroke and downstroke loads may differ greatly, as illustrated above.

From a mechanical standpoint, when the rods are elevated by the pumping unit from the bottom to the top of the stroke, a quantity of potential energy is stored in the system. When the rods are pulled down by the force of gravity from the top to the bottom of the stroke, this same quantity of potential energy is restored by elevating the counterweights.

Thus, the *net* mechanical work (raising and lowering the rods) during one crank revolution is zero. In an uncounterbalanced unit, the prime mover must labor relatively hard on the upstroke to lift both rods and fluid; on the downstroke, potential energy is no longer stored in the system. On the other hand, in a counterbalanced system the prime mover is assisted during the upstroke by the fall of the counterweights, while on the downstroke, the pull of gravity on the rod string helps the prime mover elevate the counterweights, thereby storing potential energy in the system. Thus, one of the most important aspects of pumping installation design is the proper selection of counterbalance.

Equation 2.34 gives the theoretical ideal counterbalance effect, but the actual counterbalance effect may be somewhat different, depending upon the geometry of the pumping unit, the stroke length and position of the counterweight on the walking beam or crank, or other considerations. Because of these factors, the accurate calculation of counterbalance is nearly impossible in many cases. Actually, the final adjustments in counterbalance are generally made experimentally in the field. Nevertheless, an estimate of the ideal counterbalance must be made in the design and selection of a specific pumping unit, and it must not exceed the rated counterbalance available for any specific unit selected.

Figure 2.21 shows the actual counterbalance effect which can be obtained from a counterweight, W_c, and by considering the geometry of a conventional unit. The total counterbalance effect, C_t, is the sum of the counterbalance effect, C_w, produced by the counterweights, W_c, plus the counterbalance effect, C_s, which may result from structural unbalance of the pumping unit selected. Thus, the total counterbalance effect at the polished rod is given by:

$$C_t = C_s + C_w \tag{2.35}$$

and

$$C_w = W_c(d/r)(l_1/l_2) \tag{2.36}$$

thus

$$C_t = C_s + W_c\,(d/r)(l_1/l_2) \tag{2.37}$$

where:

d = the distance from the crankshaft to the center of gravity of the counterweight
r = the distance from the crankshaft to the pitman bearing
l_1 = the distance from the saddle bearing to the tail bearing
l_2 = the distance from the saddle bearing to the bridle
C_s = approximate structural unbalance (given by the manufacturer for a specific unit), lb
W_c = weight of counterweight used with the unit, lb

As an example, Equation 2.37 could be used for a particular unit with particular geometry to calculate the position d of the counterweight on the crank arm that would give the theoretical ideal counterbalance effect ($C_t = C_i$) calculated from Equation 2.34.

EXAMPLE PROBLEM #9

The maximum and minimum loads for a pumping installation were determined from a dynamometer card to be 15,500 lb and 3,500 lb, respectively. The well has a 2-in. plunger on ⅞-in. rods with a tubing anchor set at 5,000 ft. Counterweights on the unit weigh 6,000 lbs. The unit dimensions (see Fig. 2.21) are: $d = 40$ in., $r = 28$ in., $l_1 = l_2$. Actual total counterbalance effect is 10,000 lbs. Determine the following:
a) Structural unbalance, C_s.
b) Ideal counterbalance effect, C_i.

Solution:

a) $C_t = C_s + W_c\left(\frac{d}{r}\right)\left(\frac{l_1}{l_2}\right)$ (2.37)

$$C_s = C_t - W_c\left(\frac{d}{r}\right) = 10{,}000 - 6{,}000\left(\frac{40}{28}\right)$$
$$= 1430 \text{ lbs}$$

b) $C_i = \dfrac{\text{max. load} + \text{min. load}}{2}$

$$= \frac{15{,}500 + 3{,}500}{2}$$
$$= 9{,}500 \text{ lbs}$$

2.82 Torque considerations

Torque can be defined as a force acting (at right angles) at the end of a lever arm, multiplied by the length of the arm, tending to produce rotation and work. When applied to pumping units, torque refers to the number of inch-pounds of force applied to the crank by the low-speed shaft of the gear reducer; it is created by the pitman pull due to well loads and by an oppos-

ing effect from counterbalance moments and by the prime mover.

Basically, the net crankshaft torque of a beam pumping unit is the difference between well load torque and counterbalance torque at any position of the crank. This net crankshaft torque is the actual torsional load "seen" by the prime mover and gear box during the crank cycle. The maximum net torque is referred to as peak torque and is the value to which the gear reducer is designed. Thus, in any pumping installation the actual peak torque occurring during the pumping cycle must not exceed the maximum torque rating (capacity) of the gear box or speed reducer.

On the conventional unit, the peak torque generally occurs twice during each revolution of the crank where the difference between the well load moment and the counterbalance moment (or vice-versa) is maximum. This normally occurs near the middle of the stroke (S/2). Consequently, the gear reducer must be designed to handle this peak torque. All else equal, peak net torque is a function of the difference between peak and minimum polished rod loads, i.e., the rod load range.

A simple relationship for approximate peak torque on the upstroke is:

$$PT_u = (PPRL - C_t)(S/2) \qquad (2.38)$$

where:

PT_u = peak torque on upstroke, in.-lb
PPRL = peak polished rod load, lbs
C_t = total counterbalance effect, lbs
S = polished rod stroke, in.

A simple relationship for approximate peak torque on the downstroke is:

$$PT_D = (C_t - MPRL)(S/2) \qquad (2.39)$$

where MPRL = minimum polished rod load, lb

Peak torque during both the upstroke and downstroke should be calculated in the design of a pumping installation because conditions could exist where either one could be greater than the maximum allowable torque rating on the gear reducer. Actually, the peak torque could occur at any point during a complete crank revolution, and a detailed torsional analysis (to be discussed later) should be made for any given installation. This technique requires knowledge of the continuous well load (polished rod dynamometer card), the unit's "torque factors," the counterbalance effect, and the loss in mechanical efficiency of the pumping unit's structural bearings.

2.821 Torque factors

One of the most important concepts in a beam pumping system is the *torque factor*; without it, the exact torsional load that the rod string places on the speed (gear) reducer at a particular crank position would be difficult to determine.[23]

Consider a pumping unit whose moving parts (i.e. the walking beam, pitmans, horsehead, cross yoke, etc.) are all weightless and the counterbalance moment is zero. Suppose further that momentarily the cranks are positioned at 90°. If a load is hung on the horsehead of 1 lb and a torque wrench, applied to the crankshaft, measures 30 in.-lb of torque, then the torque factor at this 90° position of the crank would be 30.

In other words, the crankshaft torque of 30 in.-lbs divided by 1 lb at the polished rod equals 30 in., or the torque factor is 30. Thus, the torque factor at any crank position (assuming zero counterbalance torque) is the ratio of the torque at the crankshaft to the load at the polished rod that produced it, assuming the components of the system are weightless. The torque factor can also be defined as that factor which, when multiplied by the instantaneous polished rod load, will give the (gross or well load) torque on the crankshaft at that crank position.

A different torque factor exists for every crank position of any particular pumping unit, and every different pumping unit has a different set of torque factors. Generally speaking, torque factors are calculated for every 15° of crank rotation around the crank circle of 360°; for convenience, there would be 24 different torque factors, each 15° apart. A close approximation to the maximum torque factor in a symmetrical pumping unit is equal to one-half the stroke length, (S/2).

The equations given previously for peak torque on the upstroke and downstroke can now be expressed in terms of the unit's maximum torque factors (wherever they occur in the crank cycle). A relationship called "load torque," which gives an approximate torque for *any* unit geometry (conventional, air balance, or Mark II), is as follows:

$$PT = \frac{\frac{1}{2}[PPRL(TF_1) - MPRL(TF_2)]}{0.93} \qquad (2.40)$$

where:

PT = peak torque for the entire crank cycle, in.-lbs
PPRL = peak polished rod load, lb
MPRL = minimum polished rod load, lb
TF_1 = maximum upstroke torque factor
TF_2 = maximum downstroke torque factor

The factor of 0.93 in the denominator simply assumes an arbitrary 7% loss in mechanical efficiency due to friction in the structural bearings.

If $TF_1 = TF_2$ (as in the symmetrical conventional and air balance units) and the counterbalance effect, C_t, is approximately one-half of the sum of PPRL and MPRL (see Equation 2.32), then Equation 2.40 can be written as:

$$PT = \frac{(PPRL - C_t)(TF_{max})}{0.93} \qquad (2.41)$$

where TF_{max} is the maximum torque factor, wherever it occurs. Equation 2.41 is a special case of the general formula (2.40) and applies to symmetrical unit geometries such as conventional and air balance units. Equation 2.40 would have to be used for non-symmetrical unit geometry, such as Mark II units. Both Equations 2.40 and 2.41 assume that the maximum peak torque occurs at 90° of the upstroke and that the maximum counterbalance effect occurs at the same position.

Table 2.12 gives approximate maximum torque factors for conventional units; Table 2.13 gives approximate maximum torque factors for Mark II units.

TABLE 2.12
APPROX. MAXIMUM TORQUE FACTORS
CONVENTIONAL UNITS[23]

Stroke, in.	Torque factor	Stroke, in.	Torque factor
16	8.5	64	34
24	13	74	39
30	16	86	45
36	19	100	52
42	22	120	63
48	26	144	75
54	29	168	87

TABLE 2.13
APPROX. MAXIMUM TORQUE FACTORS
MARK II UNITS[23]

Stroke, in.	TF_1	TF_2
64	29	37
74	34	43
86	39	51
100	47	57
120	55	71
144	66	88
168	79	102

EXAMPLE PROBLEM #10

Estimate peak and minimum polished rod loads, counterbalance required, and peak torque for both Mark II and conventional units for the following conditions:

Pumping depth = 5900 ft
Desired fluid production = 150 B/D
Volumetric efficiency = 80%
Stroke length = 64 in.
Pumping speed = 16.5 spm
Pump diameter = 1¼ in.
Rod number = API No. 76
Fluid specific gravity = 1.0

Solution:

1) Conventional unit:

W_r = weight of rods
W_r = (5900 ft)(1.814 lb/ft) (Table 2.7)
W_r = 10,703 lbs

W_f = weight of fluid
$W_f = 0.433G(L A_p - 0.294 W_r)$ (2.16)
$W_f = 0.433(1.0)[5900(1.227) - 0.294(10,703)]$
W_f = 1772 lb

PPRL = peak polished rod load
$PPRL = W_f + W_r(1 + \alpha)$ (2.12)

$$\alpha = \frac{S N^2}{70,500} = \frac{64(16.5)^2}{70,500}$$

α = 0.247 (Table 2.10)

so,

PPRL = 1772 + (10,703)(1 + 0.247)
PPRL = 15,119 lb

MPRL = minimum polished rod load
$MPRL = W_r(1 - \alpha - 0.127\,G)$ (2.14)
MPRL = 10,703 [1 − 0.247 − 0.127 (1.0)]
= 6700 lb

C_1 = counterbalance required

$$C_1 = \frac{PPRL + MPRL}{2} \quad (2.32)$$

$$= \frac{15119 + 6700}{2}$$

C_1 = 10,910 lb

PT = peak torque

$$PT = \frac{(PPRL - C_1)(TF_{max})}{0.93} \quad (2.41)$$

$$= \frac{(15,119 - 10,910)(34)}{0.93} \quad \text{(Table 2.12)}$$

PT = 153,900 in-lb

2) Mark II unit:

$PPRL = W_f + W_r(1 + 0.6\alpha)$ (2.12)
= 1772 + (10,703)[1 + 0.6(0.247)]
PPRL = 14,061 lb

$MPRL = W_r(1 - 1.4\alpha - 0.127\,G)$ (2.14)
= (10,703)[1 − 1.4(0.247) − 0.127]
MPRL = 5643 lb

$$C_1 = \frac{PPRL + MPRL}{2}$$

$$= \frac{14,061 + 5643}{2}$$

C_1 = 9852 lbs

$$PT = \frac{½[PPRL(TF_1) - MPRL(TF_2)]}{0.93} \quad (2.40)$$

From Table 2.13,
TF_1 = 29; TF_2 = 37

$$PT = \frac{½[14,061(29) - 5643(37)]}{0.93}$$

PT = 107,000 in.-lb

CLASS PROBLEMS

(1) For the data of Class Problem #1 in Section 2.731, determine:
(a) Peak polished rod load, PPRL
(b) Minimum polished rod load, MPRL
(c) Counterbalance required
(d) Peak torque
(2) Repeat Problem 1 for a conventional unit. Compare the results. What can you conclude?

2.822 An important aspect of energy conservation in a beam and sucker rod pumping system

When elevating fluid from a particular depth with an electrically driven sucker rod pumping system, part of the input energy is converted into useful mechanical work at the polished rod while the remainder is dissipated in the motor as heat loss. Consequently, the input energy must equal the sum of (1) the fluid-lifting, mechanical work and friction, plus (2) the work equivalent of the heat loss. Obviously, the smaller the heat loss for a given input, *the greater the amount of energy the beam system devotes to useful, mechanical pumping work.*

Suppose two electric motors, one 80% efficient and the second 90% efficient, are alternately applied to a *constant* torque load. With the first motor, approximately 80% of its input energy is converted into beneficial mechanical work output, while 20% is dissipated as heat loss. In the second motor, 90% of its input energy is transformed into useful work, while only 10% is converted into wasted heat.

For any electrically-driven beam and sucker rod pumping application, the total heat loss is a function of the average (constant torque) efficiency of the motor as well as a function of the magnitude of the current or power fluctuations, which are in turn porportional to the pumping unit's torque load variation. All else equal, the greater the torque variation that the unit presents to its prime mover, the higher the rms (root-mean-square) current value; the lower and smoother these pumping unit torque excursions, the smaller the rms current.

For a given pumping application, the closer the rms value of the motor current approaches its average value, the greater the proportion of input energy devoted to useful mechanical work and the smaller the amount dissipated as heat loss. This means that the smoother the mechanical torque load pattern (i.e. the lower the torque range) the beam unit presents to its electric prime mover, the closer the rms current approaches the average current, the smaller the energy (heat) loss, and the more efficient a beam and sucker rod system becomes.

A well-known authority on sucker rod pumping clearly demonstrates this in Figure 2.22.[24] In each of the three (torque) amperage curves, the average current and average pumping unit mechanical torque are the same—indicating the same work load at the polished rod. In Figure 2.22(a), torque loading is relatively uniform and the rms current value closely approaches the average current value, effecting a highly efficient condition with minimal heat loss.

For convenience assume that the useful mechanical work is a function of the area underneath the average current (or torque) value, while the heat loss, due to torque load variation, is a function of the area between the rms current value and the average current (or torque) value. With this mechanical torque smoothing, the area between the two current lines is very small, indicating small heat loss and high motor efficiency.

In Figure 2.22(b), the average current, mechanical torque, and work output of the unit are the same; however, the wider torque fluctuations produce a higher rms current because the heat loss is a function of the square of the current. Consequently, the heat loss is significantly greater because of the unit's wider torque fluctuations. This higher heat loss is still proportional to the area between the average rms current and the average current and in this application is significantly greater than in Figure 2.22(a), which presents a smoother torque load to its prime mover.

Figure 2.22(c) shows an extremely wide fluctuating torque load with its proportionally higher rms current value and accompanying increased heat loss. Again, in Figure 2.22(c) the pumping unit is performing the same average work at the polished rod with the same average torque as in Figures 2.22(a) and 2.22(b), but the heat loss is still greater and is again proportional to the area between the rms current and the average current, which is considerably greater than in Figure 2.22(b).

Although the average current and work output are the same for a particular beam pumping application, the smoother the torque load that a pumping unit presents to its prime mover, the greater the amount of useful mechanical work output, and the smaller the amount of heat loss, with a significantly increased degree of energy conservation.

This discussion emphasizes why smoothing out the torque load mechanically, before it reaches the prime mover, can become a significant method of energy conservation. By mechanically smoothing out the pumping unit's torque load—without sacrificing prime mover efficiency—and presenting a relatively uniform torque to the prime mover, substantial energy savings result.

2.83 Prime movers

The optimum selection of the type and size of prime mover is one of the more important aspects of pumping installation design. There are two types which are commonly used:

(1) internal combustion engines
(2) electric motors

Internal combustion engines, either single-cylinder, two-cylinder, or multi-cylinder, have been the mainstay of pumping unit installations for many years. However, for various reasons—some of which were indicated in Section 2.2—electric motors have become increasingly popular over the years; perhaps no other component of the beam pumping system has proven to be more reliable and more trouble-free than the electric motor prime mover.

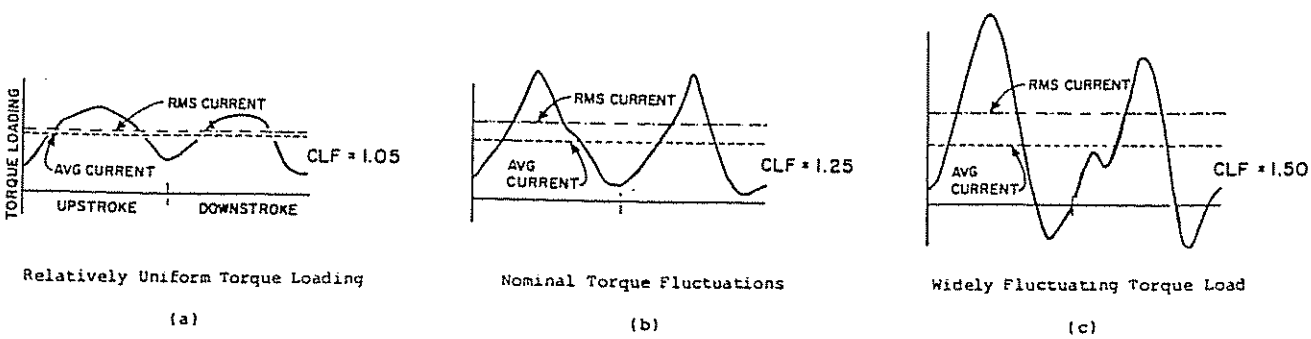


Fig. 2.22 Relationship of the RMS Current to the (Constant) Average Current Typical Beam Pumping Application, As Torque Load Range Increases (after Ref. 24)

There are four popular types of electric motors used in oilfield pumping installations:[26]

(1) *NEMA design "C"*. Normal slip motor (less than 5% slip). Popular prime mover because of moderately high starting torque characteristics and relatively low initial cost.
(2) *NEMA design "D"*. Medium slip (5% to 8% slip). More popular for oil well pumping because of higher starting torque and higher-slip characteristics.
(3) *NEMA design "D"*. High slip (8% to 13% slip). Currently the most popular choice as a prime mover for a pumping unit.
(4) *Ultra-high slip motors*. (slip as high as 30 to 40%) Latest development in prime movers. Ultra-high-slip characteristics make it more fully loaded through the pumping cycle than lower-slip motors. Applications are still being evaluated.

Slip is the difference between unloaded motor speed and the speed under load, expressed as percent of unloaded speed.[22] In general, the lower the slip, the smaller the initial cost and the higher the average efficiency, when considered across the normal torque range of the motor.[26] Higher slip motors, however, are often used because of more effective utilization of inertial torque.

2.84 Prime mover horsepower requirements

Most of this discussion will apply to electric motors because of the popularity and increasing use of these motors. However, the principles discussed will apply, to a great extent, equally to the use of internal combustion engines.

The total nameplate motor horsepower, HP_{np}, can be determined from:

$$HP_{np} = \frac{(PRHP) \times (CLF)}{E_s} \tag{2.42}$$

where:

PRHP = polished rod horsepower
CLF = cyclic load factor
E_s = surface efficiency of the pumping system.

The polished rod horsepower is the rate of the work energy delivered to the polished rod by the pumping unit. Hydraulic work is the amount of frictionless work which must be performed by the bottomhole pump in lifting the fluid to the surface. In other words, the bottomhole pump must deliver an amount of energy equal to the value of the hydraulic work plus any *frictional work* (subsurface losses). Thus:

$$PRHP = HP_h + HP_f \tag{2.43}$$

where:

HP_h = Hydraulic horsepower
HP_f = Friction horsepower (or subsurface losses expressed as horsepower)

The polished rod horsepower can be determined in several ways:

(1) It can be measured by a dynamometer (to be discussed later).
(2) It can be derived from API RP-11L, "Design Calculations For Pumping Systems" (to be discussed later).
(3) If structural efficiency is assumed to be 100% the PRHP could be determined from:

$$PRHP = \frac{(\text{Net Torque})_{avg.} \times N}{63{,}025} \tag{2.44}$$

In this method, it would only be necessary to sum up the net torque values (accounting for whether the torque is + or −) and divide by the number of points taken. This would give the average torque per stroke. Multiplying this average torque by the pumping speed, N, and dividing by the conversion constant 63,025, would give the value of the polished rod horsepower.
(4) In the absence of a more rigorous method, the PRHP can be estimated by approximation formulas, as follows:

HP_h = hydraulic horsepower
= power required to lift a given volume of fluid vertically, through a given distance in a given period of time.

$$HP_h = \frac{Q \times L_n \times (350 \text{ lb/bbl}) \times G}{(33{,}000 \text{ ft. lb/min/HP})(24)(60)}$$

$$HP_h = 7.36 \times 10^{-6}\, QGL_n \tag{2.45}$$

Where L_n = net lift, ft

Now, if it is recognized that $0.433L_NG$ = psi = pressure exerted by a column of fluid of length L_N, then Equation 2.45 can be written:

$$HP_h = 0.000017 \times Q \times \text{psi} \tag{2.46}$$

Hydraulic horsepower can then be determined, for a given net lift, from Equation 2.46 or Figure 2.23.

2.841 Net lift

The net lift, L_N, represents the total pressure differential through which the fluid moves, in pressure units of feet of fluid. If the pump is set at the working fluid level in the casing (assuming no gas pressure in the casing annulus), then the net lift is simply the depth to the pump. But if the pump is set at a depth below the working fluid level, then the net lift will be affected by casing pressure exerted at the pump. In either case, the net lift may also be affected by any surface tubing back-pressure which tends to act against the lifting of the fluids.

If the pump is set at the working fluid level (L = D), then the net lift is given by:

$$L_N = L + \frac{P_t \text{ lb/in.}^2 \times 144 \text{ in.}^2/\text{ft}^2}{62.4\ G \text{ lb/ft}^3}$$

$$= L + \left(\frac{2.31\ P_t}{G}\right) \tag{2.47}$$

If there is no tubing pressure ($P_t = 0$), then

$$L_N = L = D \tag{2.48}$$

For the case where the pump is set at a depth below the working fluid level (L > D), the net lift is given by:

$$L_N = L - (L - D) + \left(\frac{2.31\ P_t}{G}\right)$$

$$= D + \left(\frac{2.31\ P_t}{G}\right) \tag{2.49}$$

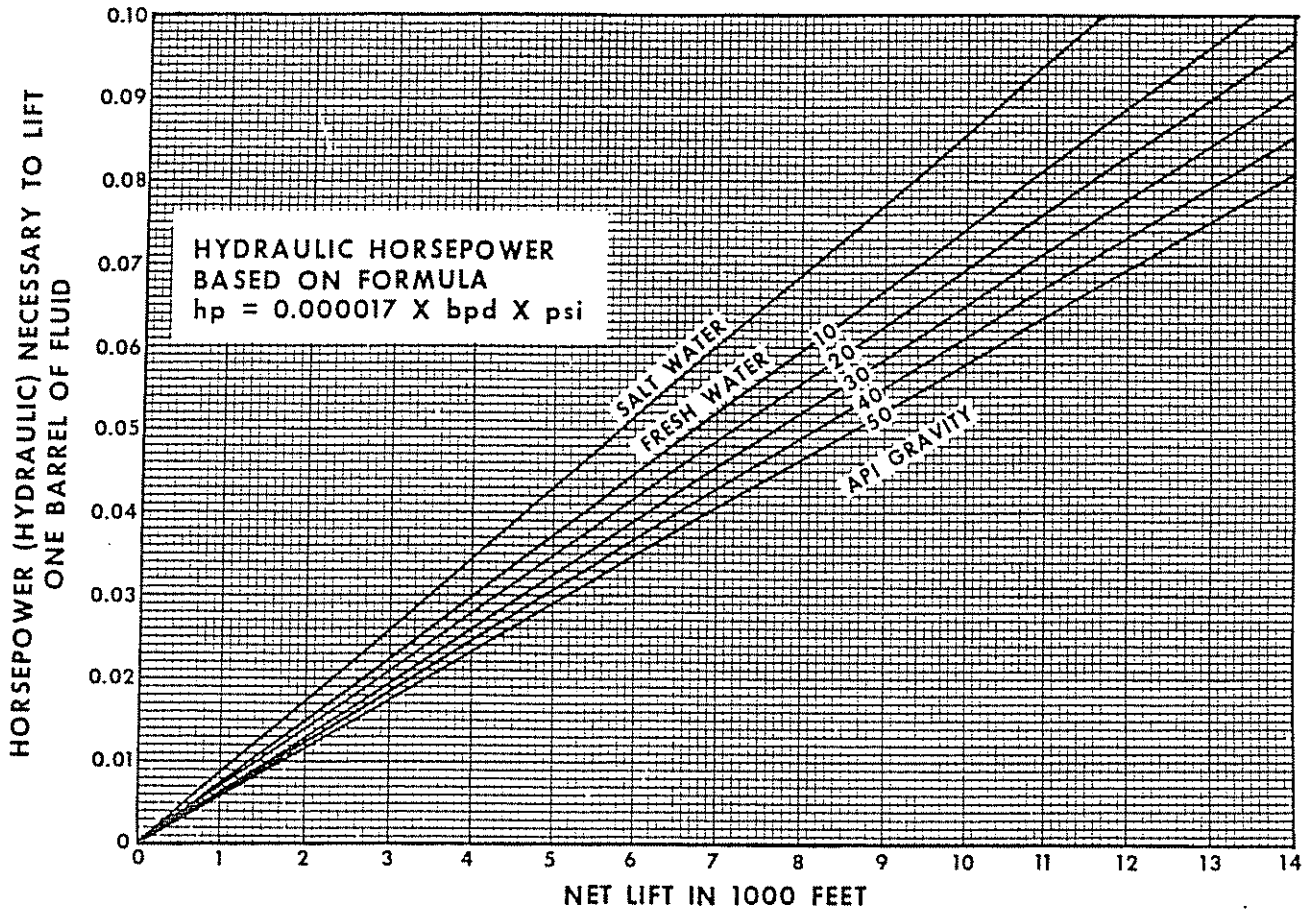


Fig. 2.23 HYDRAULIC HORSEPOWER (after Ref. 2)

Note that in both cases, if the tubing pressure is negligible, the net lift is simply the depth to the working fluid level in the casing annulus.

2.842 Frictional horsepower

Only a portion of the horsepower required in a pumping installation is used in effective lifting of fluids. Additional horsepower is required to overcome or compensate for the frictional losses between the pump and polished rod. Frictional losses may be difficult to determine accurately, but an empirical relationship has been presented by which these frictional losses can be *approximated* (in the absence of a more definitive relationship).[14]

$$HP_f = \frac{(W_r/8)(2S)(N)}{(33{,}000)(12)} \qquad (2.50)$$

where:

H_f = horsepower to overcome friction
W_r = weight of rods, lb
S = stroke length, in.
N = pumping speed, strokes per minute

Equation 2.50 can be written:

$$H_f = 6.31 \times 10^{-7}\ W_r\ SN \qquad (2.51)$$

Another reference suggests (for air-balanced and Mark II systems):[22]

$$HP_f = 6.25 \times 10^{-7}\ W_r SN \qquad (2.52)$$

This approach to determining frictional horsepower is only an approximation and is not recommended when better approaches are available.

2.843 Cyclic load factor (CLF)

In a typical beam pumping system, the prime mover must deliver enough power to meet (1) hydraulic horsepower requirements, (2) surface equipment losses (i.e., belts, bearings, gears, stuffing box, etc.), (3) subsurface losses (i.e., downhole friction of all kinds, thermal losses, etc.), and (4) additional reserve power to handle the cyclic nature of the beam pumping load effectively. This last component is called the "Cyclic Load Factor" (CLF).

The average motor current is proportional to the pumping load; if the pumping load is constant, the RMS or thermal current drawn by the motor closely approaches the average current.

On cyclic loads such as beam type pumping units where high peak currents are drawn because of the variable nature of the pumping load, the RMS or thermal amperes substantially exceed the average current. The ratio of the RMS current to the aver-

age current in a beam pumping unit is defined as the Cyclic Load Factor.

$$CLF = \frac{\text{RMS current}}{\text{Avg. current}} = \frac{I_{RMS}}{I_{AVG.}}$$

or,

$$CLF = \frac{\sqrt{\frac{I_1^2 + I_2^2 + I_3^2 + \cdots + I_n^2}{n}}}{\left[\frac{I_1 + I_2 + I_3 + \cdots + I_n}{n}\right]} \qquad (2.53)$$

where:

I = instantaneous motor currents at crank positions 1, 2, 3, . . ., n.

For a given polished rod work load, the lower the torque peaks and the more uniform the torque range, the lower the cyclic load factor, as illustrated in Figure 2.22. As the cyclic load factor decreases—all else equal—less thermal loss occurs in the electric prime mover. Prime mover size requirement is reduced and power costs are lowered as well.

For *estimation* purposes, experience has shown average cyclic load factors for nonuniform torque pumping units (air balance and conventional units) to be as follows:[23]

CLF = 1.375 for NEMA "D" electric motors and slow speed engines

CLF = 1.897 for NEMA "C" electric motors and multi-cylinder engines.

For uniform torque units (Mark II units), the average cyclic load factors are:[23]

CLF = 1.10 for NEMA "D" electric motors and slow speed engines

CLF = 1.517 for NEMA "C" electric motors and multi-cylinder engines

If a computed cyclic load factor is unavailable, the typical (appropriate) empirical CLF given above for determining the total prime mover (nameplate) horsepower required may be used (see Equation 2.42).

$$HP_{np} = \frac{(PRHP) \times (CLF)}{E_s} \qquad (2.42)$$

2.844 Surface efficiency in a beam and sucker rod pump system

Any formula for motor nameplate horsepower rating must recognize each of the three factors in Equation 2.42 accurately and rationally. However, because subsurface losses and the CLF vary widely with well characteristics and conditions of operation, in the past it has been impossible to establish a formula with the value of these factors fixed to give an accurate motor nameplate horsepower for all wells. Local conditions must be considered and formula factors evaluated accordingly.

With the advent of advanced and sophisticated beam pumping unit predictive technology, much of which is due to the outstanding work of Dr. S. G. Gibbs, a motor nameplate horsepower rating may be determined more precisely than before. The following is a brief examination of these three variables.

Polished rod horsepower can be developed with precision by planimetering a surface dyamometer card or by integrating instantaneous force ordinates throughout one revolution of the crank, while accounting for appropriate machinery losses.

The cyclic load factor can be developed in the conventional manner by determining rms values of the various instantaneous motor currents throughout one crank revolution and dividing this value by the average motor current across the same interval. A mechanical CLF could also be used if all torsional moments, both regular and inertial, are properly considered at the motor output, and positive and negative sign conventions are also recognized.

The one variable that has not been precisely determined is the appropriate *surface efficiency* across one revolution of the crank. This figure would consider losses per crank revolution in wirelines, structural bearings, transmissions, V-belts, and the electric motor.

The problem of determining surface efficiency has been studied, and several years ago an engineer with Humble Oil & Refining collaborated with Lufkin Industries' engineering department to determine a curve of surface efficiency based on a percentage loading of the surface components. This was an improvement over a single fixed constant for all applications and conditions, but lacked much rigor and rationale. This curve was developed empirically, and its validity was not established conclusively. While this surface efficiency curve may leave something to be desired, Figure 2.24 can estimate surface efficiency, and a subsequent example illustrates the procedure.

Recently, a more rigorous surface efficiency relationship has been developed.[23] Its assumptions are minimal, and its rationale is clear.

In considering the efficiencies of wirelines, structural bearings, transmission, belts, and electric motor prime mover, the efficiencies are divided into two parts: (1) the motor efficiency and (2) the combined efficiencies of the remaining mechanical (moving) components. Looking first at mechanical efficiencies, an average per stroke loss in the wirelines and structural bearings would be approximately 3% and would not significantly vary with the degree of loading but could be considered a reasonably constant value. Loss in the transmission, while probably more variable, should be in the neighborhood of 4% at low speeds. This gear and gearbox bearing loss could also be considered approximately constant throughout the crank revolution. If it did vary, 4% is such a small value that consideration of variable efficiencies would be of little significance. Also, it was determined that a 3% loss across the V-belts would be fairly representative and approximately constant.

Assuming that all three mechanical efficiencies would be nominally independent of the degree of loading, it is reasonable to assume that on the typical beam and sucker rod pumping unit a 90% overall average mechanical efficiency exists, almost uniformly, across one crank revolution.

Fortunately, the contour of the motor efficiency curve is well-known and upon request will normally be furnished by the manufacturer. Thus, the remaining problem is of weighting the various instantaneous

$$\text{Nominal H.P.} = \frac{T \times N}{63{,}000 \times 1.57}$$

API Torque Rating In. Lbs.	Nominal Horsepower Rating At 20 SPM	API Torque Rating In. Lbs.	Nominal Horsepower Rating At 20 SPM
40,000	8	456,000	93
57,000	11	640,000	130
114,000	25	912,000	185
160,000	33	1,280,000	260
228,000	46	1,824,000	370
320,000	65	2,560,000	526

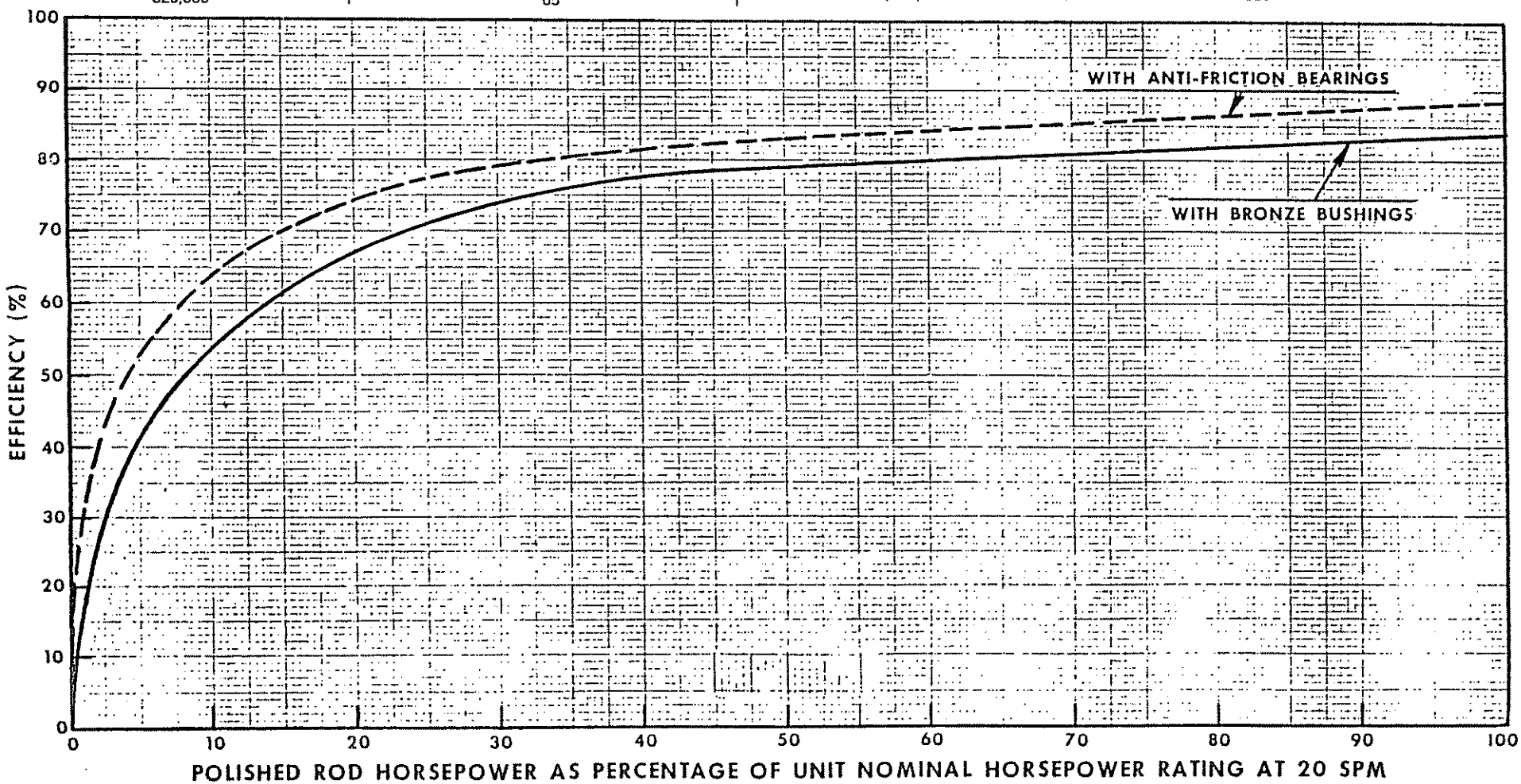


Fig. 2.24 (after Ref. 23) SPEED REDUCER EFFICIENCY VS. HORSEPOWER LOADING

motor efficiencies throughout one revolution of the crank. It would not be rigorous to assume an average unweighted efficiency throughout 360°, because more torque demand or more current flow might occur at one point in the cycle than at another.

Thus, it is necessary to determine the contour of the average motor efficiency per stroke after weighting each efficiency ordinate by the amount of current flowing at a particular instant. When this weighted average motor efficiency curve is determined and depreciated by a 10% mechanical loss (i.e., 90% mechanical efficiency), a close determination of surface efficiency can be made. Thus, the weighted average surface efficiency number for all components (lines, structural bearings, transmission, belts, and motor) across one crank cycle is as follows:

$$E_s = (0.90)\frac{\sum_{0°}^{360°}(746 \times \text{ouptut HP}_{\text{inst.}})}{\sum_{0°}^{360°}\left[\frac{(746 \times \text{output HP}_{\text{inst.}})}{(\text{motor eff.})_{\text{inst.}}}\right]} \quad (2.54)$$

A series of checks have shown this method of surface efficiency determination to be accurate, both from a theoretical and a practical standpoint.

To predict electric motor nameplate horsepower in the most accurate manner, use of this more rigorous relationship to determine surface efficiency of both the electrical and mechanical components would be desirable and is as follows:

$$HP_{np} = \frac{(PRHP)(CLF)\sum_{0°}^{360°}\left[\frac{746 \times \text{output HP}_{\text{inst.}}}{(\text{motor eff})_{\text{inst.}}}\right]}{(0.90)\sum_{0°}^{360°}(746 \times \text{output HP}_{\text{inst.}})} \quad (2.55)$$

Computerizing and solving Equation 2.55 should predict an accurate, acceptable, and economical motor nameplate horsepower rating for any (in balance) beam pumping unit geometry, over any normal application.

EXAMPLE PROBLEM #11

Well and pumping unit data:

Pump depth = 4500 ft
Production (100% VOL. EFF.) = 150 bf/d
RODS: No. 76 (⅞" and ¾")
Plunger Diameter = 1.25 in.
Stroke length = 64 in.
Pumping speed = 13.2 spm (conventional)
13.1 (Mark II)

Determine prime mover (nameplate) horsepower for the following four conditions:

(1) Conventional unit driven by NEMA "D" motor
(2) Conventional unit driven by NEMA "C" motor
(3) Mark II unit driven by NEMA "D" motor
(4) Mark II unit driven by NEMA "C" motor

Solution:

(1) Conventional unit—NEMA "D" motor
(See API-11L3 conventional unit design book, page 218)
Peak torque—141,000 in.-lb (in-balance)
Unit required—160,000 in.-lb (API)
Nominal horsepower—33 (from Fig. 2.24)
Polished rod horsepower—7.0 (from API—11L3 tables) (see p. 48)

$$\frac{PRHP}{HP_{nom}} = \frac{7}{33} = 0.212$$

From Figure 2.24, 0.212 corresponds to 67.5% surface efficiency.

$$HP_{Nameplate} = \frac{7 \times 1.375}{0.675} = 14.3 \text{ NEMA "D" nameplate horsepower requirement}$$

(2) Conventional unit—NEMA "C" motor

$$HP_{Nameplate} = \frac{7 \times 1.897}{0.675} = 19.7 \text{ NEMA "C" nameplate horsepower requirement}$$

(3) Mark II unit—NEMA "D" motor (see Mark II design table) p. 49

Peak torque—94,000 in.-lb (in-balance)
Unit required—114,000 in.-lb (API)
Nominal horsepower—25 (from Fig. 2.24)
Polished rod Horsepower—6.9

$$\frac{PRHP}{HP_{nom}} = \frac{6.9}{25} = 0.276$$

From Figure 2.24, 0.276 corresponds to 78% surface efficiency.

$$HP_{Nameplate} = \frac{6.9 \times 1.1}{0.78} = 9.7 \text{ NEMA "D" nameplate horsepower requirement}$$

(4) Mark II unit—NEMA "C" motor

$$HP_{Nameplate} = \frac{6.9 \times 1.517}{0.78} = 13.4 \text{ NEMA "C" nameplate horsepower requirement}$$

Thus, on this particular application:

(1) The conventional unit driven by a NEMA "D" motor requires 14.3 nameplate horsepower, i.e. 15 HP motor.
(2) The conventional unit driven by a NEMA "C" motor requires 19.7 nameplate horsepower, i.e. 20 HP motor.
(3) The Mark II unit driven by a NEMA "D" motor requires 9.7 nameplate horsepower, i.e. 10 HP motor.
(4) The Mark II unit driven by a NEMA "C" motor requires 13.4 nameplate horsepower, i.e. 15 HP motor.

The horsepower requirements above are for perfectly counterbalanced conditions where the cyclic load factors are average for the units, loads, and prime movers noted.

2.845 Approximate horsepower formulas

Numerous prime mover horsepower formulas are used by various companies to approximate horsepower requirements. To show the wide divergence of approaches to determine prime mover horsepower requirements for beam pumping, some of these approaches are summarized here.

(Portion of API-11L3 tables reproduced)

	Rod No. 76		Pump depth 4500.			Production 150.				
Pump dia.	Stroke	SPM	PPRL	MPRL	Stress	PT	PRHP	CBE	Prod. Unanch.	WRF
1.06	30.	29.7	13781.	3530.	22930.	71.	9.6	8344.	135.1	7012.
1.06	36.	26.6	13269.	3550.	22078.	86.	9.8	8344.	136.5	7012.
1.06	42.	24.6	13189.	3582.	21945.	105.	10.3	8344.	137.8	7012.
1.06	48.	22.6	13160.	3399.	21896.	123.	10.5	8344.	138.7	7012.
1.06	54.	20.9	13108.	3255.	21811.	141.	10.6	8344.	139.5	7012.
1.06	64.	17.3	12673.	3590.	21086.	158.	9.7	8344.	141.4	7012.
1.06	74.	15.0	12535.	3908.	20856.	175.	9.6	8344.	142.5	7012.
1.06	86.	13.2	12490.	4145.	20782.	198.	9.8	8344.	143.4	7012.
1.06	100.	11.4	12254.	4356.	20389.	223.	9.5	8344.	144.2	7012.
1.06	120.	9.5	11982.	4735.	19936.	253.	9.4	8344.	145.1	7012.
1.06	144.	8.0	11792.	5094.	19620.	292.	9.6	8344.	145.9	7012.
1.06	168.	6.9	11724.	5088.	19507.	340.	9.4	8344.	146.5	7012.
1.25	24.	29.4	13094.	4543.	21787.	51.	7.7	8746.	132.4	7055.
1.25	30.	25.7	12892.	4385.	21451.	63.	7.9	8746.	134.7	7055.
1.25	36.	22.9	12752.	4258.	21219.	77.	7.9	8746.	136.2	7055.
1.25	42.	20.8	12767.	4060.	21243.	96.	8.1	8746.	137.5	7055.
1.25	48.	18.3	12646.	4214.	21042.	113.	7.8	8746.	138.8	7055.
1.25	54.	15.7	12392.	4491.	20618.	123.	7.2	8746.	140.4	7055.
1.25	64.	13.2	12266.	4825.	20410.	141.	7.0	8746.	142.0	7055.
1.25	74.	11.6	12162.	4932.	20236.	157.	7.0	8746.	142.9	7055.
1.25	86.	10.0	11963.	5230.	19905.	170.	7.1	8746.	143.9	7055.
1.25	100.	8.6	11804.	5513.	19640.	187.	7.3	8746.	144.7	7055.
1.50	24.	25.8	12931.	4992.	21516.	48.	6.5	9378.	117.8	7126.
1.50	30.	22.1	13069.	4865.	21745.	62.	6.7	9378.	122.4	7126.
1.50	36.	19.4	13021.	3003.	21665.	72.	6.6	9378.	125.6	7126.
1.50	42.	17.0	13014.	4143.	21655.	87.	6.6	9378.	128.3	7126.
1.50	48.	13.9	12776.	5210.	21258.	103.	5.9	9378.	132.4	7126.
1.50	54.	12.2	12749.	5210.	21213.	119.	5.8	9378.	134.6	7126.
1.50	64.	10.2	12657.	5271.	21060.	143.	5.8	9378.	137.1	7126.
1.50	74.	8.7	12497.	5636.	20793.	157.	5.8	9378.	139.0	7126.
1.75	30.	20.2	13829.	4166.	23010.	62.	6.1	10120.	116.6	7204.
1.75	36.	17.3	13805.	4360.	22970.	72.	6.1	10120.	121.0	7204.
1.75	42.	14.6	13801.	5325.	22963.	83.	6.0	10120.	125.4	7204.
1.75	48.	11.6	13655.	5538.	22720.	100.	5.3	10120.	130.3	7204.
1.75	54.	10.1	13643.	5625.	22700.	120.	5.3	10120.	132.9	7204.
2.00	42.	13.4	15155.	5677.	25216.	89.	5.3	10984.	111.3	7303.
2.00	48.	10.7	15008.	5727.	24972.	105.	5.1	10984.	119.2	7303.
2.25	48.	10.4	16723.	5863.	27825.	111.	5.0	11958.	102.1	7409.

(1) Correspondence with one major oil company engineer revealed that he uses the approach outlined in Continental EMSCO Bulletin A-267:[27]

$$HP = \frac{N \times D}{PMF} \qquad (2.56)$$

where:

HP = Prime mover horsepower
D = Well depth (to pump), ft
PMF = Constant for various types of prime movers
N = (K)(SPM)(S)
K = Plunger constant
SPM = Pumping speed, strokes per minute
S = Stroke length, in.

The PMF constants are as follows:

C-E engine	85,000
C-E Green Triangle electric motor	85,000
C-E Green Triangle convertible motor	65,000
Other make NEMA "C" motor	65,000
Other make NEMA "D" motor	75,000
Multi-cylinder engine	65,000

The engineer stated:

"The difference between C-E's formula and the formula that I used is in the value of the PMF constant. C-E recommends values of 65,000 to 85,000, but from a practical standpoint and from the experience we have had, values of 60,000 to 65,000 should be used for electrical motor design in the NEMA "D" design. The constant that I used is obviously more liberal from a design standpoint. This formula has obvious advantages over hydraulic horsepower or theoretical horsepower, etc., when converting from gas engines to electrical motors in that the motor only knows what the pumping unit is doing and what size plunger the pump has and does not really care how many barrels a day the well is making. A motor size based on hydraulic horsepower will almost always result in too small a motor. All other formulas that I have seen published resulted in too conservative a design.

"If one is specifying the motor size for an air

(Portion of Mark II design tables reproduced—after Ref. 23)

Rod No. 76 | Pump depth 4500 | Production 150

Pump dia.	Stroke	SPM	PPRL	MPRL	Stress	GRD	PT	CBM	ITE	PRHP	PMHPD	PMHPC	EI	WRF
1.06	64	17.2	12475	2868	20758	46	104	256	32.6	9.3	11.9	16.5	1.208E+01	7012
1.06	74	15.1	11844	2928	19708	41	110	320	34.2	9.1	11.7	16.2	1.189E+01	7012
1.06	86	13.1	11331	3212	18853	37	129	379	32.3	8.7	12.3	16.9	1.275E+01	7012
1.06	100	11.4	10902	3563	18140	33	151	531	31.0	8.4	12.0	16.6	1.388E+01	7012
1.06	120	9.5	10708	3852	17817	31	176	679	31.7	8.4	12.0	16.5	1.580E+01	7012
1.06	144	7.9	10490	4208	17455	28	216	856	30.7	8.3	13.3	18.3	1.888E+01	7012
1.06	168	6.8	10315	4553	17163	25	271	1055	28.0	8.2	14.5	20.0	2.281E+01	7012
1.25	64	13.1	11691	3842	19453	36	94	293	35.1	6.9	9.4	13.0	7.601E+00	7055
1.25	74	11.3	11403	4003	18973	34	116	350	32.7	6.8	9.3	12.9	8.919E+00	7055
1.25	86	9.8	11374	3983	18925	34	126	422	35.2	6.8	10.4	14.3	9.804E+00	7055
1.25	100	8.4	11270	4158	18752	32	158	562	32.4	6.8	10.4	14.3	1.220E+01	7055
1.50	64	9.9	12513	4652	20820	37	114	328	33.2	6.0	8.5	11.8	6.562E+00	7126
1.50	74	8.7	12517	4648	20827	38	133	388	32.9	6.0	9.5	13.1	1.001E+01	7126

Rod No. 87 | Pump depth 4500 | Production 150

Pump dia.	Stroke	SPM	PPRL	MPRL	Stress	GRD	PT	CBM	ITE	PRHP	PMHPD	PMHPC	EI	WRF
1.06	64	17.0	15972	3876	20347	44	126	322	31.3	10.6	14.2	19.6	2.141E+01	9319
1.06	74	14.9	15067	4129	19194	39	136	394	32.0	10.3	13.8	19.1	2.098E+01	9319
1.06	86	13.0	14445	4525	18401	34	155	469	31.3	10.0	13.6	18.7	2.224E+01	9319
1.06	100	11.2	13756	5059	17523	30	172	639	31.0	9.5	13.1	18.1	2.260E+01	9319
1.06	120	9.5	13495	5392	17191	27	201	807	31.9	9.7	14.7	20.3	2.628E+01	9319
1.06	144	7.9	13200	5840	16816	25	245	1013	31.3	9.6	14.7	20.3	3.124E+01	9319
1.06	168	6.8	13000	6249	16560	22	307	1237	28.6	9.5	16.1	22.2	3.773E+01	9319
1.25	64	12.9	14479	5207	18444	32	113	361	33.3	7.7	10.3	14.2	1.258E+01	9355
1.25	74	11.1	14102	5543	17964	29	131	426	32.1	7.4	11.0	15.1	1.364E+01	9355
1.25	86	9.6	13983	5765	17813	28	144	515	33.7	7.4	10.9	15.1	1.479E+01	9355
1.25	100	8.3	13843	5997	17635	27	180	690	31.3	7.4	12.3	16.9	1.853E+01	9355
1.50	64	9.5	15069	6284	19196	31	126	394	32.5	6.2	9.7	13.4	1.184E+01	9406

Rod No. 109 | Pump depth 4500 | Production 150

Pump dia.	Stroke	SPM	PPRL	MPRL	Stress	GRD	PT	CBM	ITE	PRHP	PMHPD	PMHPC	EI	WRF
1.06	64	16.8	23790	6355	19389	39	169	473	30.6	13.8	17.6	24.3	5.574E+01	14805
1.06	74	14.7	22555	7147	18382	34	190	577	29.6	13.1	18.2	25.2	5.625E+01	14805
1.06	86	12.8	21466	8006	17495	29	203	694	30.5	12.5	17.6	24.3	5.448E+01	14805
1.06	100	11.2	20361	8769	16594	25	219	898	31.3	12.2	17.3	23.8	5.426E+01	14805
1.06	120	9.4	19947	9313	16257	23	260	1118	32.0	12.4	19.4	26.7	6.406E+01	14805
1.06	144	7.9	19546	9399	15930	20	317	1384	31.6	12.6	19.6	27.1	7.810E+01	14805
1.06	168	6.8	19290	10456	15721	19	394	1671	29.1	12.4	21.5	29.7	9.397E+01	14805
1.25	64	12.7	21089	9174	17167	26	150	521	30.2	9.1	12.7	17.5	2.869E+01	14832
1.25	74	11.1	20442	9685	16660	23	167	616	30.2	8.9	12.5	17.2	3.034E+01	14832
1.25	86	9.6	20207	10045	16469	22	188	737	31.1	8.9	13.9	19.2	3.366E+01	14832
1.25	100	8.2	19921	10474	16235	20	232	948	29.2	8.9	13.9	19.1	4.098E+01	14832
1.50	64	9.1	21116	10393	17210	23	154	563	31.2	6.9	10.5	14.5	2.266E+01	14871

balanced pumping unit, the motor size should be increased by the additional horsepower used to drive the air compressor. Otherwise, my experience indicates that the formula would be good for any sucker rod installation regardless of pump depth or tubing size.

"A deviated hole would definitely have some effect on the sizing as well as ultra low gravity production and wells with serious waxing tendencies."

(2) Another major oil company, in its sucker rod pumping manual, uses the following approach.[22]

1. Conventional unit

$$EBHP = PRHP \times 1.25 \quad (2.57)$$

where:

EBHP = engine brake horsepower
PRHP = polished rod horsepower determined from the API RP 11L design procedure (to be discussed in a later section).

2. Air balanced unit

$$EBHP = PRHP \times 1.25 \quad (2.58)$$

where:

$$PRHP = HP_h + HP_f$$
$$= (PD \times L \times 7.4 \times 10^{-6}) + (6.25 \times 10^{-7} \times N \times S \times W_r)$$
$$= (\text{Equation 2.45}) + (\text{Equation 2.52})$$

for

$$G = 1.0$$

3— Mark II unit

$$EBHP = PRHP \quad (2.59)$$

where PRHP is determined in the same manner as for the air balanced unit.

For normal slip (NEMA "C") motors

$$HP_{np} = \frac{EBHP}{0.60} \quad (2.60)$$

For high slip (NEMA "D") motors

$$HP_{np} = \frac{EBHP}{0.80} \quad (2.61)$$

(3) One pumping unit manufacturer suggests that horsepower requirements can be approximated as follows:

Conventional unit

$$HP = \frac{D \times B/D}{56{,}000}$$ for slow speed engines and high slip motors (2.62)

$$HP = \frac{D \times B/D}{45{,}000}$$ for multi-cylinder engines and normal slip motors (2.63)

where:

D = Pump depth in feet
BPD = Barrels fluid per day based upon 100% volumetric efficiency

Air Balanced Unit
(Same formulas as for conventional units)

Mark II unit

$$HP = \frac{D \times B/D}{56{,}000} \times 0.8$$ for slow speed engines and high slip motors (2.64)

$$HP = \frac{D \times B/D}{45{,}000} \times 0.8$$ for multi-cylinder engines and normal slip motors (2.65)

Although there are other approaches available, these indicate the state of the art.

CLASS PROBLEM

For the data given in Example #11, determine prime mover horsepower by the approximation formulas given above and compare the results to that of Example #11. What conclusions can be drawn?

2.85 Speed reduction and engine sheave size

The relationship for determining engine sheave size for a given unit sheave size (connected by means of V-belts) is as follows:

$$d_e = \frac{N_u d_u}{N_e} \quad (2.66)$$

where:

d_e = diameter of engine sheave, inches
d_u = diameter of unit sheave, inches
N_e = speed of engine sheave, rpm
N_u = speed of unit sheave, rpm

Now, let Z equal the speed ratio of the speed reducer.

Then,

$$Z = N_u/N \quad (2.67)$$

where:

N = pumping speed, spm

Then,

$$d_e = \frac{N Z d_u}{N_e} \quad (2.68)$$

CLASS PROBLEM

A pumping unit, pumping at 22.6 spm, has a gear ratio of 31.297 and the unit sheave size is 18 in. The prime mover has a rated speed of 520 rpm for 25 BHP. Determine the engine sheave size required.

2.86 API unit ratings

Before attempting the design and selection of a pumping installation, it is advisable to discuss the API ratings of pumping units and the meaning of manufacturers' code designations.

For example, in the case of a pumping unit Model C-228D-200-74, Figure 2.25 explains the meaning of this designation.

The first letter C refers to a conventional unit geometry. The next three digits, 228, refer to the API peak torque rating of the gear reducer in thousands of in.-lb; in this example the torque rating would be 228,000 in.-

EXPLANATION OF PUMPING UNIT DESIGNATIONS

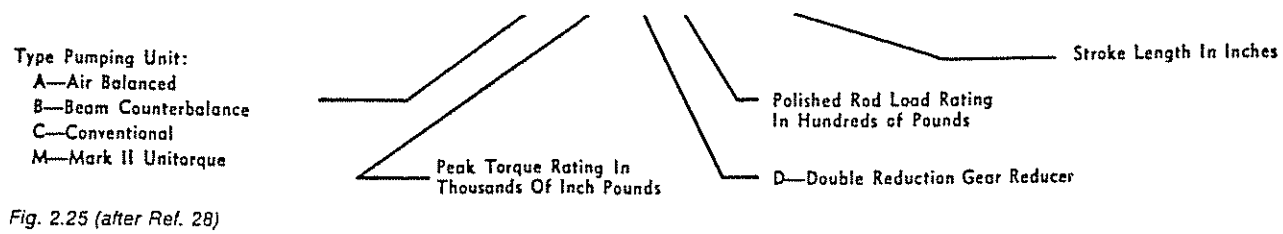


Fig. 2.25 (after Ref. 28)

lb. The letter D means that the gear reducer is of the double reduction type. The next three digits, 200, refer to the maximum structure capacity (peak polished rod load) in hundreds of pounds—in this case, 20,000 lb PPRL.

The last two digits, 74, refer to the maximum stroke length available, in inches, with that particular unit. In this case, a stroke length of 74 in. can be obtained. For this unit, according to manufacturer's literature, the actual stroke lengths available are 74, 64, and 54 in. One would have to refer to manufacturer's literature to determine the maximum counterbalance effect to be obtained with a given combination of available counterweights. Other unit dimensions and specifications for this unit will be given in the manufacturer's literature.

As another example, consider the Lufkin Mark II unit M-1280D-427-216. This unit has a maximum gear reducer torque rating of 1,280,000 in.-lb, a polished rod load capacity of 42,700 lb, and a maximum stroke length available of 216 in. The actual stroke lengths which can be obtained with this unit are 216, 192, and 167 in.

Table 2.14, reproduced from API Std. 11E, shows API

Table 2.14 (After Ref. 20)
PUMPING UNIT SIZE RATINGS

1	2	3	4
Pumping unit size	Reducer rating, in.-lb	Structure capacity, lb	Max. stroke length, in.
6.4— 32— 16	6,400	3,200	16
6.4— 21— 24	6,400	2,100	24
10— 32— 24	10,000	3,200	24
10— 40— 20	10,000	4,000	20
16— 27— 30	16,000	2,700	30
16— 53— 30	16,000	5,300	30
25— 53— 30	25,000	5,300	30
25— 56— 36	25,000	5,600	36
25— 67— 36	25,000	6,700	36
40— 89— 36	40,000	8,900	36
40— 76— 42	40,000	7,600	42
40— 89— 42	40,000	8,900	42
40— 76— 48	40,000	7,600	48
57— 76— 42	57,000	7,600	42
57— 89— 42	57,000	8,900	42
57— 95— 48	57,000	9,500	48
57—109— 48	57,000	10,900	48
57— 76— 54	57,000	7,600	54
80—109— 48	80,000	10,900	48
80—133— 48	80,000	13,300	48
80—119— 54	80,000	11,900	54
80—133— 54	80,000	13,300	54
80—119— 64	80,000	11,900	64
114—133— 54	114,000	13,300	54
114—143— 64	114,000	14,300	64
114—173— 64	114,000	17,300	64
114—143— 74	114,000	14,300	74
114—119— 86	114,000	11,900	86
160—173— 64	160,000	17,300	64
160—143— 74	160,000	14,300	74
160—173— 74	160,000	17,300	74
160—200— 74	160,000	20,000	74
160—173— 86	160,000	17,300	86
228—173— 74	228,000	17,300	74
228—200— 74	228,000	20,000	74
228—213— 86	228,000	21,300	86
228—246— 86	228,000	24,600	86
228—173—100	228,000	17,300	100
228—213—120	228,000	21,300	120
320—213— 86	320,000	21,300	86
320—256—100	320,000	25,600	100
320—305—100	320,000	30,500	100
320—213—120	320,000	21,300	120
320—256—120	320,000	25,600	120
320—256—144	320,000	25,600	144
456—256—120	456,000	25,600	120
456—305—120	456,000	30,500	120
456—365—120	456,000	36,500	120
456—256—144	456,000	25,600	144
456—305—144	456,000	30,500	144
456—305—168	456,000	30,500	168
640—305—120	640,000	30,500	120
640—256—144	640,000	25,600	144
640—305—144	640,000	30,500	144
640—365—144	640,000	36,500	144
640—305—168	640,000	30,500	168
640—305—192	640,000	30,500	192
912—427—144	912,000	42,700	144
912—305—168	912,000	30,500	168
912—365—168	912,000	36,500	168
912—305—192	912,000	30,500	192
912—427—192	912,000	42,700	192
912—470—240	912,000	47,000	240
912—427—216	912,000	42,700	216
1280—427—168	1,280,000	42,700	168
1280—427—192	1,280,000	42,700	192
1280—427—216	1,280,000	42,700	216
1280—470—240	1,280,000	47,000	240
1280—470—300	1,280,000	47,000	300
1824—427—192	1,824,000	42,700	192
1824—427—216	1,824,000	42,700	216
1824—470—240	1,824,000	47,000	240
1824—470—300	1,824,000	47,000	300
2560—470—240	2,560,000	47,000	240
2560—470—300	2,560,000	47,000	300
3648—470—240	3,648,000	47,000	240
3648—470—300	3,648,000	47,000	300

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pumping unit size ratings. These are the most common, but others do exist.

2.9 API RECOMMENDED DESIGN PROCEDURE

The API recommended design procedure for a conventional unit sucker rod pumping system is detailed in API RP 11L.[3] The method is based upon correlations of research test data, and the results are presented in terms of nondimensional parameters which may be determined from a series of curves. These curves are reproduced here for convenience.

The design procedure is a trial and error method. Three steps are generally required in the procedure:[3]

(1) A preliminary selection of components for the installation must be made.
(2) The operating characteristics of the preliminary selection are calculated by use of the formulas, tables, and figures in the API RP 11L.
(3) The calculated pump displacement and loads are compared with the volumes, load ratings, stresses, and other limitations of the preliminary selection.

The minimum amount of information which must either be known or assumed is as follows:

(1) Fluid level—D, the net lift in ft
(2) Pump depth—L, ft
(3) Pumping speed—N, strokes per minute
(4) Length of surface stroke—S, in.
(5) Pump plunger diameter—D_p, in.
(6) Specific gravity of the fluid—G
(7) The nominal tubing diameter and whether it is anchored or unanchored
(8) Sucker rod size and design

With these known data, the following design factors can be determined:

(1) Plunger stroke—S_P, in.
(2) Pump displacement—PD, B/D
(3) Peak Polished Rod Load—PPRL, lb
(4) Minimum Polished Rod Load—MPRL, lb
(5) Peak Crank Torque—PT, in.-lb
(6) Polished Rod Horsepower—PRHP
(7) Counterweight required—CBE, lb

These design factors are determined from the following equations.

$$S_p = [(S_p/S) \times S] - [F_o \times 1/k_t] \quad (2.69)$$

When the tubing is anchored, $1/k_t$ is zero.

The term F_o is the gross plunger load, and its significance is illustrated on Figure 2.26. The value of S_p/S is determined from Figure 2.27.

$$F_o = 0.340 \times G \times D_p^2 \times D \quad (2.70)$$

$$1/k_t = E_t \times L \quad (2.71)$$

where E_t is the elastic constant of the tubing and may be determined from Table 2.4.

The pump displacement, PD, is given by:

$$PD = 0.1166 \times S_p \times N \times D_p^2 \quad (2.72)$$

Peak polished rod load, PPRL, is given by:

$$PPRL = W_{rf} + [(F_1/Sk_r) \times Sk_r] \quad (2.73)$$

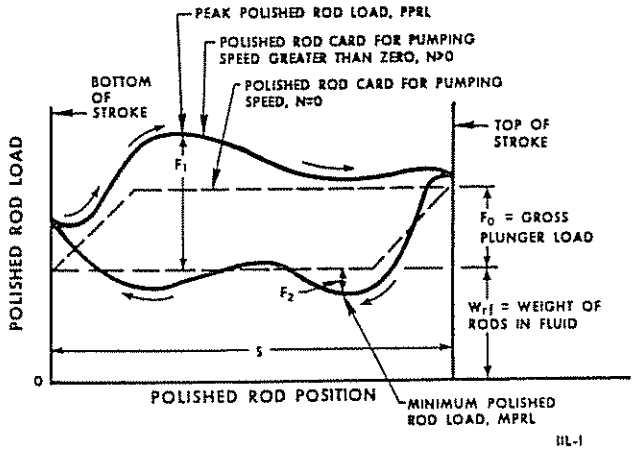


Fig. 2.26 Basic Dynagraph Card (after Ref. 3)

where W_{rf} is the weight of the rods in the fluid and is determined by:

$$W_{rf} = W_r \times L \times (1 - 0.128G) \quad (2.74)$$

The weight of the rods, W_r, in lb/ft, can be determined from Table 2.7.

The non-dimensional parameter (F_1/Sk_r) is determined from Figure 2.28.

The value of $1/k_r$) is given by:

$$1/k_r = E_r \times L \quad (2.75)$$

where the elastic constant of the rods, E_r, can be obtained from Table 2.7.

Minimum polished rod load, MPRL, is given by:

$$MPRL = W_{rf} - [(F_2/Sk_r) \times Sk_r] \quad (2.76)$$

The parameter (F_2/Sk_r) is determined from Figure 2.29.

Peak torque, PT, is given by:

$$PT = (2T/S^2k_r) \times Sk_r \times T_a \times S/2 \quad (2.77)$$

where T_a is an adjustment for peak torque for values of (W_{rf}/Sk_r) other than 0.3. The adjustment is obtained from Figure 2.30. The value of $(2T/S^2k_r)$ is obtained from Figure 2.31.

Polished rod horsepower, PRHP, is given by:

$$PRHP = (F_3/Sk_r) \times Sk_r \times S \times N \times 2.53 \times 10^{-6} \quad (2.78)$$

The parameter (F_3/Sk_r) is obtained from Figure 2.32.

Finally, the counterweight required is determined by:

$$CBE = 1.06\ (W_{rf} + \tfrac{1}{2}F_o) \quad (2.79)$$

In the previous figures, the value of the term (N/N_o) is determined by:

$$N/N_o = NL/245{,}000 \quad (2.80)$$

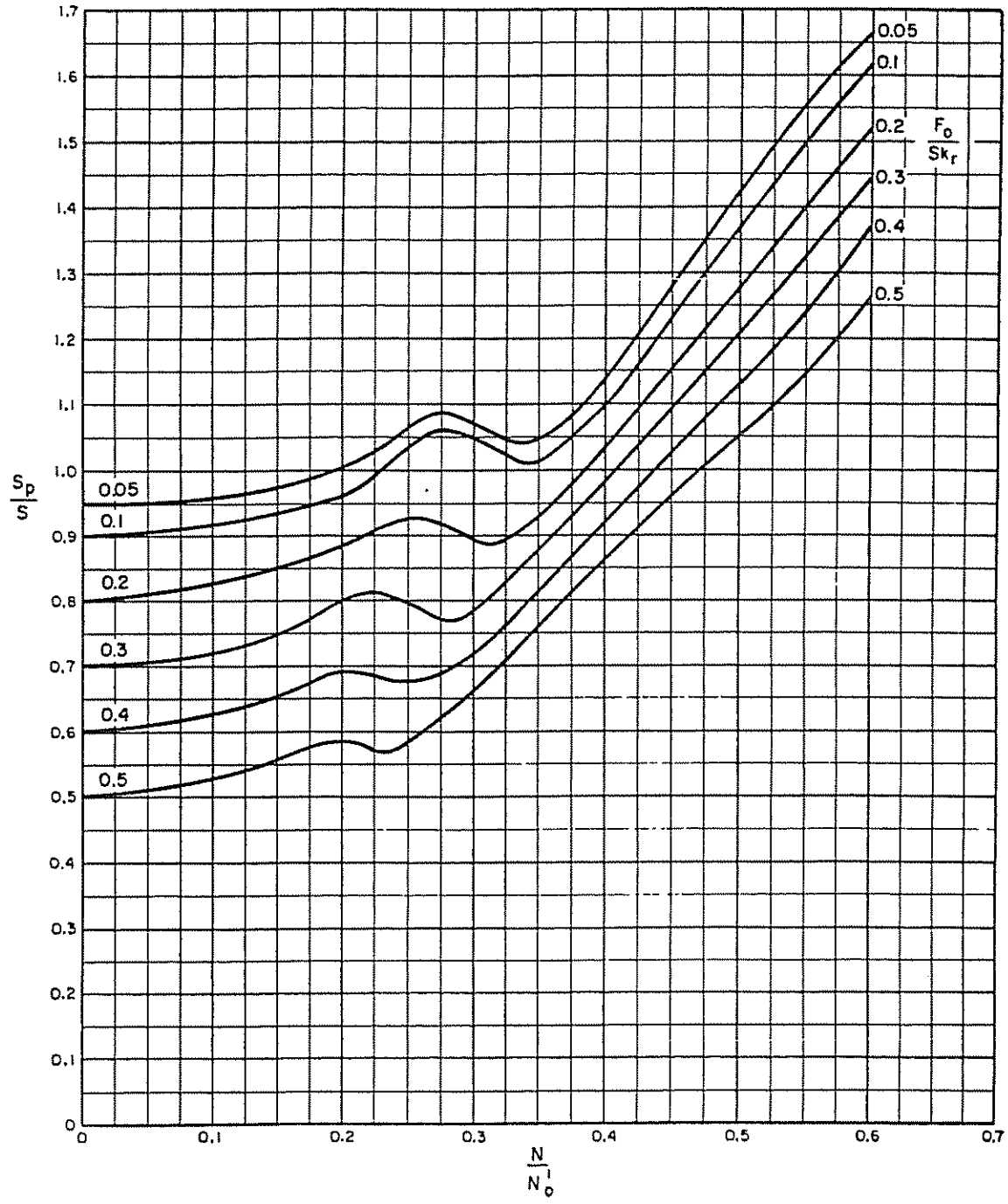


Fig. 2.27 (after Ref. 3)

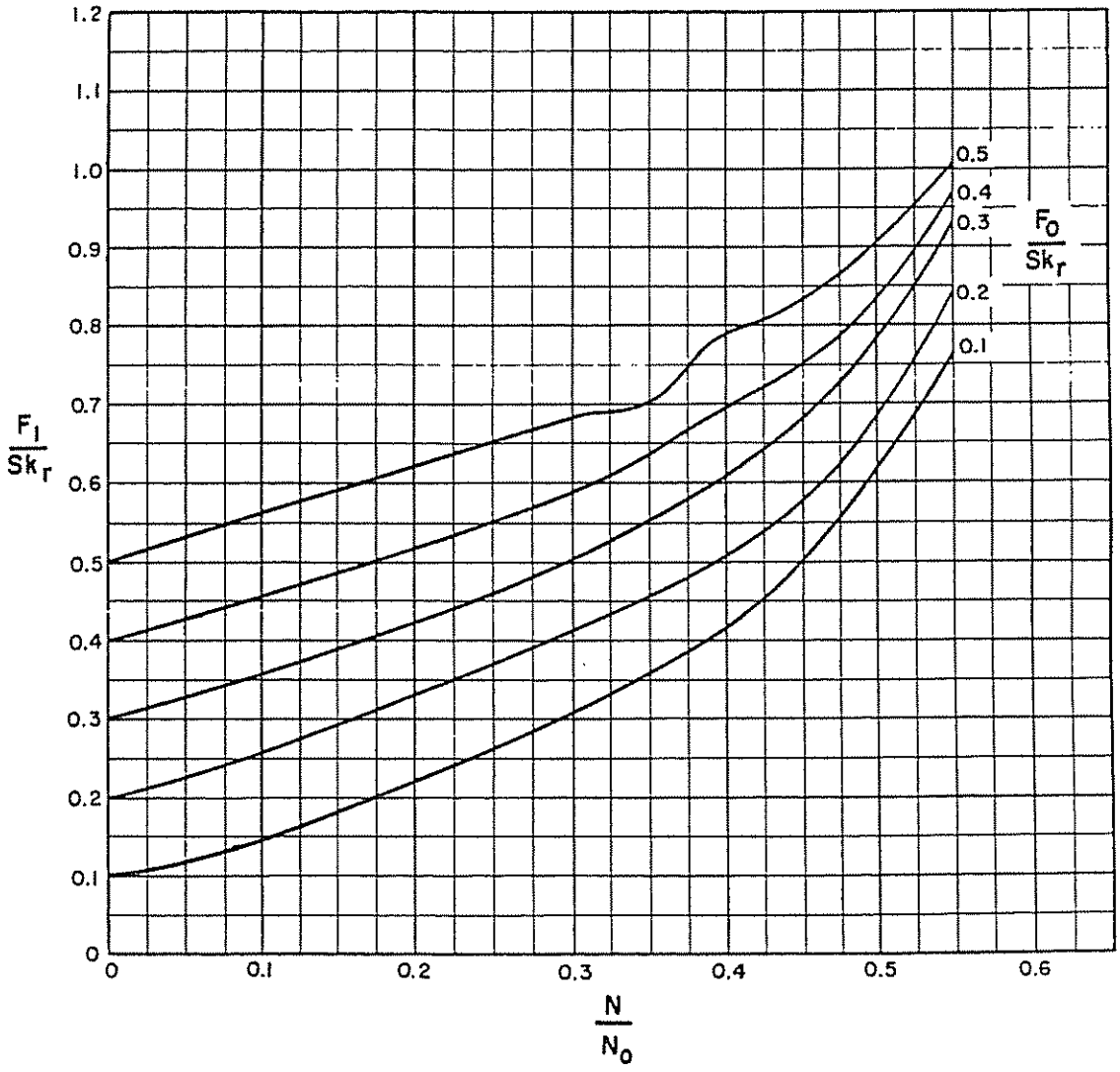
The term (N/N_o') is determined from:

$$N/N_o' = (N/N_o)/F_c \quad (2.81)$$

where F_c (frequency factor) is a constant of proportionality which depends upon the rod design. The dimensionless pumping speed (N/N_o') is an important index of the behavior of the rod string. The frequency factor, F_c, can be obtained from Table 2.7. For those rod strings not given in Table 2.7, Figures 2.33—2.38 can be used to determine the frequency factor from the relationship:

$$F_c = 1.0 + \% \text{ from chart} \quad (2.82)$$

To illustrate the API procedure, an example design calculation is given on Table 2.15. A blank example design sheet is also included as Table 2.16 to illustrate the step-by-step procedure (pp. 65-66).



$\frac{F_1}{Sk_r}$, PEAK POLISHED ROD LOAD

Fig. 2.28 (after Ref. 3)

EXAMPLE PROBLEM #12 (after ref. 29)

Assume:

(1) $E_v = 70\%$
(2) $q_0 = 166$ bf/d
(3) Pump Intake set at 5050 ft
(4) 1.5 in. pump
(5) S.G. of fluid = 0.976
(6) Tubing = 2⅜-in. O.D.

Determine:

(1) PD and associated speed, N, in spm
(2) Select a rod string design.

Solution:

(1) $E_v = q/V$ where $V = PD$
so $PD = q/E_v = 166/0.70 = 237$ bf/d
from API, $PD = 0.1166\ S_p\ ND_p^2$
(*Note:* Keep $S_p/S \geq 0.80$, or $S_p = 0.8 \times S$)
so $PD = 0.1166 \times 0.8 \times SND_p^2$
or $PD = 0.09328\ SND_p^2$

Now,

$$PD = 237 \text{ bf/d}$$

so,

$$SND_p^2 = \frac{PD}{0.09328} = \frac{237}{0.09328} = 2541$$

or,

$$SN = \frac{2541}{D_p^2} = \frac{2541}{(1.5)^2} = 1130$$

Thus,

$$N = \frac{1130}{S}$$

Now assume values for S and calculate N:
(Note: Values of S selected from "API Spec. for Pumping Units," API Std. 11E, pp 6 & 7, Table 2.)

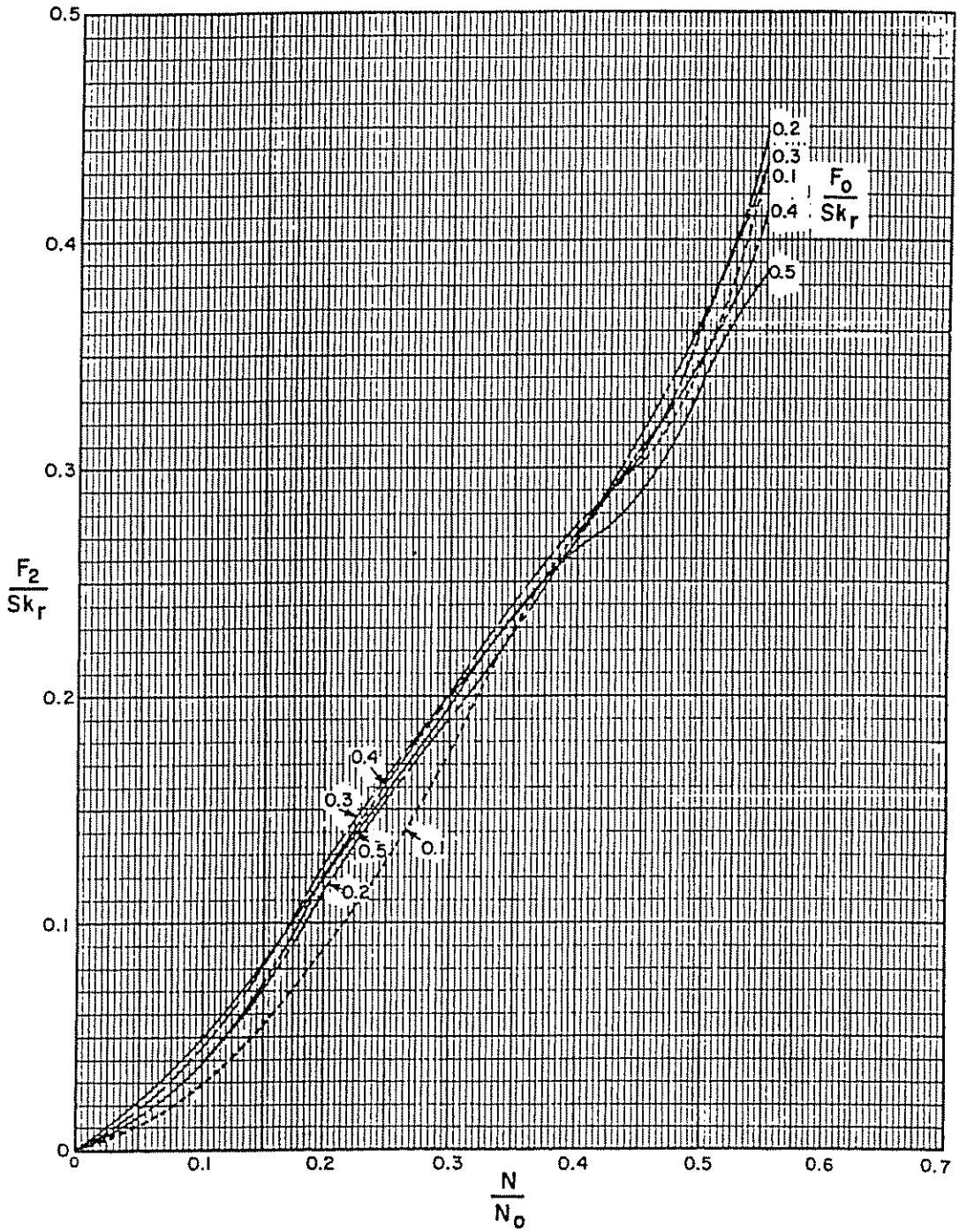
S	N	$C = \alpha$	N/N_o*
42	26.9	0.432	0.555
48	23.6	0.376	0.486
54	20.9	0.337	0.431
64	17.7	0.285	0.365
74	15.3	0.245	0.315
86	13.15	0.210	0.271

$$*N/N_o = \frac{NL}{245{,}000} = \frac{N \times 5050}{245{,}000} = 0.0206\ N$$

Note: Acceleration factor ($C = \alpha$) should not exceed 0.3 nor be less than 0.225. Select the S and N that have the smallest S and an acceleration factor less than 0.3.

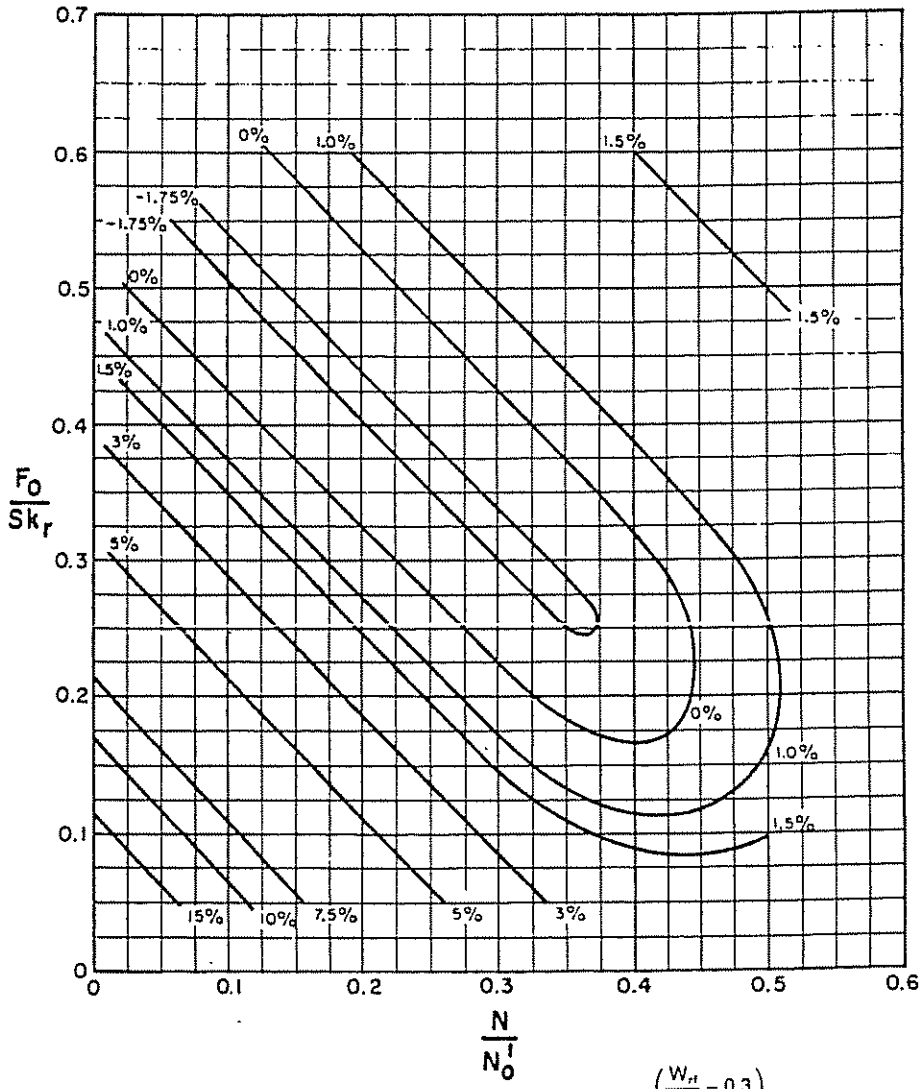
From the Table above, S = 64 in. at N = 17.7 spm.

Note that $N/N_o > 0.35$ ($N/N_o = 0.365$). Thus a tapered rod string will probably be selected. This will result in the dimensionless pumping speed, N/N_o', decreasing to less than 0.35 since



$\frac{F_2}{Sk_r}$, MINIMUM POLISHED ROD LOAD

Fig. 2.29 (after Ref. 3)



TO USE: MULTIPLY % INDICATED ON CURVE BY $\frac{\left(\frac{W_{rf}}{Sk_r} - 0.3\right)}{0.1}$

FOR EXAMPLE: $\frac{W_{rf}}{Sk_r} = 0.600$

$\frac{N}{N_o'} = 0.200 \quad \frac{F_o}{Sk_r} = 0.188$

ADJUSTMENT = 3% FOR EACH 0.1 INCREASE IN $\frac{W_{rf}}{Sk_r}$ ABOVE .03

TOTAL ADJUSTMENT = 3 × 3% = 9%

$T_a = 1.00 + 0.09 = 1.09$

NOTE: IF $\frac{W_{rf}}{Sk_r}$ IS LESS THAN 0.3 ADJUSTMENT BECOMES NEGATIVE

T_a, ADJUSTMENT FOR PEAK TORQUE FOR VALUES OF $\frac{W_{rf}}{Sk_r}$ OTHER THAN 0.3

Fig. 2.30 (after Ref. 3)

N/N_o must be divided by the frequency factor F_c, which is larger than 1.0.

so,

$N/N_o' = N/N_o \div F_c = 0.33.$

(2) Select rod string

Study API curves of S_p/S vs N/N_o'.

To maintain S_p/S of 0.80 with an N/N_o' of 0.33, the value of F_o/Sk_r, the dimensionless rod load, must be about 0.37.

Note: S.G. = 0.976, so assume G = 1.0 since some oil will bypass water in the tubing.

From API: $F_o = 0.340 \times G \times D_p^2 \times D$

Assume D = 5050′ (assume lift depth, D, equals pump setting depth, L, for design purposes).

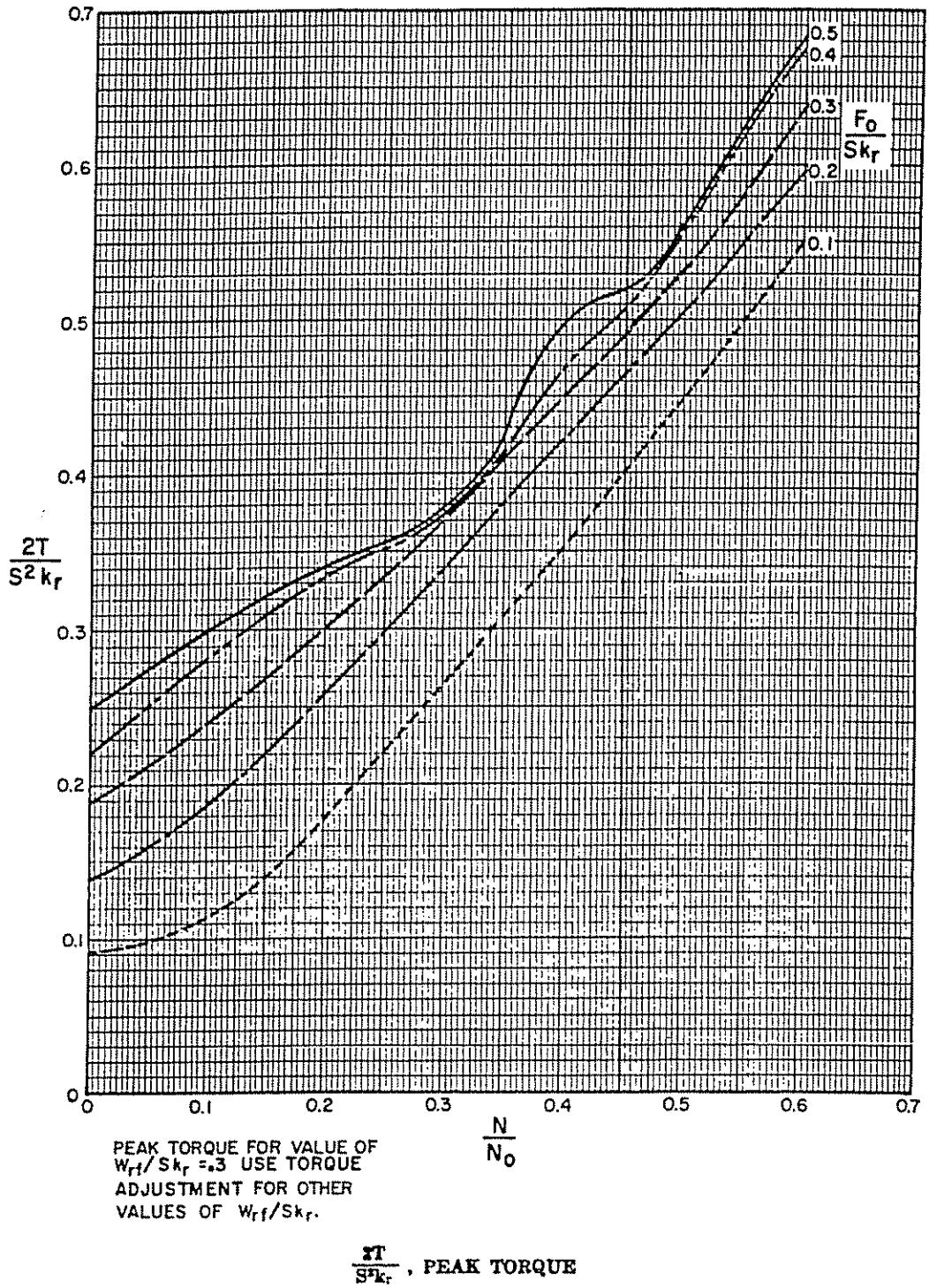


Fig. 2.31 (after Ref. 3)

$F_o = 0.340 \times 1.0 \times (1.5)^2 \times 5050 = 3860$ lb

$F_o/Sk_r = 0.37 = 3860/Sk_r$

$Sk_r = 3860/0.37 = 10{,}170$ lb

$k_r = 10{,}170/S = 10{,}170/64 = 159$

$1/k_r = E_r L$ (from API)

$1/159 = E_r \times 5050$

$E_r = 1/(159 \times 5050) = 1.246 \times 10^{-6} \frac{\text{in}}{\text{lb-ft}}$

From API table of rod & pump data, select a rod string for a 1.5-in. pump that will stretch less than 1.246×10^{-6} in/lb-ft.

This is a No. 64 rod string, or a No. 65, or a No. 75.
Use smallest possible, so delete No. 75.
Also, ½-in. rods are easily damaged and should not be used in 2⅜-in. tubing, so delete No. 64.
Thus, use No. 65 rod string.

CLASS PROBLEM

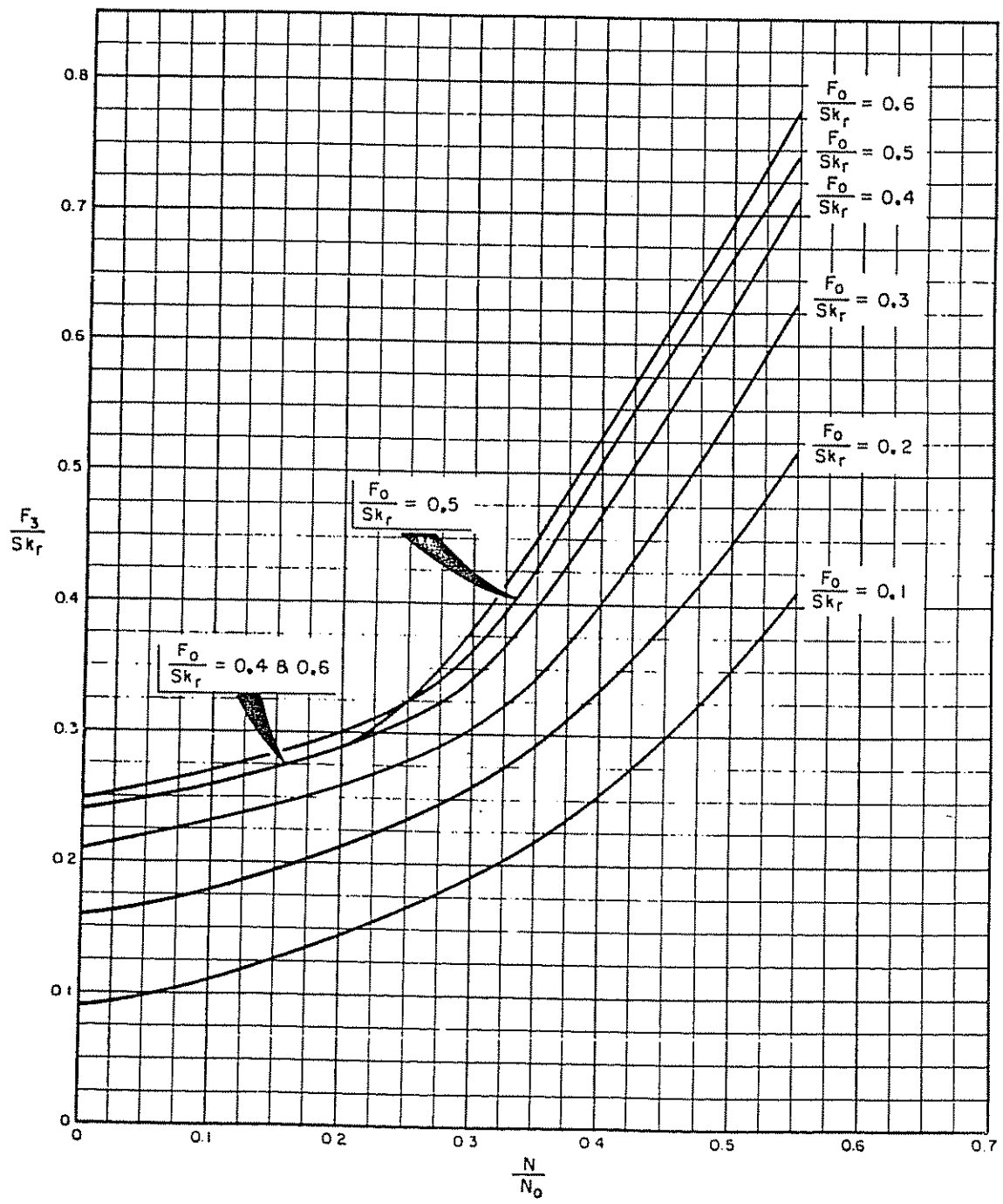
Perform the API Design Calculations (Table 2.16) for the following (p. 66).

Fluid level = 4000 ft
Pump depth = 4000 ft
Pumping speed = 15 spm
Stroke length = 54 in.
Plunger diameter = 1½ in.
Fluid specific gravity = 0.85
API rod number = 65
Tubing size (unanchored) = 2 in.
Required pump displacement = 150 B/D
Maximum allowable rod stress = 22,000 psi

What do you conclude?

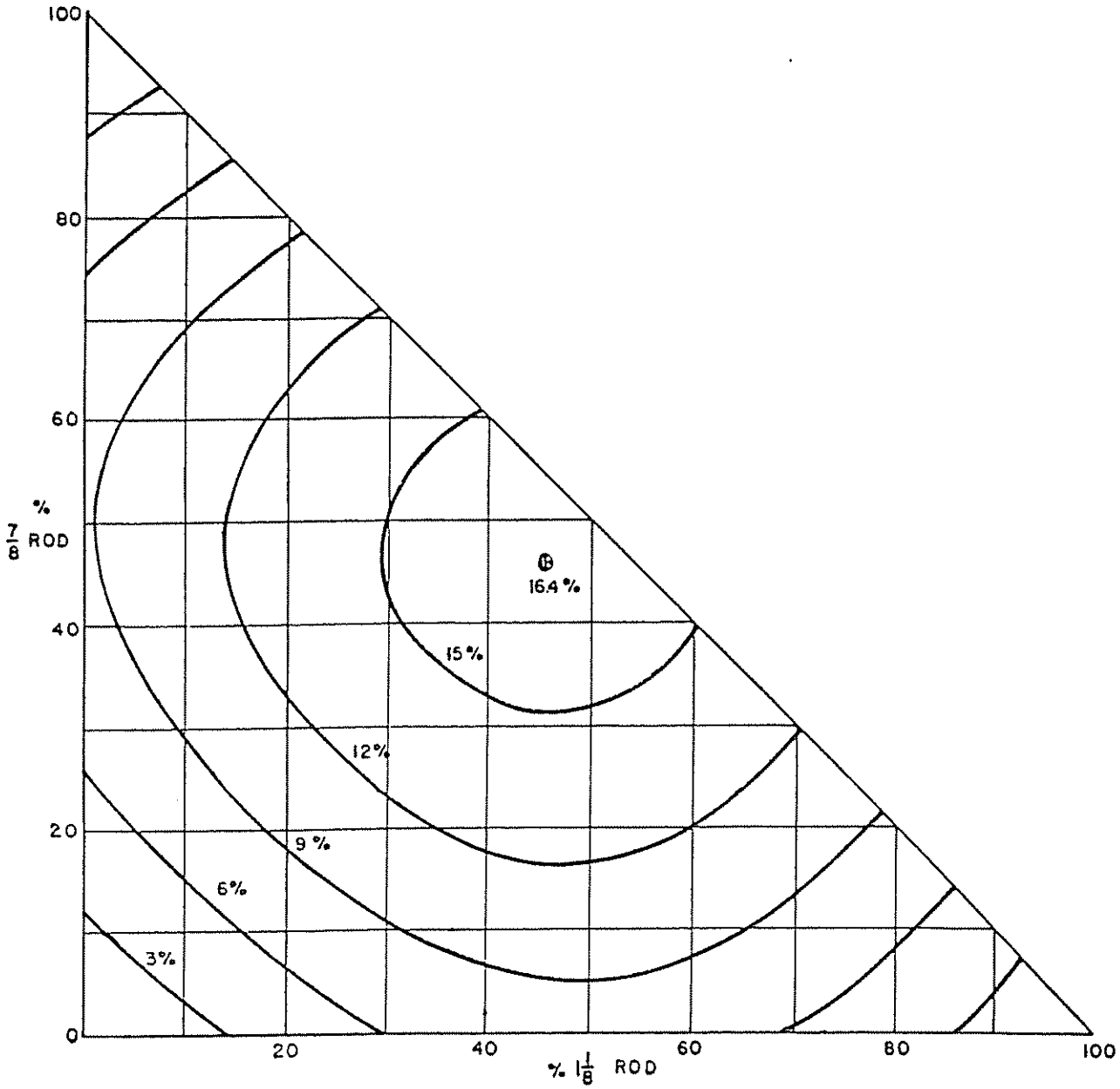
CLASS PROBLEM

For the class problem above, calculate for both API Grade C and Grade D rods the following:
(1) Maximum allowable stress, S_A
(2) Minimum stress, S_{min}
(3) Maximum allowable stress range, ΔS_A



$\frac{F_s}{Sk_r}$, POLISHED ROD HORSE POWER

Fig. 2.32 (after Ref. 3)



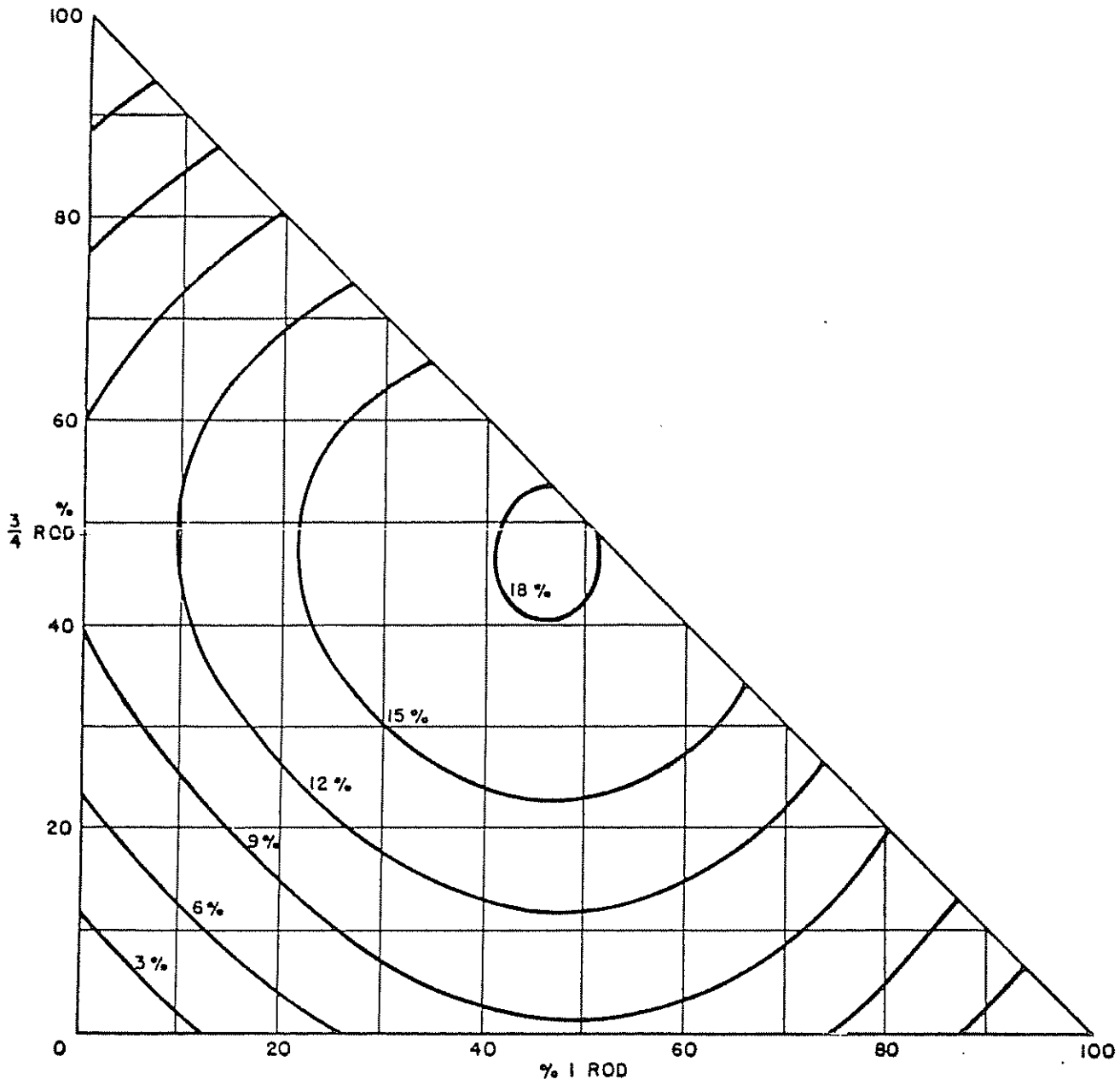
PERCENTAGE INCREASE IN FUNDAMENTAL FREQUENCY
1-1/8, 1, AND 7/8-INCH THREE-WAY TAPER STRING

Fig. 2.33 (after Ref. 3)

2.10 DYNAMOMETERS AND DYNAGRAPHS

The dynamometer is a recognized tool for measuring polished rod load, peak load, peak torque, and horsepower requirements in conjunction with the design and selection of a pumping unit.[22] The dynamometer can also be an effective aid as a trouble-shooting instrument to evaluate pumping well problems prior to workover. The proper use of dynamometers and the correct interpretation of data obtained from them can help assure proper installation of new pumping units and can simplify the location of sources of trouble in older units. The proper interpretation of such "dynamometer cards" is essential to any well study.

A detailed discussion of dynamometers and the recorded data (dynamometer cards) is beyond the scope



PERCENTAGE INCREASE IN FUNDAMENTAL FREQUENCY
1, 7/8, AND 3/4-INCH THREE-WAY TAPER STRING

Fig. 2.34 (after Ref. 3)

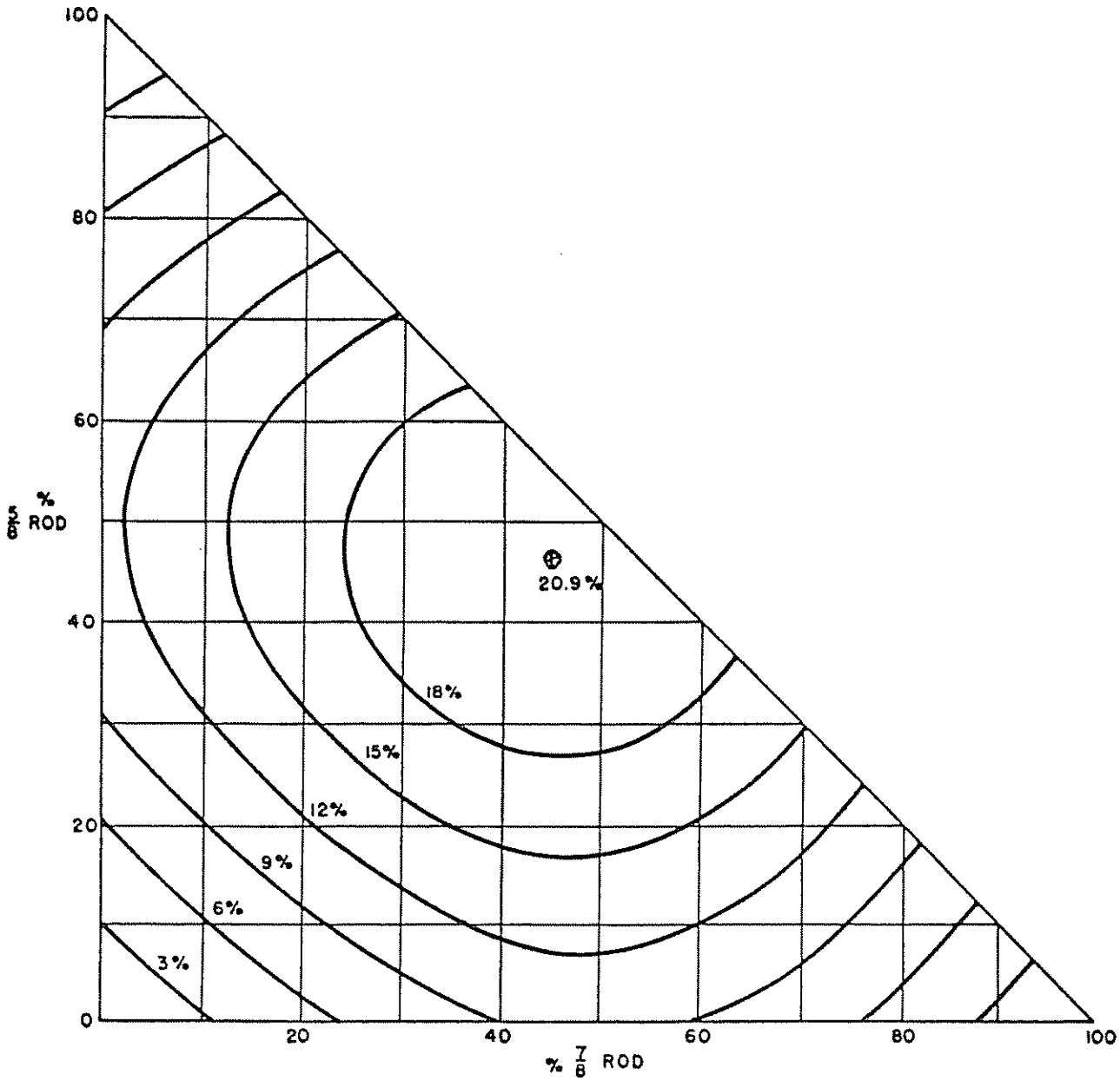
of this text. Only the basic principles of operation and interpretation, as well as factors which influence the shape of dynamometer cards, will be discussed.

The most commonly used dynamometers today are:
(1) The Johnson-Fagg dynamometer
(2) The Leutert LL-57 dynamometer
(3) The Delta-X Corporation DYNALOG DXD-01
(4) The Delta II dynamometer

Consideration will be given only to the records obtained (the dynamometer cards or dynagraphs), not to the operational details of these instruments.

2.101 The dynamometer card (dynagraph)

The dynamometer card (load-displacement diagram) is a continuous record of the resultant of all forces acting



PERCENTAGE INCREASE IN FUNDAMENTAL FREQUENCY
7/8, 3/4, AND 5/8-INCH THREE-WAY TAPER STRING

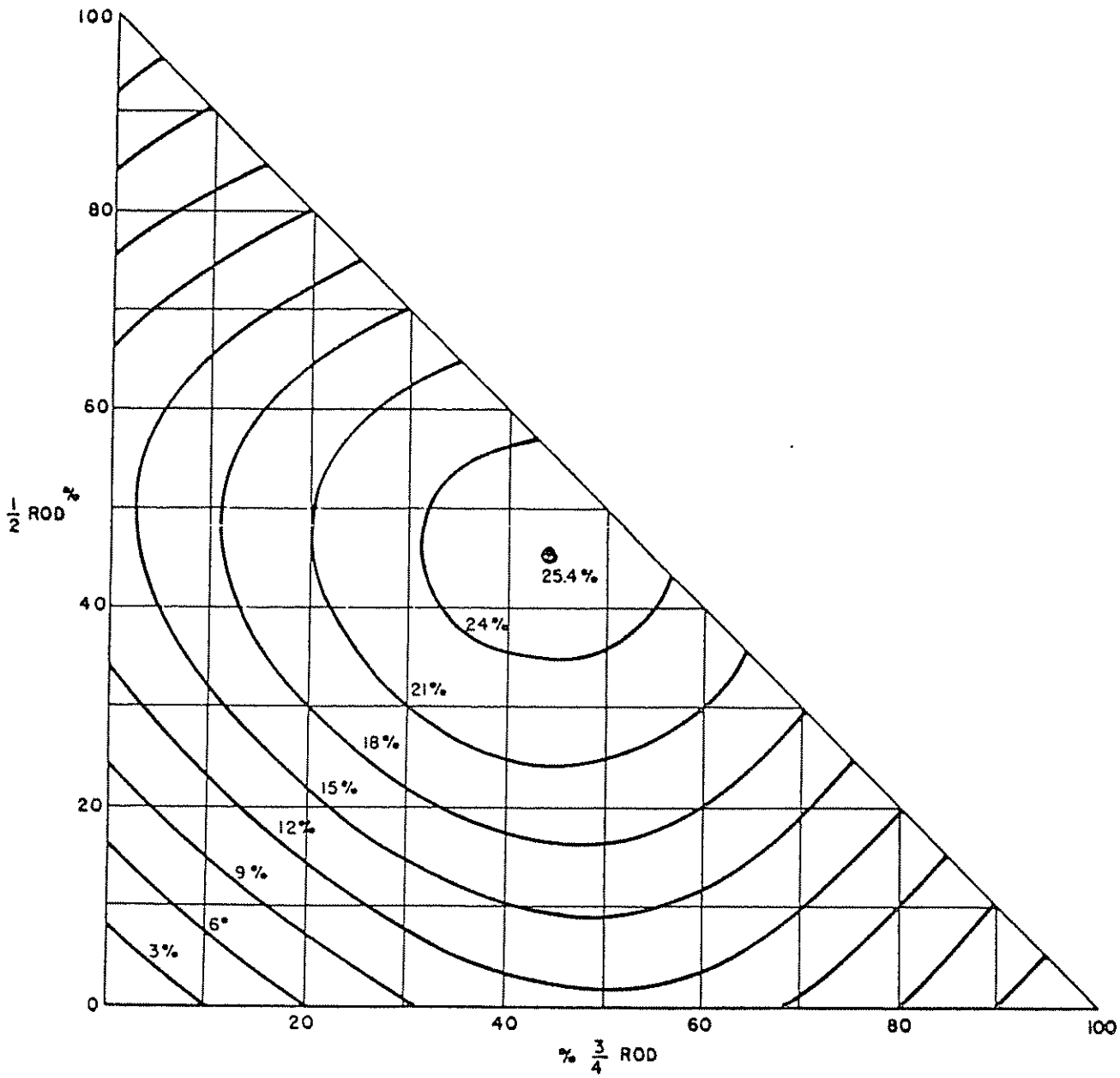
Fig. 2.35 (after Ref. 3)

on the polished rod at any instant during the pumping cycle and is recorded with respect to polished rod position. The composite record can be analyzed to determine loads in both directions, i.e. through the rod string to the pump and through the pumping unit to the prime mover. This record is the basis for stress analysis of the pumping unit and sucker rod string. The recorded load diagram is given with the rod position (stroke) on the abscissa and the loads on the ordinate.

The chief value of the polished rod dynamometer is that it furnishes important information in three major areas: (1) loading of the surface equipment from the polished rod to the prime mover, (2) rod loading, and (3) downhole pump performance.

Since the polished rod dynamometer provides a continuous plot of polished rod force vs. displacement, the following information is developed:

(1) Peak and minimum pumping unit structural loads
(2) Torsional load on the speed reducer and prime



PERCENTAGE INCREASE IN FUNDAMENTAL FREQUENCY
3/4, 5/8, AND 1/2-INCH THREE-WAY TAPER STRING

Fig. 2.36 (after Ref. 3)

mover, when the unit's torque factors are known and properly applied
(3) Work done by the polished rod against the elevation of the fluid and against friction
(4) Proper counterbalance
(5) Peak and minimum rod loads—rod stress, and load range
(6) Number of rod load fluctuations per crank cycle

Not only is the magnitude of the polished rod load pattern of interest, but of equal importance is its shape—for this profile tells much about pump performance. Sucker rod pumping can be treated as a communication system with the rod string acting as the transmission line—the downhole pump as a transmitter, and the polished rod dynamometer as the receiver.

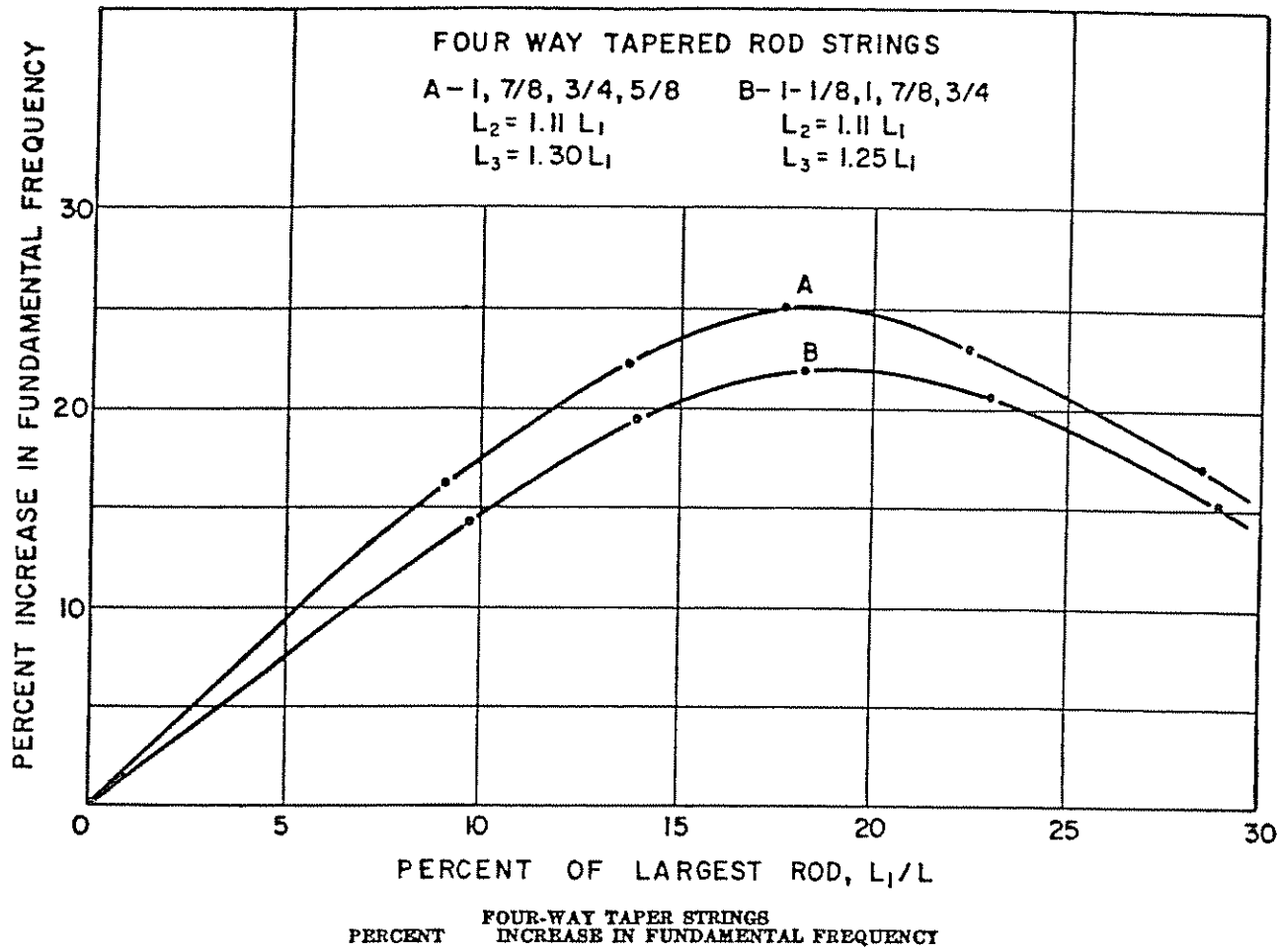


Fig. 2.37 (after Ref. 3)

Each time the downhole pump makes a stroke, a unique force signal is sent along the sucker rod string at the speed of about 15,800 feet per second to the surface where it is recorded by the dynamometer.

Since each different bottom hole condition sends a different force signal to the surface, an experienced analyst can in some cases determine downhole pump performance by interpreting the distinctive profile of the dynamometer card to detect some or all of the following:

(1) Condition of the traveling and standing valves
(2) Whether a gas or fluid pound exists; its magnitude
(3) A gas locked pump
(4) Excessive friction
(5) Whether or not the well is pumped off
(6) Whether the dynamometer card is an overtravel or an undertravel card

2.102 Limitation of visual interpretation

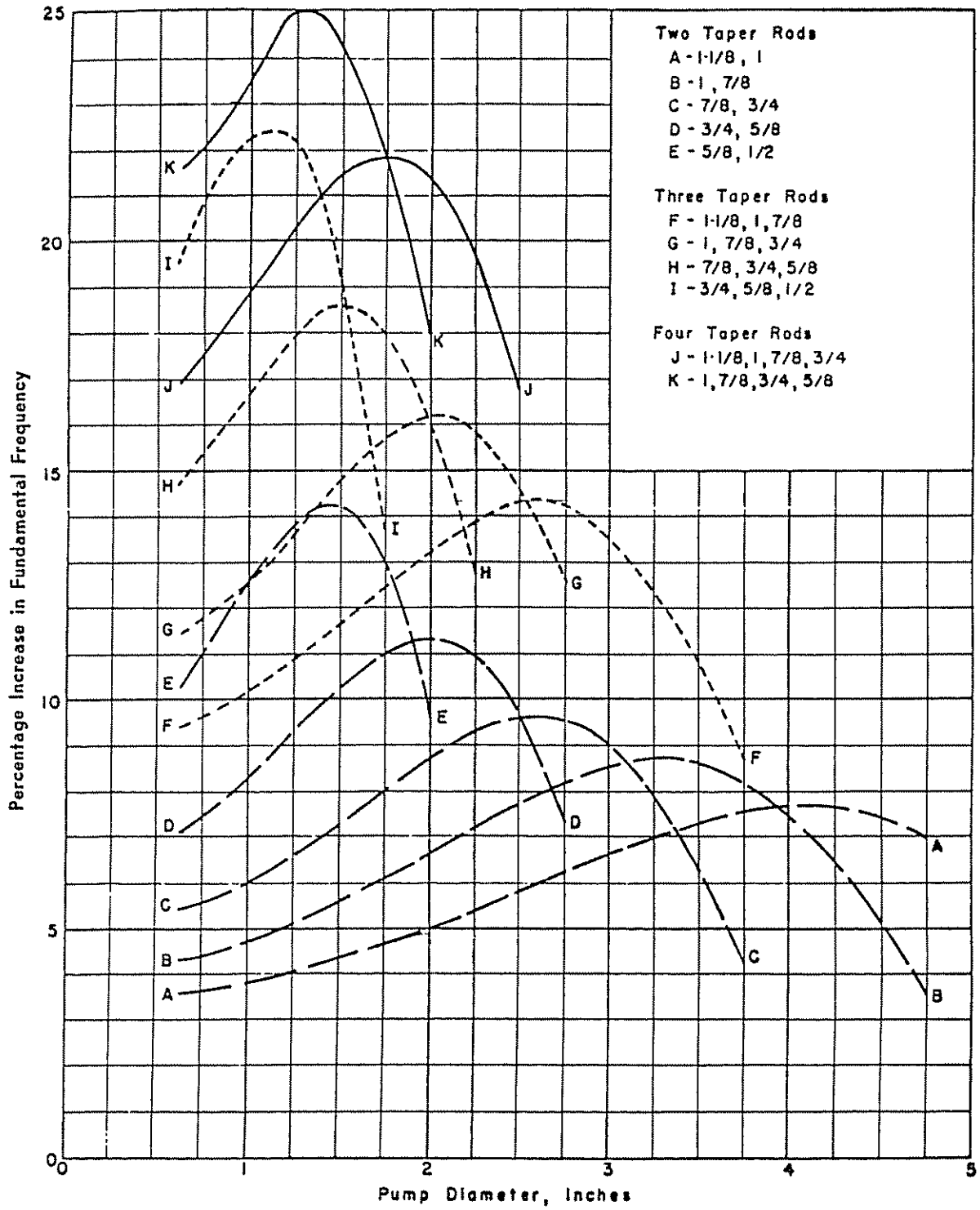
In shallow wells where rod and fluid loads behave like a concentrated mass, only a surface dynamometer card is required to determine loading and to diagnose trouble from the prime mover to the bottom hole pump. But in deep wells the elastic nature of the rod string presents a more complex load pattern and accurate visual diagnosis of downhole problems from a surface dynamometer card is seldom possible. Though much information can be gained from visual interpretation of surface data, this information is qualitative in nature, and success in visual interpretation is directly dependent upon the skill of the dynamometer analyst. Often, even the most experienced analysts are misled into an incorrect diagnosis.

A theoretical dynamometer card is shown in Figure 2.39 for both inelastic and elastic rods. Part A of Figure 2.39 would represent a situation when:[9]

(1) The rods were completely rigid
(2) There was no time lag in transmitting motion from the surface to the plunger
(3) There were no dynamic effects
(4) There was no vibration
(5) All components of the system perform with 100 percent efficiency

At point (a) the traveling valve closes at the beginning of the upstroke and the rod string takes up the whole load instantaneously, represented from (a) to (b). From (b) to (c) the load remains constant until the top of the upstroke is reached at (c). Then the traveling valve opens, the standing valve closes, and the load is removed from the rods from (c) to (d) instantaneously. The load again remains constant from (d) to (a) as the rod string is falling freely. At point (a) the cycle is repeated.

Since sucker rods are not rigid, B of Figure 2.39 rep-



PERCENTAGE INCREASE IN FUNDAMENTAL FREQUENCY SPECIFIC ROD COMBINATIONS

Fig. 2.38 (after Ref. 3)

resents an elastic system (with the other simplifying characteristics mentioned above). At the beginning of the upstroke (a), the load begins to increase gradually because of the stretch of the rods. The maximum polished rod load is reached at (b) and remains constant to point (c), at which point the rods start downward and begin to contract. At point (d) the fluid load has been transferred to the standing valve and the minimum polished rod load again remains constant from (d) to (a); then the cycle is repeated.

The theoretical card is seldom encountered in practice because of many factors. Figure 2.40 represents a

TABLE 2.15 (After Ref. 3)
EXAMPLE DESIGN CALCULATIONS
CONVENTIONAL SUCKER ROD PUMPING SYSTEM

Object: To solve for—Sp, PD, PPRL, MPRL, PT, PRHP, and CBE
Known or assumed data:

Fluid level, D = 4,500 ft.
Pump depth, L = 5,000 ft.
Tubing size 2 in. Is it anchored? Yes, (No)
Pumping speed, N = 16 SPM
Length of stroke, S = 54 in.
Sucker rods 33.8% - ⅞" & 66.2% - ¾"
Plunger diameter, D_p = 1.50 in.
Spec. grav. of fluid, G = 0.9

Record factors from tables

1. W_r = 1.833 lb./ft.
2. E_r = 0.804×10^{-6}
3. F_c = 1.082
4. E_t = 0.307×10^{-6}

Calculate non-dimensional variables:

5. $F_o = .340 \times G \times D_p^2 \times D = .340 \times 0.9 \times 2.25 \times 4{,}500 = 3{,}098$ lbs.
6. $1/k_r = E_r \times L = 0.804 \times 10^{-6} \times 5{,}000 = 4.020 \times 10^{-3}$ in/lb.
7. $Sk_r = S \div 1/k_r = 54 \div 4.020 \times 10^{-3} = 13{,}433$ lbs.
8. $F_o/Sk_r = 3{,}098 \div 13{,}433 = 0.231$
9. $N/N_o = NL \div 245{,}000 = 16 \times 5{,}000 \div 245{,}000 = 0.326$
10. $N/N_o' = N/N_o \div F_c = 0.326 \div 1.082 = 0.301$
11. $1/k_t = E_t \times L = 0.307 \times 10^{-6} \times 5{,}000 = 1.535 \times 10^{-3}$ in/lb.

Solve for S_p and PD:

12. S_p/S = 0.86
13. $S_p = [(S_p/S) \times S] - [F_o \times 1/k_t] = [0.86 \times 54] - [3{,}098 \times 1.535 \times 10^{-3}] = 41.7$ in.
14. $PD = 0.1166 \times S_p \times N \times D_p^2 = 0.1166 \times 41.7 \times 16 \times 2.25 = 175$ barrels per day

If the calculated pump displacement fails to satisfy known or anticipated requirements, appropriate adjustments must be made in the assumed data and steps 1 through 14 repeated. When the calculated pump displacement is acceptable, proceed with the Design Calculation.

Determine non-dimensional parameters:

15. $W = W_r \times L = 1.833 \times 5{,}000 = 9{,}165$ lbs.
16. $W_{rl} = W[1 - (.128G)] = 9{,}165 [1 - (.128 \times 0.9)] = 8{,}110$ lbs.
17. $W_{rl}/Sk_r = 8{,}110 \div 13{,}433 = 0.604$

Record non-dimensional factors from figures

18. F_1/Sk_r = 0.465
19. F_2/Sk_r = 0.213
20. $2T/S^2k_r$ = 0.37
21. F_3/Sk_r = 0.29
22. T_a = 0.997

Solve for operating characteristics:

23. $PPRL = W_{rl} + [(F_1/Sk_r) \times Sk_r] = 8{,}110 + [0.465 \times 13{,}433] = 14{,}356$ lbs.
24. $MPRL = W_{rl} - [(F_2/Sk_r) \times Sk_r] = 8{,}110 - [0.22 \times 13{,}433] = 5{,}249$ lbs.
25. $PT = (2T/S^2k_r) \times Sk_r \times S/2 \times T_a = 0.37 \times 13{,}433 \times 27 \times 0.997 = 133{,}793$ in.-lbs
26. $PRHP = (F_3/Sk_r) \times Sk_r \times S \times N \times 2.53 \times 10^{-6} = 0.29 \times 13{,}433 \times 54 \times 16 \times 2.53 \times 10^{-6} = 8.5$
27. $CBE = 1.06 (W_{rl} + \frac{1}{2}F_o) = 1.06 \times (8{,}110 + 1{,}549) = 10{,}239$ lbs.

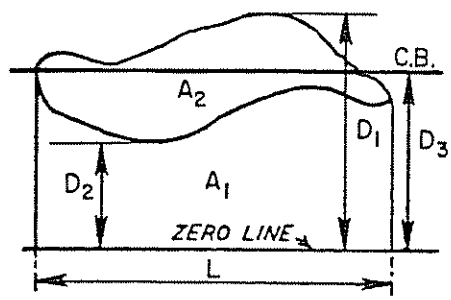
typical pumping cycle on a dynamometer card. In this illustration, the "scale" is 6950 lb/in. of card. The card is self-explanatory.

Figure 2.41 is another illustration of a typical dynamometer card for a crank-counterbalanced unit.

2.103 Loads from dynamometer cards

The loads imposed upon a sucker-rod pumping installation can be determined from the dynamometer card. The procedure is illustrated in Figure 2.42. According to the figure, the nomenclature is as follows:

C = calibration constant of the dynamometer, pounds per inch of card height
D_1 = maximum deflection (along y-axis), in.
D_2 = minimum deflection, in.
A_1 = lower area of card, sq in.
A_2 = upper area of card, sq in.



DYNAMOMETER CARD showing areas and deflections needed for calculating loads.

Fig. 2.42 (after Ref. 9)

$$\text{Maximum load} = C \times D_1 \quad (2.83)$$
$$\text{Minimum load} = C \times D_2 \quad (2.84)$$

TABLE 2.16 (After Ref. 3)
DESIGN CALCULATIONS
for
CONVENTIONAL SUCKER ROD PUMPING SYSTEMS

Refer to API RP 11L for the explanation of this form and the necessary tables and figures required for this calculation.

Company _____ Well Name _____ Date _____
Field _____ County _____ State _____
Required pump displacement, PD _____ bbls./day Maximum allowable rod stress _____ psi
Fluid level, D = _____ ft. Pumping speed, N = _____ SPM Plunger diameter, D_p = _____ in.
Pump depth, L = _____ ft. Length of stroke, S = _____ in. Spec. grav. of fluid, G = _____
Tubing size _____ in. Is it anchored? Yes, No Sucker rods _____

Record factors

1. W_r = _____ lb/ft
2. E_r = _____
3. F_c = _____
4. E_t = _____

Calculate non-dimensional variables:

5. $F_o = .340 \times G \times D_p^2 \times D = .340 \times$ _____ × _____ × _____ = _____ lbs.
6. $1/k_r = E_r \times L =$ _____ × _____ = _____ in/lb.
7. $Sk_r = S \div 1/k_r =$ _____ ÷ _____ = _____ lbs.
8. $F_o/Sk_r =$ _____ ÷ _____ = _____
9. $N/N_o = NL \div 245{,}000 =$ _____ × _____ ÷ 245,000 = _____
10. $N/N_o' = N/N_o \div F_c =$ _____ ÷ _____ = _____
11. $1/k_t = E_t \times L =$ _____ × _____ = _____ in/lb.

Solve for S_p and PD:

12. $S_p/S =$ _____
13. $S_p = [(S_p/S) \times S] - [F_o \times 1/k_t] =$ [_____ × _____] − [_____ × _____] = _____ in.
14. $PD = 0.1166 \times S_p \times N \times D_p^2 = 0.1166 \times$ _____ × _____ × _____ = _____ barrels per day

If calculated pump displacement is unsatisfactory make appropriate adjustments in assumed data and repeat steps 1 through 14.

Determine non-dimensional parameters:

15. $W = W_r \times L =$ _____ × _____ = _____ lbs.
16. $W_{rf} = W[1 - (.128G)] =$ _____ [1 − (.128 × _____)] = _____ lbs.
17. $W_{rf}/Sk_r =$ _____ ÷ _____ = _____

Record non-dimensional factors

18. $F_1/Sk_r =$ _____
19. $F_2/Sk_r =$ _____
20. $2T/S^2k_r =$ _____
21. $F_3/Sk_r =$ _____
22. $T_a =$ _____

Solve for operating characteristics:

23. $PPRL = W_{rf} + [(F_1/Sk_r) \times Sk_r] =$ _____ + [_____ × _____] = _____ lbs.
24. $MPRL = W_{rf} - [(F_2/Sk_r) \times Sk_r] =$ _____ − [_____ × _____] = _____ lbs.
25. $PT = (2T/S^2k_r) \times Sk_r \times S/2 \times T_a =$ _____ × _____ × _____ × _____ = _____ in.-lbs.
26. $PRHP = (F_3/Sk_r) \times Sk_r \times S \times N \times 2.53 \times 10^{-6} =$ _____ × _____ × _____ × _____ × 2.53×10^{-6} = _____
27. $CBE = 1.06 (W_{rf} + \frac{1}{2} F_o) = 1.06 \times$ (_____ + _____) = _____ lbs.

Remarks _____

$$\text{Range of load} = C(D_1 - D_2) \quad (2.85)$$

$$\text{Average upstroke load} = C(A_1 + A_2)/L \quad (2.86)$$

$$\text{Average downstroke load} = C A_1/L \quad (2.87)$$

Work per stroke delivered to the rod string against the elevation of fluid and against friction = A_2 (converted to ft-lb of work).
Work per stroke performed against rod elevation (or rod fall) = A_1 (converted to ft-lb of work).

2.104 Counterbalance effect from dynamometer cards

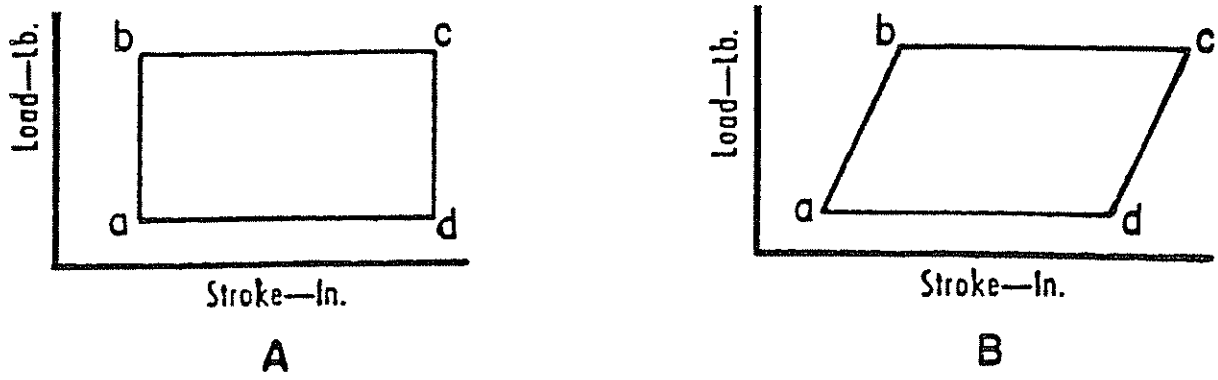
The counterbalance effect from dynamometer cards is determined for conventional units as follows:

(1) The counterbalance line (CB) is drawn on the dynamometer card by stopping the pumping unit at the position of maximum counterbalance effect. This will occur when the crank arm is horizontal on the upstroke, at $\Theta = 90°$ and 270°, where Θ is the angle measured in the clockwise direction from the 12 o'clock position.
(2) The approximate "ideal" counterbalance is then

$$\text{Ideal C.B.} = \frac{PPRL + MPRL}{2} \quad (2.88)$$

(3) Computed counterbalance effect is

$$\text{Actual C.B.} = C \times D_3 \text{ (See Fig. 2.42)} \quad (2.89)$$



THEORETICAL dynanometer cards; nonelastic rods (A), elastic rods (B).

Fig. 2.39 (after Ref. 9)

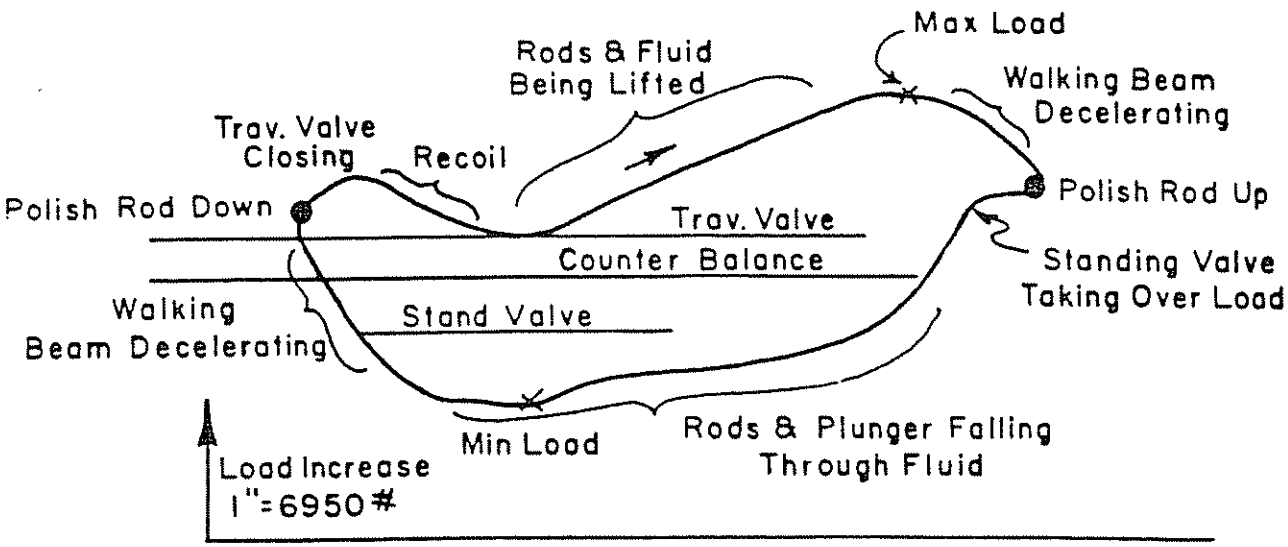


Fig. 2.40 (after Ref. 17)

(4) The correct counterbalance can be determined from the relationship

$$\text{Correct C.B.} = \tfrac{1}{2}(\text{Avg. upstroke load} + \text{avg. Downstroke load}) \qquad (2.90)$$

If the counterbalance line has not been drawn, then the approximate counterbalance can be determined from:

$$\text{Approximate Correct C.B.} = C(A_1 + A_2/2)/L \qquad (2.91)$$

This relationship gives the same result as above.

2.105 Polished rod horsepower from dynamometer cards

The polished rod horsepower can be determined from the dynamometer card according to the following relationship.:

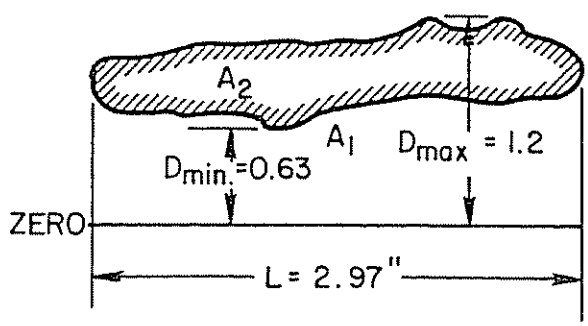
$$PRHP = C(A_2/L) \times \frac{S \times N}{33{,}000(12)} \qquad (2.92)$$

where:

S = stroke length, in.
N = strokes per minute
A_2 = area of dynamometer card, sq in.
L = length of card, inches

EXAMPLE PROBLEM #13

Computations of equipment performance (derived from dynamometer cards)
Following data taken from card:



$D_{max} = 1.20'' =$ max. def.
$D_{min} = 0.63'' =$ min. deflection
$A_1 = 2.10$ sq in. (lower area)
$A_2 = 1.14$ sq in. (upper area)
$L = 2.97'' =$ length of card

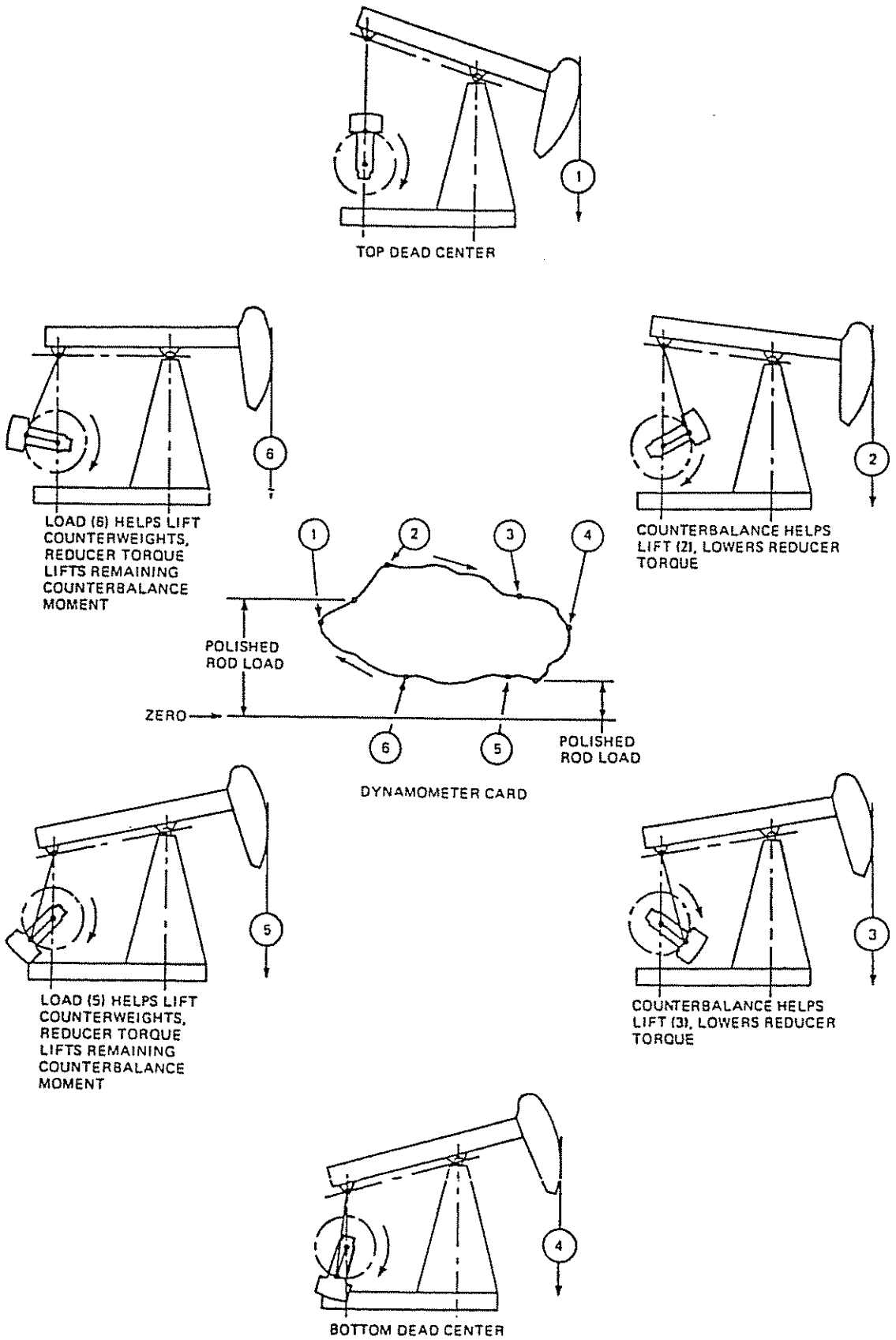


Fig. 2.41 Crank-Counterbalanced Diagram (alter Ref. 22)

C = Calibration constant of dynamometer (In above case assumed to be 12,800 lb/in.)
S = 45 in.
N = 18.5 spm

Solution:

(1) Max. load = $C \times D_{max}$ = 12,800 × 1.20″ = 15,400 lb.

(2) Min. load = $C \times D_{min}$ = 12,800 × 0.63″ = 8100 lb.

(3) Average upstroke load = $C\left(\frac{A_2 + A_1}{L}\right)$
$= 12{,}800\left(\frac{1.14 + 2.10}{2.97}\right)$
= 14,000 lb.

(4) Average downstroke load = $C\left(\frac{A_1}{L}\right)$
$= 12{,}800\left(\frac{2.10}{2.97}\right)$
= 9100 lb

(5) Correct counterbalance = $C\left(\frac{A_1 + \frac{A_2}{2}}{L}\right)$
$= 12{,}800\left(\frac{2.10 + \frac{1.14}{2}}{2.97}\right)$
= 11,500 lb.

(6) H.P. at polished rod = $C\left(\frac{A_2}{L}\right)$ × length of stroke, inches × $\frac{\text{strokes per min.}}{33{,}000(12)}$

PRHP = HP at polished rod =
$12{,}800\left(\frac{1.14}{2.97}\right) \times \frac{45 \times 18.5}{12 \times 33{,}000}$

PRHP = 10.3

2.106 Torque from dynamometer cards

An accurate method for determining the instantaneous torque throughout the pumping cycle is the *torque factor method,* which uses torque factors and polished-rod position data together with maximum counterbalance moments according to API standard 11-E.[20] This method is now in general use.

The torque factor method uses torque factors and corresponding positions of the polished rod. The API standard 11-E requires manufacturers of pumping units to supply the purchaser, upon request, stroke and torque factors for each 15° position of the crank. Torque factors are derived from the geometry of the particular pumping unit.

The instantaneous torque due to net well load at a given crank position is the torque factor at that position multiplied by the net well load at that position. Net well load is:

$$W_n = \text{net well load} = (W - B) \tag{2.93}$$

where:

W = well load at a specific crank angle
B = structural unbalance of the pumping unit (either plus or minus value)

Thus, torque due to net well load is:

$$T_{wn} = \overline{TF} \times W_n \tag{2.94}$$

The torque due to rotary counterweights is M sin Θ, where M is the maximum moment of crank and counter weights about the crankshaft (supplied by manufacturer). The net torque on the speed reducer is the difference between the torque due to net well load and the torque due to rotary counterweights:

$$T_{NET} = \overline{TF}\,(W - B) - M \sin \Theta \tag{2.95}$$

The torque factor analysis method can best be illustrated by the following example for a conventional unit.

EXAMPLE PROBLEM #14

This example calculation is taken from the Lufkin Engineering Manual and is used with their permission.[30]

Field data are shown on Figure 2.43. The field-measured dynamometer card has been traced on the upper left hand space of the dynamometer report (see Fig. 2.44). Note that the zero lines of the dynamometer card and the report form have been carefully matched.

The procedure is as follows:

(1) Draw two lines perpendicular to the zero line, touching the ends of the card curve. The card length is proportional to the length of stroke, and the position of rods (see Fig. 2.45) is a fraction of this distance. The polished rod position on the card for 120° crank angle equals 0.781 times 4.85″ (card length). Locate "position of rods" (Fig. 2.45) on the card curve for every 15° position of the crank.
CAUTION: Position of rods and torque factor (Fig. 2.45) are expressed clockwise from the 12 o'clock crank position with wellhead to the right. If rotation of unit in question is counterclockwise, "position of crank" on Figure 2.45 should be changed to read 345° for 15°, 330° for 30°, 315° for 45°, etc. and the signs of "torque factor" values reversed. (Referring to Fig. 2.45, with counterclockwise crank rotation, a 345° crank position, position of rods = 0.017, torque factor = −18.465; at 90° crank position, position of rods = 0.573, torque factor = +60.964.)

(2) Skip columns 3 and 4 on calculation sheet, Figure 2.46. Obtain "A" distances at each angular position of the crank as shown on Figure 2.44, record in column 5.

(3) Obtain column 6 by multiplying the dynamometer constant times "A" in column 5.

(4) Column 7 is column 6 minus the structural unbalance (SU) obtained from Figure 2.47. Observe sign conventions.

(5) Fill in column 8 obtaining torque factors from Figure 2.45. Use the proper stroke length and observe crank rotation (see "Caution" note, Step 1).

(6) The well load torque in column 9 is the product of columns 7 and 8. This would be the torque with no counterbalance torque offsetting it.

(7) Maximum counterbalance torque "Q" in the upper right hand corner of Figure 2.46 is obtained from one of the charts on Figure 2.47.

ENGINEERING MANUAL - LUFKIN FOUNDRY & MACHINE COMPANY

DYNAMOMETER CARD DATA

COMPANY _____ LEASE _____ WELL NO. _____

WELL LOCATION: TOWN Sylvia STATE Kansas

WELL DATA:

Well Depth 3715 Ft.
Pump Depth 3650 Ft.
Plunger 2 In.
Tubing 2 1/2 In.
Anchored? Yes
Fluid Level 3193 Ft.

RODS: 1-1/8 for _____ Ft.
1 for _____ Ft.
7/8 for 1575 Ft.
3/4 for 2100 Ft.
5/8 for _____ Ft.
POLISHED ROD 1 1/4 In.

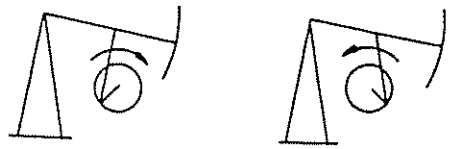
PRODUCTION:

Fluid _____ Bbl./Day
Oil _____ Bbl./Day
Gravity _____
Water _____ Bbl./Day
Pumping _____ Hrs./Day

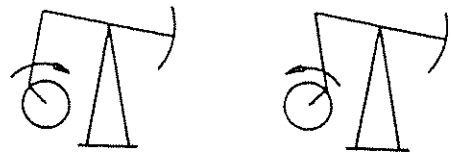
OBSERVED DATA:

Unit C-456D-256-120, Order 211954, Serial 3207B
Strokes 18 in 60 Sec, Actual Stroke Length (at Polished Rod) 85 In.
Crank No. 849SB, No. C'Wts. 4 C'Wt. No. 3CR0 No. Aux. Wts. _____ Aux. Wt. No. _____
Location C'Wts. 28 30 32 30 In. From Long End of Cranks.
Beam Weights: Number _____, Part No. _____, Wt. Location _____.
Air Pressure: Max. _____ PSI, Min. _____ PSI.
Prime Mover Lufkin H-795.
Dynamometer: Type J.F.
Constant 8,950 Lbs/In.

Unit Rotation * Clockwise.



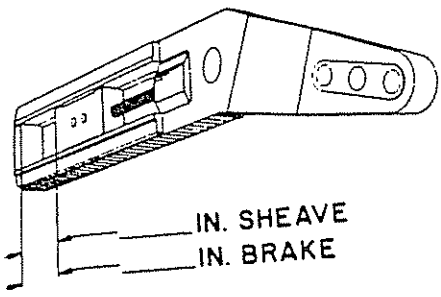
*Clockwise *Counterclockwise



REASON FOR WEIGHING WELL: _____

REMARKS: _____

SKETCH:



WEIGHED:

By _____

Date _____

Fig. 2.43 (after Ref. 30)

ENGINEERING MANUAL - LUFKIN FOUNDRY & MACHINE COMPANY

LUFKIN FOUNDRY AND MACHINE COMPANY

LUFKIN, TEXAS

DYNAMOMETER REPORT

COMPANY _____, WELL _____, LOCATION _____

WELL DATA:

Well Depth 3715 Ft.
Pump Depth 3650 Ft.
Plunger 2 In.
Tubing 2 1/2 In.
Anchored? Yes
Pump Submergence 3193 Ft.

RODS: 1 1/8 for _____ Ft.
1 for _____ Ft.
7/8 for 1575 Ft.
3/4 for 2100 Ft.
5/8 for _____ Ft.
1/2 for _____ Ft.

PRODUCTION:

Fluid 720 Bbl./Day
Water 584 Bbl./Day
Oil 140 Bbl./Day
Gravity 42.3
Pumping _____ Hrs/Day

OBSERVED DATA:

Unit C-4560-256-120, Order 211954, Serial 32078
Stroke 85 In., SPM 18, Prime Mover LUFKIN H-795, Polished Rod 1 1/4 In.
Cranks 84958, No. C'Wts. 4, C'Wt. No. 3220, No. Aux. Wts. ____, Aux. Wt. No. ____
Location C'Wts. 30 In. From End of Cranks, Location Trim Wts. ____ In. From End of Pocket.
Air Pressure: Max. ____ PSI, Min. ____ PSI; Number of Beam Weights ____
Dynamometer Type U.F., Constant 8.950 Lbs./In. Crank Rotation: CLOCKWISE

CALCULATED:

Assumed Structural Unit Efficiency 93 %, Polished Rod H.P. 21.5
Peak Torque, Upstroke 176,500 In.-Lbs., Downstroke 550,000 In.-Lbs.
Peak P.R. Load 15,750 Lbs., Rod Stress _____ PSI.

REMARKS: LOCATE C'WTS. GG FROM LONG END OF CRANK TO MAKE TORQUE PEAKS EQUAL AT 359,000 IN LBS

*Viewed with Well Head to the Right.

WEIGHED: By _____ Date _____

CALCULATED: Method TORQUE FACTOR By _____, Date _____

CHECKED: By _____, Date _____

CRANK ANGLE - MARK II ONLY

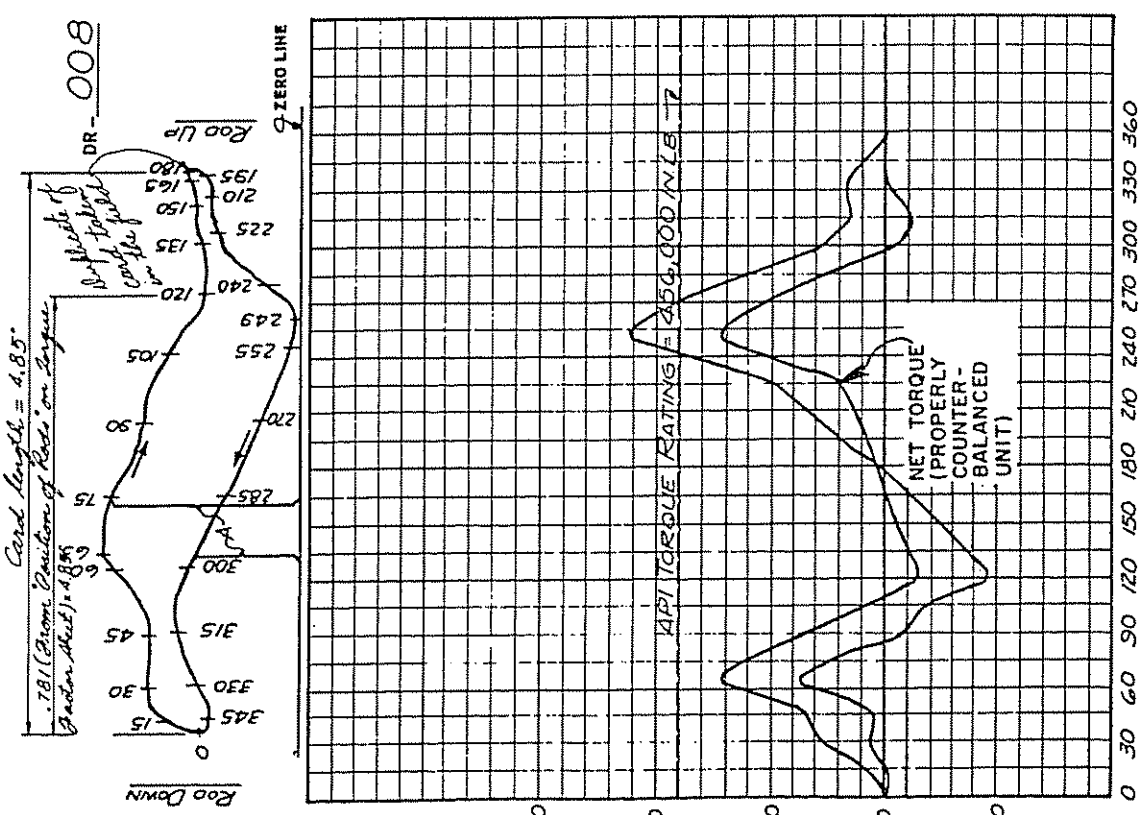


Fig. 2.44 (after Ref. 30)

PUMPING UNIT STROKE AND TORQUE FACTOR

Name of Manufacturer: LUFKIN FOUNDRY & MACHINE COMPANY

Designation of Unit: C-456D-213-120, C-320D-213-120, C-456D-256-120, C-320D-256-120

120" STROKE

POSITION OF CRANK (Deg.)	POSITION OF RODS	TORQUE FACTOR
0	0.000	- 3.052
15	0.017	18.465
30	0.079	38.673
45	0.181	53.725
60	0.306	61.244
75	0.440	61.602
90	0.569	57.049
105	0.686	49.980
120	0.785	41.961
135	0.867	33.648
150	0.930	25.038
165	0.975	15.726
180	0.998	5.118
195	0.996	- 7.241
210	0.965	-21.143
225	0.904	-35.264
240	0.814	-47.555
255	0.701	-56.348
270	0.573	-60.964
285	0.440	-61.411
300	0.310	-57.837
315	0.193	-50.255
330	0.096	-38.533
345	0.029	-22.627
360	0.000	- 3.052

102" STROKE

POSITION OF CRANK (Deg.)	POSITION OF RODS	TORQUE FACTOR
0	0.000	- 1.814
15	0.018	15.641
30	0.079	31.827
45	0.177	44.121
60	0.299	50.885
75	0.431	52.164
90	0.561	49.282
105	0.680	43.852
120	0.784	37.090
135	0.869	29.623
150	0.934	21.596
165	0.978	12.849
180	0.999	3.120
195	0.993	- 7.703
210	0.959	-19.277
225	0.895	-30.655
240	0.804	-40.523
255	0.691	-47.726
270	0.564	-51.616
285	0.431	-51.982
300	0.302	-48.797
315	0.185	-42.057
330	0.090	-31.772
345	0.026	-18.129
360	0.000	- 1.814

85" STROKE

POSITION OF CRANK (Deg.)	POSITION OF RODS	TORQUE FACTOR
0	0.000	- 0.976
15	0.018	12.807
30	0.078	25.479
45	0.172	35.302
60	0.291	41.157
75	0.422	42.927
90	0.552	41.315
105	0.674	37.331
120	0.781	31.843
135	0.869	25.401
150	0.936	18.235
165	0.981	10.374
180	0.999	1.789
195	0.991	- 7.434
210	0.953	-16.932
225	0.887	-26.021
240	0.794	-33.826
255	0.681	-39.546
270	0.554	-42.636
285	0.421	-42.820
300	0.293	-40.000
315	0.178	-34.175
330	0.085	-25.450
345	0.024	-14.145
360	0.000	- 0.976

67" STROKE

POSITION OF CRANK (Deg.)	POSITION OF RODS	TORQUE FACTOR
0	0.000	- 0.448
15	0.019	10.103
30	0.076	19.592
45	0.168	27.170
60	0.283	32.003
75	0.412	33.909
90	0.542	33.201
105	0.666	30.447
120	0.776	26.216
135	0.868	20.936
150	0.937	14.870
165	0.982	8.162
180	1.000	0.931
195	0.989	- 6.635
210	0.948	-14.201
225	0.879	-21.269
240	0.785	-27.254
255	0.671	-31.602
270	0.543	-33.896
285	0.411	-33.881
300	0.284	-31.452
315	0.171	-26.625
330	0.081	-19.563
345	0.022	-10.624
360	0.000	- 0.448

GEOMETRIC FACTOR, AT MAXIMUM STROKE: .980

1. Position of cranks is the angular displacement measured clockwise from the 12 O'clock position viewed with the well head to the right for crank balanced units. For air balanced units the angular displacement is measured clockwise from the 6 O'clock position.
2. Position of rods is expressed as a fraction of stroke above lowermost position.
3. Torque factor = $\frac{T}{W}$ where T = torque on pumping unit reducer due to polished rod load W.
4. Negative signs on torque factor indicate a clockwise torque on crankshaft.

Fig. 2.45 (after Ref. 30)

CRANKSHAFT TORQUE CALCULATION SHEET

DR- 008

UNIT C-456D-256-120
WELL
UNIT ROTATION Clockwise
STRUCTURAL UNBALANCE, S.U. = 55 LBS.
DYNAMOMETER CONSTANT, D.C. = 8,950 LBS./IN.
ASSUMED STRUCTURAL EFFICIENCY, EFF. 99 %

CRANK OFFSET ANGLE, α = ____°
STROKE 85 IN. SPM 18

AIR C'Bal. $B = \frac{\text{MAX. EFFECTIVE C'B.}}{\text{D.C.}} = \_\_\_\_ = \_\_\_\_$ IN.

AIR C'Bal. $C = \frac{\text{MIN. EFFECTIVE C'B.}}{\text{D.C.}} = \_\_\_\_ = \_\_\_\_$ IN.

MAX. C'BALANCE TORQUE, Q = 355,800 IN-LBS.

PEAK P.R. LOAD = MAX. A X D.C. = 476 x 8,950 = 15,750 LBS.

$\text{P.R.H.P.} = \frac{\text{AREA CARD X D.C. X STROKE X SPM}}{\text{LENGTH OF CARD X 12 X 33,000}} = \frac{3.02 \times 8,950 \times 85 \times 18}{4.85 \times 12 \times 33,000} = 21.5$

1	2	3	4	5	6	7	8	9	10	11	12
θ	SIN θ	$(\theta + \alpha)$	SIN $(\theta + \alpha)$	A DYNAMOMETER CARD	A X D.C.	(A X D.C.)-(S.U.)	TORQUE FACTOR	WELL LOAD TORQUE 7 X 8	C'B. TORQUE 9 X 2 CONV. UNIT 9 X 4 MARK II UNIT	T_T THEORETICAL NET TORQUE 9-10 CONV. AND MARK II 9 X B AIR BAL.	NET TORQUE (+) 11 X 100/EFF. (-) 11 X EFF/100
0	.00000			.88	7,880	7,825	-0.976	- 7,640	0	- 7,640	- 7,100
15	.25882			1.18	10,550	10,495	12.807	134,500	144,000	- 9,500	- 8,840
30	.50000			1.34	12,000	11,945	25.479	304,000	278,000	26,000	28,000
45	.70711			1.32	11,800	11,745	35.302	414,000	393,000	21,000	22,600
60	.86603			1.66	14,850	14,795	41.157	607,000	481,000	126,000	135,500
75	.96592			1.68	15,050	14,995	42.927	644,000	537,000	107,000	115,000
90	1.0000			1.39	12,450	12,395	41.315	512,000	556,000	- 44,000	- 40,900
105	.96592			1.16	10,400	10,345	37.331	386,000	537,000	-151,000	-140,500
120	.86603			.84	7,520	7,465	31 843	237,000	481,000	- 244,000	- 227,000
135	.70711			.88	7,870	7,815	25.401	198,500	393,000	-194,500	-181,000
150	.50000			.93	8,320	8,265	18.235	150,500	278,000	-127,500	-118,500
165	.25882		-	.96	8,600	8,545	10.374	88,700	144,000	- 55,300	- 51,400
180	.00000		-	.97	8,690	8,635	1.789	15,450	0	15,450	16,600
195	- .25882		-	.84	7,520	7,465	-7.434	- 55,500	-144,000	88,500	95,200
210	- .50000		-	.80	7,160	7,105	16.932	- 120.000	-278,000	158,000	170,000
225	- .70711		-	.73	6,530	6,475	-26.021	-168,500	- 393,000	224,500	241,000
240	- .86603		-	.28	2,510	2,455	-33.826	- 83,100	- 481,000	397,900	428,000
255	- .96592		-	.08	716	661	-39.546	- 26,100	-537,000	510,900	549,500
270	-1.0000		-	.35	3,130	3,075	-42.636	- 131,000	- 556,000	425,000	457,000
285	- .96592		-	.65	5,820	5,765	-42.820	- 247,000	-537,000	290000	312,000
300	- .86603		-	.99	8,860	8,805	-40.000	- 352,000	481,000	129,000	139,000
315	- .70711		-	1.06	9,490	9,435	34.175	- 322,000	- 393,000	71,000	76,400
330	- .50000		-	.90	8,030	7,995	-25.450	- 204 000	-278,000	74,000	79,600
345	- .25882			.79	7,070	7,015	-14.145	- 99,200	- 144,000	44,800	48,200
63	.89 (b)			1.76	15,750	15,695	42.0	659,000 (a)	495,000	164,000	176,500
249	- 95 (d)			.03	269	214	-37.3	- 7,980 (c)	- 519,000	511,020	550,000

FOR CONVENTIONAL UNIT USE COLUMNS 1, 2, 5 - 12.
FOR MARK II UNIT USE COLUMNS 1, 3 - 12.
FOR AIR BALANCED UNIT USE COLUMNS 1, 5, 6, 8, 11, 12
CALCULATED: BY CWD DATE

To Counterbalance Unit Properly, Find Q_R = Req'd. C'Balance Torque, As Follows:

Upstroke Peak Torque = Downstroke Peak Torque

(a) (b) (c) (d)

$659,000 - .8910Q_R = -7,980 - (-.9340Q_R)$

$659,000 - .8910Q_R = .9340Q_R - 7,980$

$1.8250Q_R = 666,980$

$Q_R = \frac{666,980}{1.825} = 365,000$ in-lbs

New Theoretical Torque Peaks For Properly C'Balanced Unit:

$T_T = 659,000 - .891Q_R = 659,000 - (.891 \cdot 365,000)$

$T_T = 659,000 - 325,000 = 334,000$

New Net Torque Peaks = $\frac{T_T}{.93} = \frac{334,000}{.93(e)} = 359,000$ in-lbs

C'Balance Torque Chart Shows 66° From Long End Of Cranks For Q_R = 365,000 in-lbs Note "Remarks" On Dynamometer Report Sheet

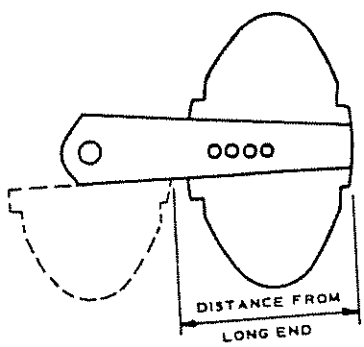
Caution: When Q_R Differs Drastically From Q, Location Of Torque Peaks May Shift From Original Crank Position

Fig. 2.46 (after Ref. 30)

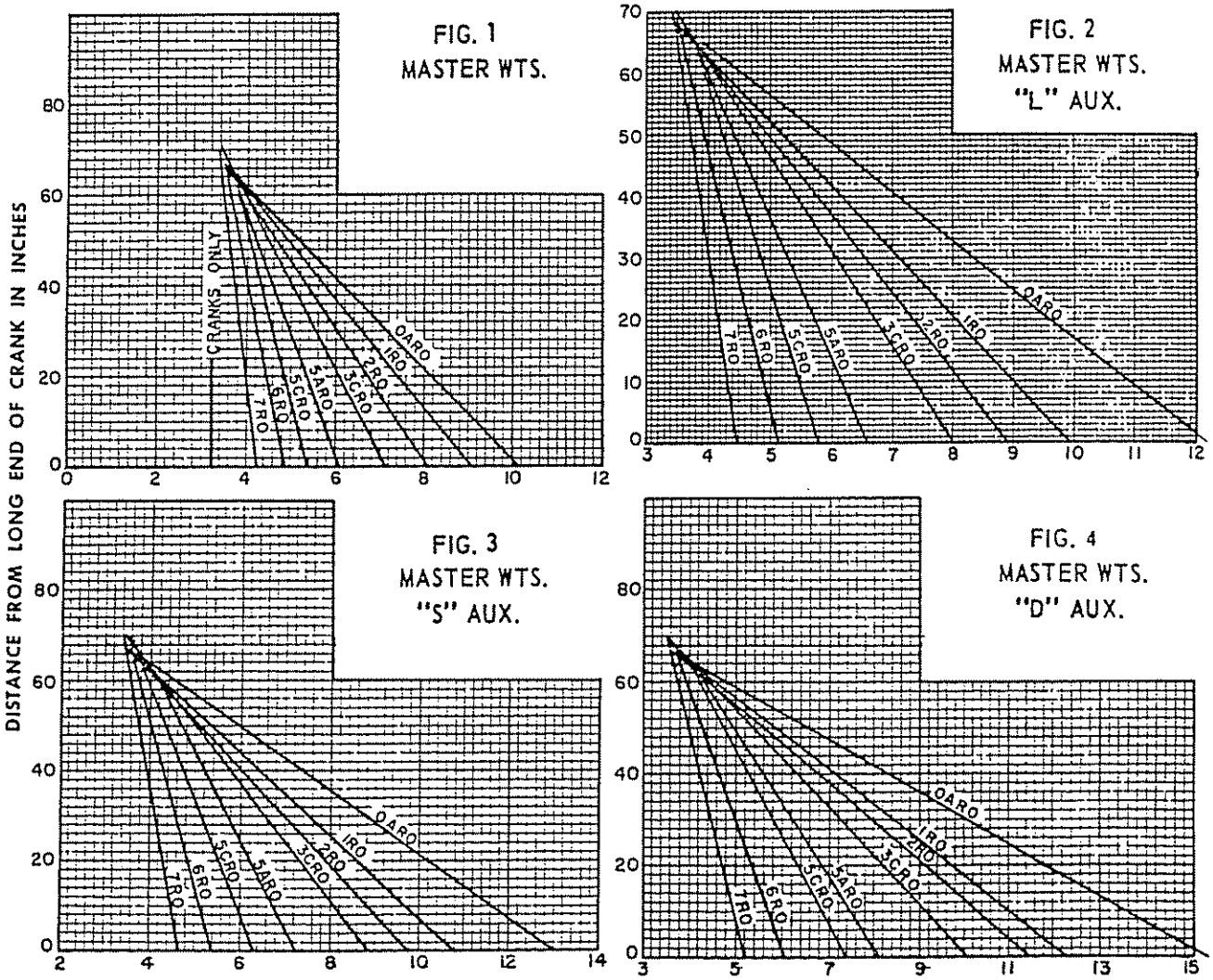
LUFKIN FOUNDRY AND MACHINE COMPANY – LUFKIN, TEXAS

COUNTERBALANCE CHART

No. 8495B Cranks



UNIT	STRUCTURAL UNBALANCE
C-640D-365-100	+ 620 Lbs.
C-640D-304-120	- 120 Lbs.
C-456D-304-120	- 120 Lbs.
C-456D-256-120	+ 55 Lbs.
C-456D-213-120	0 Lbs.
C-456D-365-100	+ 620 Lbs.
C-456D-298-100	+ 550 Lbs.
C-456D-256-100	+ 500 Lbs.
C-456D-298-86	+1000 Lbs.
C-320D-256-120	+ 55 Lbs.
C-320D-213-120	0 Lbs.
C-320D-298-100	+ 550 Lbs.
C-320D-256-100	+ 500 Lbs.
C-320D-256-86	+1000 Lbs.
C-320D-246-86	+ 800 Lbs.
C-228D-246-86	+ 800 Lbs.



MAXIMUM COUNTERBALANCE TORQUE ABOUT CRANKSHAFT (CRANKS HORIZONTAL), 100,000 IN LBS.

$$\text{*COUNTERBALANCE AT POLISHED ROD (LBS.)} = \frac{\text{MAX. C'BALANCE TORQUE}}{.5 \times \text{STROKE LENGTH}} + \text{STRUCTURAL UNBALANCE}$$

*MAX. C'BALANCE TORQUE (IN. LBS.) = (C'BALANCE AT POLISHED ROD – STR. UNBALANCE) X (.5 X STROKE LENGTH)

*THESE ARE CLOSE APPROXIMATIONS. FOR EXACT AMOUNTS USE TORQUE FACTOR FOR HORIZONTAL CRANK POSITION INSTEAD OF ".5 X STROKE LENGTH".

Fig. 2.47 (after Ref. 30)

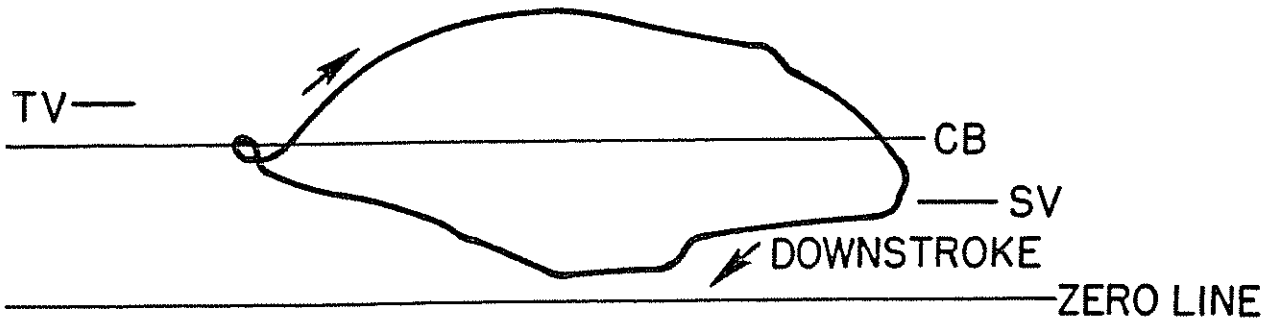


Figure for class problem below (Luling Field)

From Figure 2.43, four – 3 CRO counterweights are set at 30" (average) from the long ends of the cranks. In Figure 1 of Figure 2.47, a maximum counterbalance torque, "Q", of 555,800 in.-lb is obtained.

Column 10 values are equal to "Q" times column 2.

(8) The theoretical net torque in column 11 is equal to column 9 minus column 10.

(9) The net torque in column 12 considers an assumed structural efficiency. Positive values in column 11 are divided by 0.93; negative values are multiplied by 0.93.

(10) On Figure 2.46, an illustration is worked out to counterbalance the unit properly. New equated torque peaks and the counterweight locations to obtain these peaks can be determined.

(11) Peak polished rod load is the product of maximum "A" times dynamometer constant. Polished rod horsepower (PRHP) is obtained as shown on Figure 2.46, upper right hand corner.

CLASS PROBLEM

Above is an actual card as run in the Luling Field.

Well data (conventional unit):

2½" tubing
¾" rods
2¾" working barrel
86 rods
64" stroke
10.5 strokes in 21.6 seconds
C = 7075 lb/in.

Determine the following:

(1) Maximum load
(2) Minimum load
(3) Counterbalance
(4) PRHP
(5) Volumetric efficiency of the pump if 900 B/D of fluid is now being produced
(6) Peak torque

2.107 Factors influencing the shape of dynamometer cards

There are many factors which can cause a change in the basic shape of the theoretical dynamometer card. Among these are:

(1) Speed and pumping depth
(2) Fluid conditions
(3) Abnormal conditions at the pump
(4) Friction factors
(5) Pumping unit geometry

These factors can contribute singularly or collectively to a number of conditions which have been described variously as:

(1) Plunger undertravel
(2) Plunger overtravel
(3) Fluid leakage past traveling valve
(4) Fluid leakage past standing valve
(5) Fluid pound
(6) Gas pound
(7) Gas lock
(8) Restriction in the well
(9) Sticking plunger
(10) Excessive friction in the pumping system
(11) Synchronous pumping speeds
(12) Vibrations
(13) Abnormal load conditions

This section is devoted to the presentation of "typical" examples of each type of condition. The conditions are illustrated in the following figures:

(1) Plunger undertravel and overtravel—Figure 2.48.

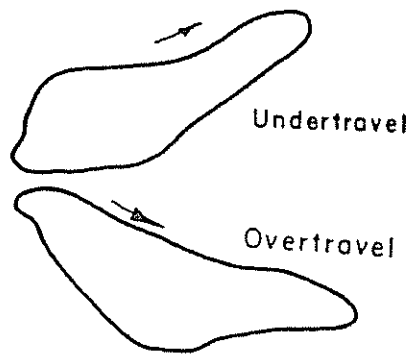


Fig. 2.48 (after Ref. 17)

(2) Fluid leakage past traveling valve—Figure 2.49.

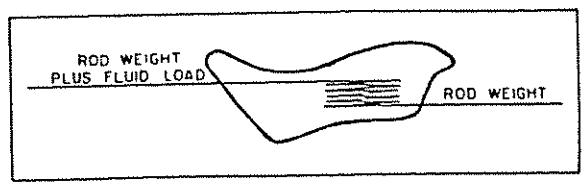


Fig. 2.49 (after Ref. 5)

(3) Fluid leakage past standing valve—Figure 2.50.

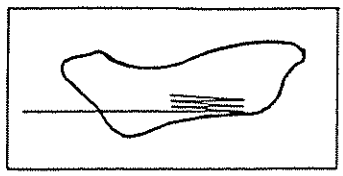


Fig. 2.50 (after Ref. 5)

(4) Fluid pound—Figure 2.51.

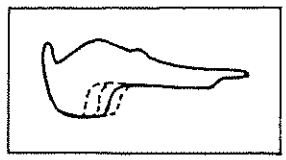


Fig. 2.51 (after Ref. 5)

(5) Gas pound—Figure 2.52.

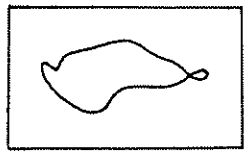


Fig. 2.52 (after Ref. 5)

(6) Gas lock—Figure 2.53.

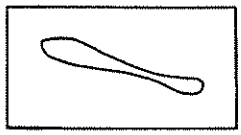


Fig. 2.53 (after Ref. 5)

(7) Restriction in the well—Figure 2.54.

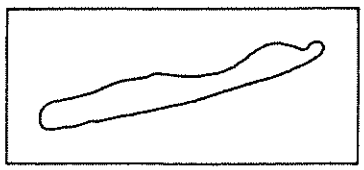


Fig. 2.54 (after Ref. 5)

(8) Sticking plunger—Figure 2.55.

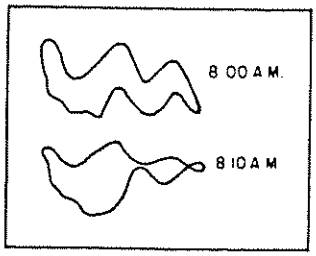


Fig. 2.55 (after Ref. 5)

(9) Excessive friction in the pumping system—Figure 2.56.

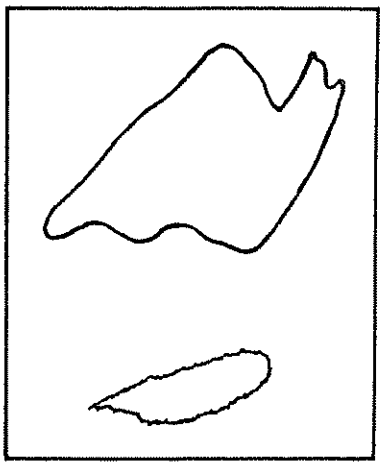


Fig. 2.56 (after Ref. 5)

(10) Vibrations—Figure 2.57.

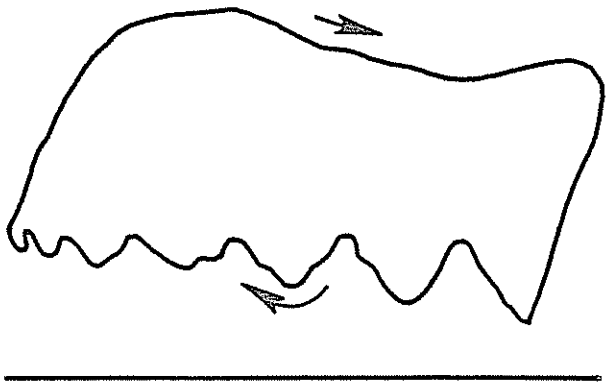
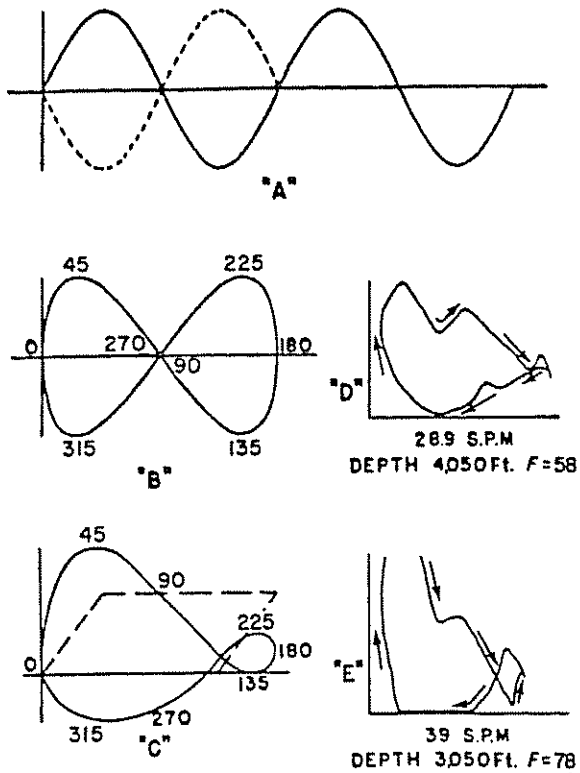


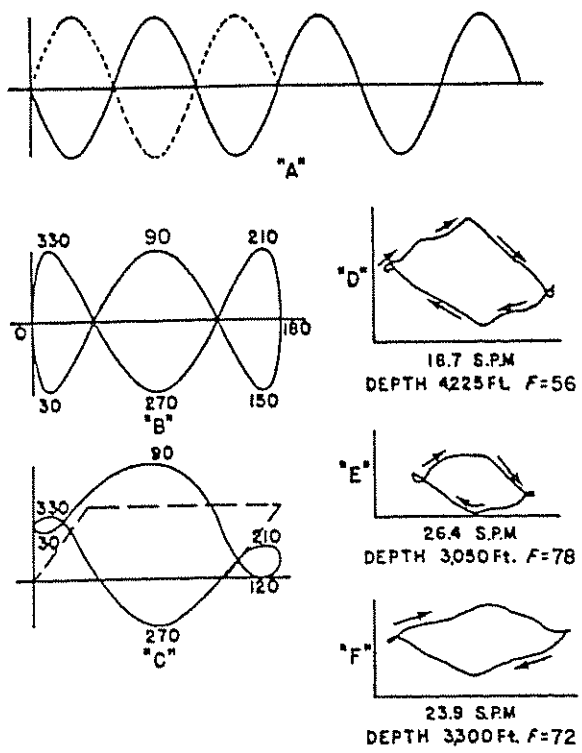
Fig. 2.57 (after Ref. 9)

(11) Synchronous pumping speeds—Figures 2.58, 2.59, 2.60, and 2.61.



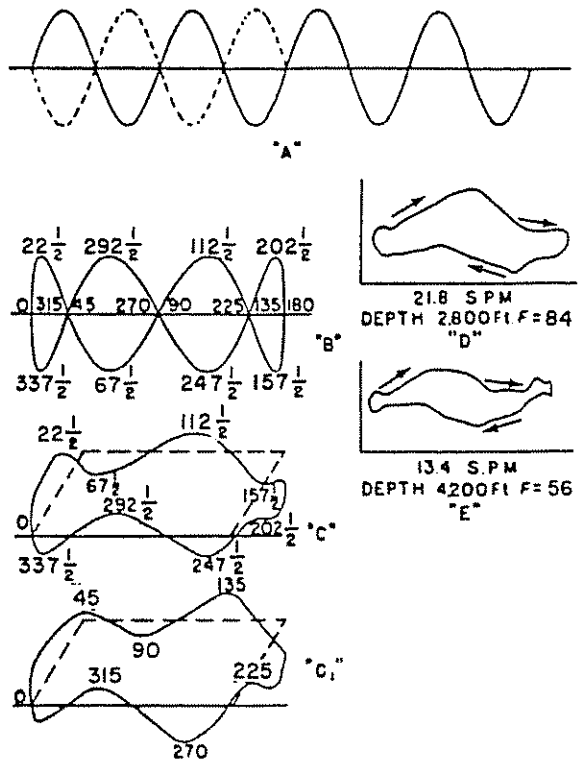
SECOND ORDER - TWO WAVES

Fig. 2.58 (after Ref. 21)



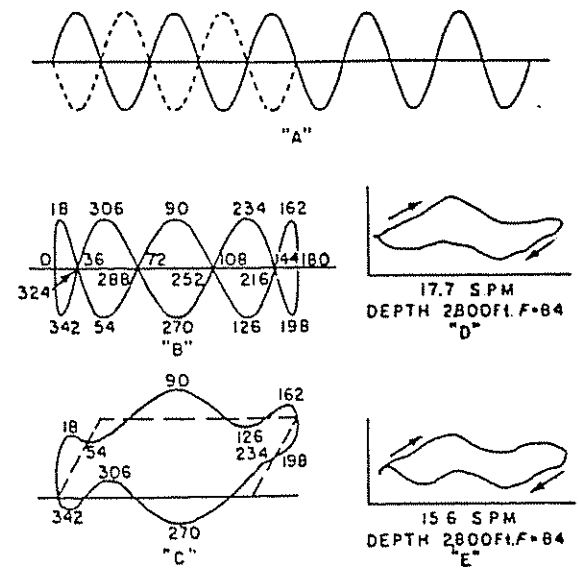
THIRD ORDER - THREE WAVES

Fig. 2.59 (after Ref. 21)

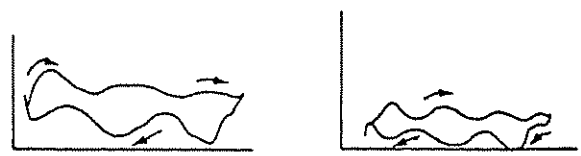


FOURTH ORDER - FOUR WAVES

Fig. 2.60 (after Ref. 21)



FIFTH ORDER - FIVE WAVES



SIXTH ORDER TENTH ORDER

Fig. 2.61 (after Ref. 21)

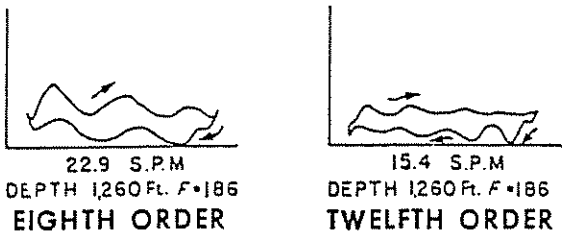


Fig. 2.61 (Continued)

2.108 Permissible load diagrams

As discussed in a previous section, all API pumping units are furnished with tables of torque factors and rod displacements applicable for a specific unit. These tables can be used to construct a permissible load diagram for that unit. "The Permissible Load on a pumping unit is the polished rod load necessary to develop a net torque at the gear reducer equal to its API rating with a fixed amount of counterbalance. This diagram can be constructed by calculating, for each 15° crank position, the permissible load, assuming the gear reducer is loaded to capacity."[22]

This diagram consists of two curves—one showing the upstroke load limit and one showing the downstroke load limit for all positions of the polished rod. To prevent gearbox overload, no load value of the dynamometer card should exceed its permissible load. Figure 2.62 shows a theoretical permissible load diagram.[22]

The permissible load can be determined from Equation 2.95 by solving for W, if it is assumed that the full gearbox rating is used.

Thus,

$$W_p = \text{permissible load} = \left(\frac{\text{Torque} + M \sin \Theta}{\overline{TF}}\right) + B$$

or,

$$W_p = \left(\frac{T_r + M \sin \Theta}{\overline{TF}}\right) + B \qquad (2.96)$$

where:

W_p = permissible load, lb
T_r = gearbox torque rating, in.-lb
M = maximum counterbalance moment
Θ = crank position, degrees
$\overline{TF}$ = torque factor corresponding to crank position Θ
B = structural unbalance, lb (either plus or minus)

When a permissible load diagram is constructed, the measured dynamometer card must fit between the upstroke and downstroke curves, as indicated by Figure 2.63, at all positions.

2.11 NON-SYNCHRONOUS PUMPING SPEEDS IN A BEAM AND SUCKER ROD SYSTEM

Many years ago, John Slonneger, an outstanding beam and sucker rod pumping scholar, proposed the idea that for any given depth well there was a series of pumping speeds detrimental to the rod string, while another series was less harmful. The series of pumping speeds supposedly damaging to the string was called "synchronous" pumping speeds, while the more beneficial pumping rates were called "non-synchronous" speeds. Synchronous pumping speeds were supposed to elevate the peak polished rod load abnormally, while non-synchronous speeds were thought to lower it.

Determination of the desirable, or *non-synchronous,* pumping speeds was made by calculating the natural frequency of the rod string (f), i.e. dividing the constant 237,000 by the depth of the well in feet and then dividing this constant by 1.5, 2.5, 3.5, 4.5, etc. The resulting pumping speeds were supposed to be less damaging to the rod string. On the other hand, the natural frequen-

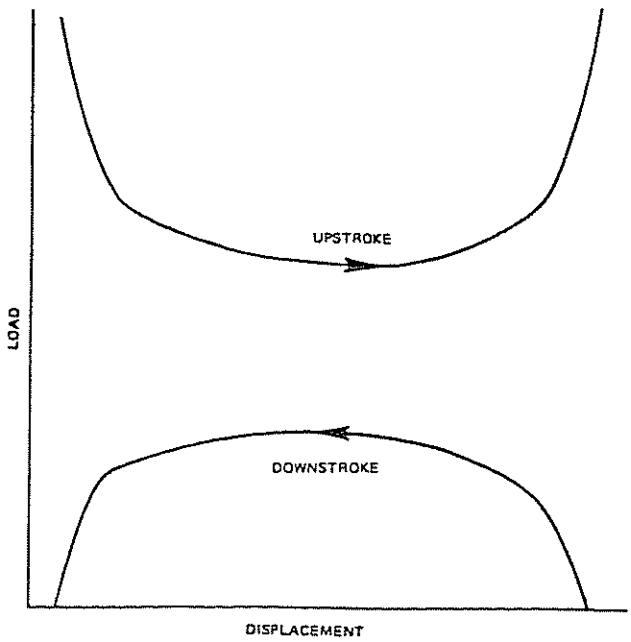


Fig. 2.62 Theoretical Permissible Load Diagram (after Ref. 22)

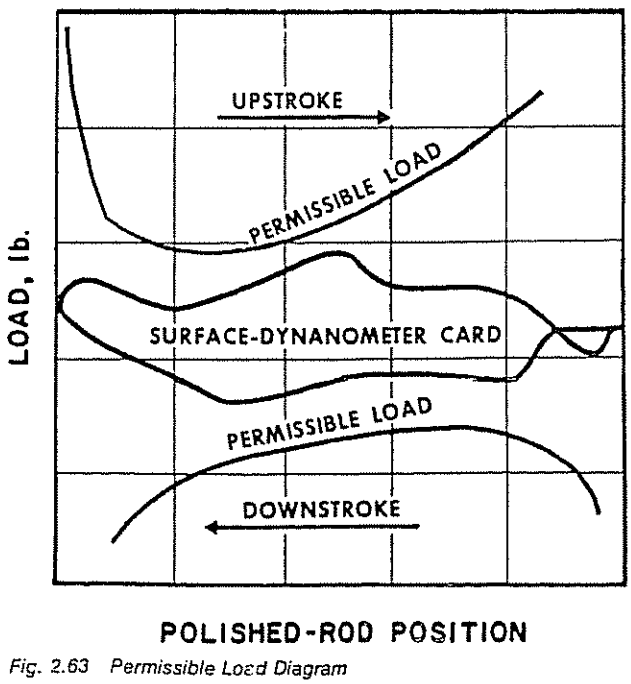


Fig. 2.63 Permissible Load Diagram

cy of the string (f) divided by 1.0, 2.0, 3.0, 4.0, etc. determined *synchronous* pumping speeds; these were thought to be more harmful to the rod string and were to be avoided wherever possible.

Theoretically, this probably makes sense. From a practical standpoint, there is some question about the need to design the installation to pump at a *non-synchronous* speed.

Perhaps the original concept was to avoid a pumping speed such that the maximum value of the stress wave, induced by the closing of the traveling valve, would arrive at the surface about the time the stretch was removed from the string and a second peak was occurring. It was probably reasoned that the addition of these two peak loads would cause an abnormally-high peak polished rod load, resulting in harmful pumping conditions for the rod string. As mechanically sound as this concept appears, it is difficult to visualize it in practice.

Some years ago the Sucker Rod Research Institute (SRI) did a study of various pumping wells where the peak polished rod load for both anchored and unanchored tubing was plotted vs. pumping speed. Figure 2.64 shows this curve for a 3,263-ft well. The natural frequency of the rod string (f) was calculated to be 72.63 C/M and, dividing that value successively by 3.5, 4.5, 5.5, 6.5, etc., the non-synchronous pumping speeds were determined and plotted as vertical lines on the continuous peak polished rod load curve of Figure 2.64. There is little choice between the synchronous and non-synchronous pumping speeds as regards peak polished rod load. As pumping speed increases, more polished rod work is being performed per unit of time, and the peak polished rod load should become greater. Still, from figure 2.64 (a rigorous pumping study) there is little to choose between synchronous and non-synchronous pumping speeds (at least for this particular application), as regards higher or lower peak polished rod loads.

Figure 2.65 shows the peak polished rod load vs. pumping speed for a 7,150-ft. well. The natural frequency of the rod string (f) would be 33.15 C/M; dividing this by 1.5, 2.5, 3.5, 4.5, etc. presumably gives the optimum or non-synchronous pumping speed. These desirable speeds are plotted as vertical lines on the peak polished rod load curve, and again, there is little

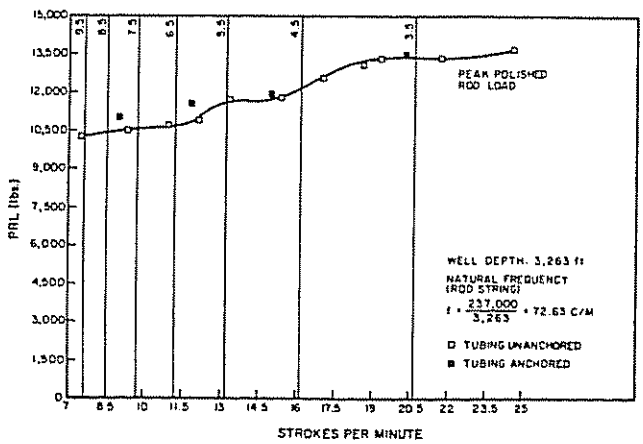


Fig. 2.64 Variation in Well Characteristics with Pumping Speed for Free and Anchored Tubing (after Ref. 23)

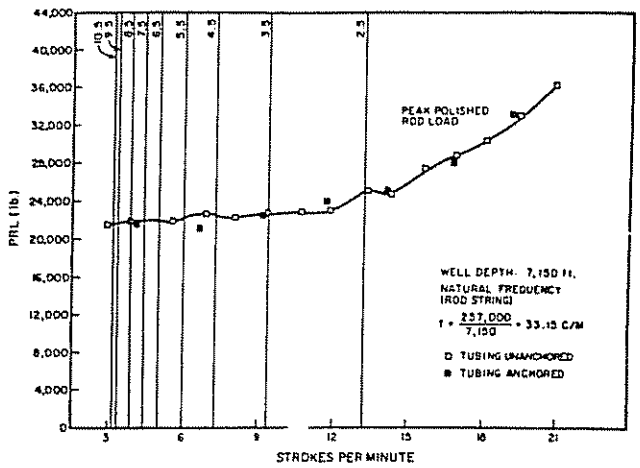


Fig. 2.65 Variation in Well Characteristics with Pumping Speed for Free and Anchored Tubing (after Ref. 23)

choice between synchronous and non-synchronous pumping speeds. The *synchronous* (undesirable) speeds would be, of course, half-way between two non-synchronous speeds. Thus, if these sucker rod research studies are correct, there is little benefit from the standpoint of peak polished rod load reduction in selecting a *non-synchronous* pumping speed over a *synchronous* pumping speed.

Possibly other applications would show a greater peak polished rod variation, but these two studies—with widely-varying well depths—show that little regard need be given to selecting a special pumping speed. Apparently an adequate design technique would be to select a rate that produced the desired fluid volume without violating any of the torsional or structural limitations of the system's components.

2.12 BEAM PUMPING GEOMETRY AND ITS EFFECT ON ROD AND UNIT LOADING AND PUMP TRAVEL

Though difficult to generalize, the geometry and kinematic design of the beam pumping unit has a significant effect upon its torsional, structural, rod loads, and bottomhole pump displacements.

Suppose two beam units of substantially different geometric design were pumping alternately, under identical conditions, over a stabilized well. There would probably be a significant difference between the unit performance in (1) peak torque, (2) structural and rod load, (3) load range, and (4) pump displacement. Since nothing was changed but unit geometry, differences in performance result simply from the difference in kinematic output of the two units.

Pumping units are different because of their geometry and counterbalance arrangement. The geometry of the unit is important because it controls the torque factor schedule and motion characteristics. To some degree, the torque-speed relationships of the prime mover also affect pumping motion.

Since practically all pumping unit geometries were of the same conventional type until the mid-1930's, little thought was given to the influence of geometry on beam pumping performance.

With the advent of the air-balanced unit, which employed a reversed or push-up geometry, observers

noted that under similar operating conditions torque, rod and structural loads, and pump displacements often changed with this new type of beam pumping configuration. In 1963, H. E. Gray showed mathematically that the kinematic arrangement of a beam unit greatly influenced its field performance.[31] Gray concluded by stating, "Pumping unit (kinematic) design exerts a significant influence on the polished rod loads, plunger stroke, and torque which are obtained during the operation of the sucker rod pumping system."

By solving the wave equation as it applied to sucker rod pumping, Gibbs was able to simulate accurately and mathematically beam and sucker rod pumping performance with any type of unit geometry operating under any given set of pumping conditions.[32] By inserting the six appropriate kinematic linkage dimensions (see Fig. 2.66) in the mathematical model, it is possible to compare beam pumping units of any geometry on an equivalent basis for any application.

Following is an example employing Gibbs' powerful predictive technique, comparing five different pumping unit geometries—A, B, C, D, and E—on a particular beam pumping application (see Fig. 2.67).[33]

These five units, each with a different geometry, were applied to a single pumping application as follows: fluid lift 6,000 ft, offbottom, with a 1½-in. plunger inside 2⅞-in. tubing, with each surface unit pumping 15 spm with a 100-in. stroke; specific gravity of the fluid, 1.0; shrinkage factor, 1.0; driving a standard API 86 rod string.

The computer, programmed with the Gibbs model, generated dynamometer cards for each of the five different geometries in this application. Peak and minimum polished rod loads were determined from each of the cards; a full API torque factor analysis was made on each, using the individual unit's unique torque factor schedule throughout one revolution of the crank.

Although the polished rod horsepower was essentially identical for all five geometries, significant differences existed in all other performance categories. Note the different dynamometer card profiles of each of the five units (Fig. 2.67).

For this particular application, comparing the performance of two of the units, A and E, Unit E was superior to unit A in the following categories:

(1) Reduced the peak rod and structural load by 9%
(2) Decreased peak in-balance torque by 41%
(3) Increased net plunger travel by 5%
(4) Reduced prime mover horsepower requirement by 33%
(5) Reduced peak rod stress by 9%
(6) Increased production by 5%

Improved pumping performance resulted from changing only one parameter—the geometry of the pumping unit.

Probably no single pumping unit geometry is superior in all performance categories for every application. Because of this, each geometry should be evaluated on a case-by-case basis similar to the analysis in Figure 2.67. These performance differences of three beam pumping units are illustrated in a qualitative manner in Table 2.17.

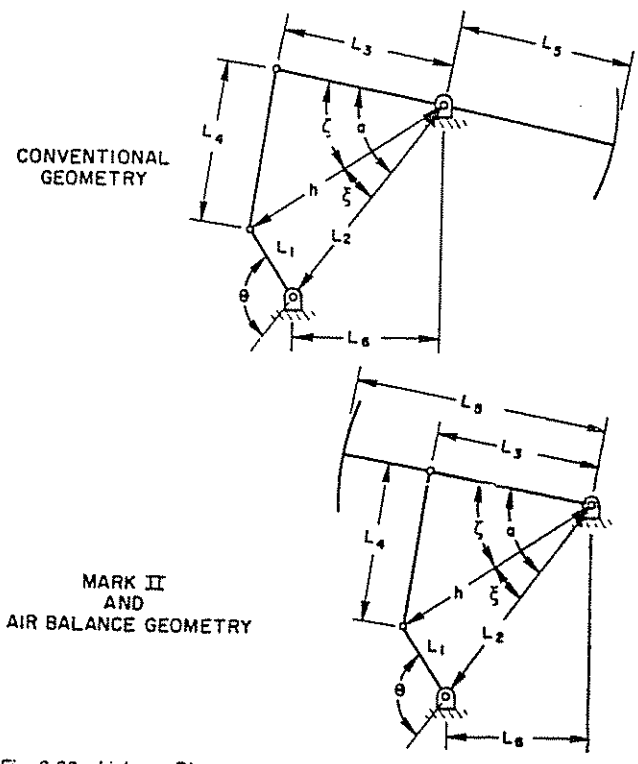


Fig. 2.66 Linkage Dimensions for Conventional and Mark II/Air Balance Geometries (after Ref. 23)

2.13 MODERN PREDICTIVE METHODS

In the mid-1950s, a group of major oil companies and equipment manufacturers organized the Sucker Rod Research Institute (SRI) to investigate the complex behavior of a conventional pumping unit driving an elastic rod string. An electrical analog computer simulated the sucker rod system.

By solving the wave equation as it applied to rod pumping, the computer generated synthetic surface and bottomhole pump dynamometer cards, making the following assumptions: (1) approximate conventional pumping unit geometry, (2) low slip prime movers, (3) full liquid fillage at the downhole pump with no gas interference or fluid pound, (4) no abnormal downhole friction or fluid acceleration effects, and (5) anchored tubing.

Data from these synthetic dynamometer cards were used to generate nondimensional design curves. These curves were then employed to (1) develop a hand calculation predictive method and (2) design a catalog of surface dynamometer cards, covering the entire pumping spectrum, in which the aforementioned five restrictive assumptions are included.

Later, much of the SRI technology was acquired by the American Petroleum Institute (API) and published in tabular form (API Bulletin 11L3), "Sucker Rod Pumping System Design Book." This bulletin pre-calculated a large number of design cases of practical application and tabulated them so an operator could size conventional pumping units quickly without resorting to hand calculations.

About this time, Dr. S. G. Gibbs was able to cast the sucker rod wave equation into a general mathematical model, solved by a digital computer.[32] Essentially, both SRI and Gibbs were able to simulate mathematically the complex field behavior of the sucker rod system. Additionally, Gibbs generalized and expanded the

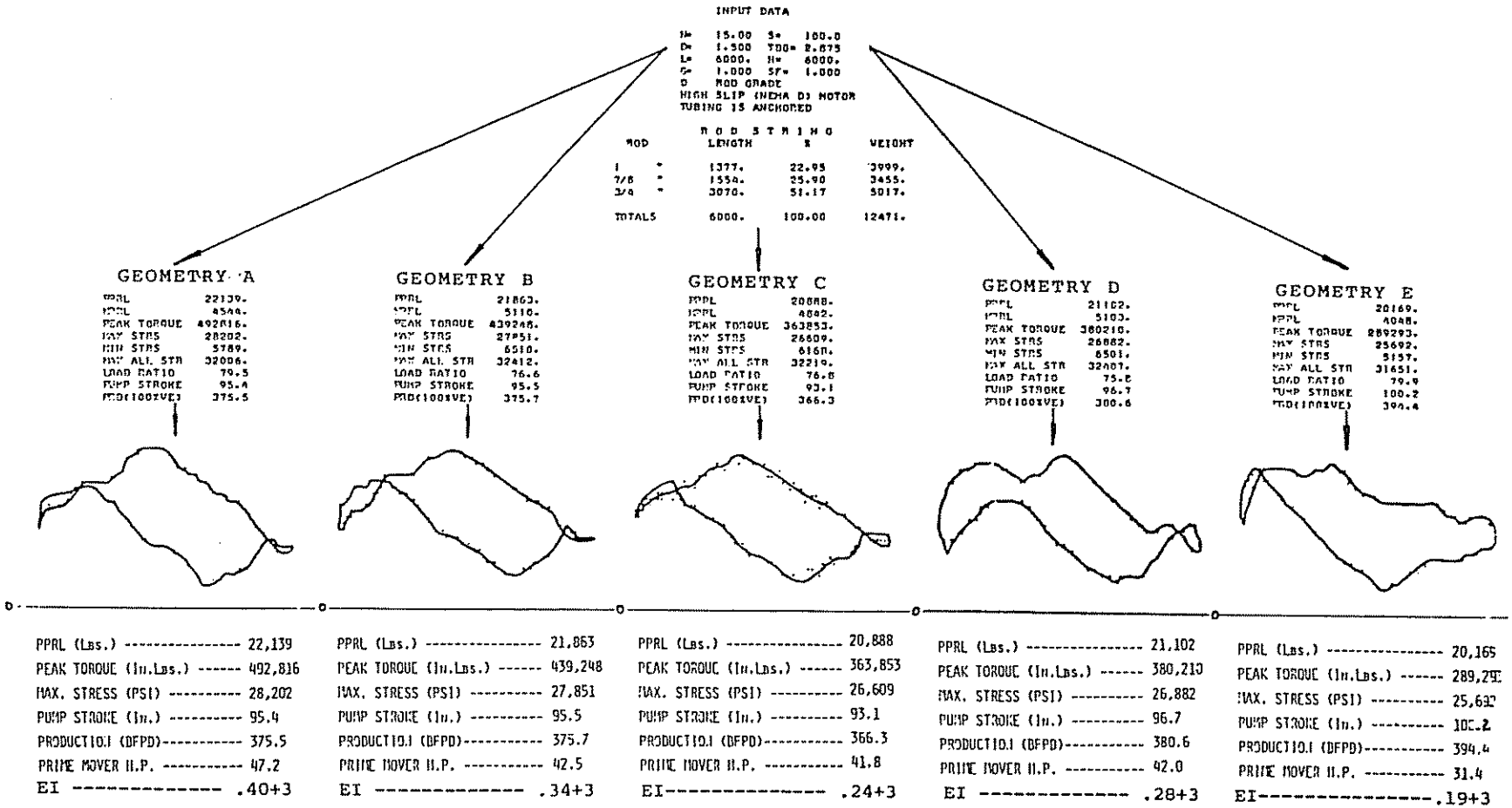


Fig. 2.67 Beam Pumping Unit Geometry and Its Relationship to Dynamometer Card Shape, Rod and Unit Loading, and Pump Travel (after Ref. 33)

TABLE 2.17 (after Ref. 23)
QUALITATIVE PERFORMANCE CHARACTERISTICS OF THREE DIFFERENT BEAM PUMPING UNIT TYPES

	Conventional unit	Air balance unit	Mark II unit
(1)	Highly efficient	Normally somewhat less efficient than other unit types	Highly efficient
(2)	High reliability factor due to simplicity of design	Most complicated of the unit types	Same reliability factor as conventional
(3)	Conventional unit cost used as reference	Occasionally higher cost than Conventional	For comparable application, often about the same cost as conventional
(4)	Portability more limited than for air balance unit	Greatest ease of portability. More compact	Normally, less portable than air balance unit
(5)	Counterbalance more difficult to adjust	Counterbalance easily adjustable	Counterbalance more difficult to adjust
(6)	Generally, widely-fluctuating torque loads	Torque peaks and range are often not as severe as on the conventional unit	Lowest and smoothest torque load with relative uniform torque system (Unitorque)
(7)	Impractical to mount on two-point suspension for minimal foundation and base movement	Impractical to mount on two-point suspension for minimal foundation and base movement	Can be mounted on two-point suspension for minimal foundation and base movement
(8)	Relatively higher power cost and larger prime mover requirement	Relatively higher power cost and larger prime mover requirement	Generally, lowest power cost and smallest prime mover requirement because of uniform torque (Unitorque) system
(9)	Generally, higher rod and structural loads	Generally, lower rod and structural loads than conventional units	Generally, lowest rod and structural loads
(10)	Normally the highest (relative) maximum pumping speed	Often slightly reduced maximum pumping speed compared to conventional unit	Often slightly reduced maximum pumping speed compared to conventional unit
(11)	Less "fill-time" for subsurface pump barrel	Less "fill-time" for subsurface pump barrel	Greater "fill-time" for the subsurface pump barrel
(12)	Normally a lesser net plunger travel per stroke	Normally a lesser net plunger travel per stroke	Normally the greatest net plunger travel per stroke
(13)	Speed reducer maintenance nominal due to widely fluctuating torque loads and relatively small "flywheel" effect of cranks	Speed reducer maintenance may be less than on the conventional unit	Normally less speed reducer maintenance required than other unit types due to relative uniform torque system and substantial "flywheel" effect of large cranks

basic technique to include conventional unit geometry and any type of surface pumping unit arrangement.

Gibbs further upgraded the mathematical model to consider virtually every major variable encountered in actual field pumping: any surface unit geometry, damping, bottomhole pump conditions (degree of gas or fluid pound), anchored or unanchored tubing, system inertia, and prime mover speed torque characteristics.

By contrast, the SRI (later API) technique assumed only an approximate conventional geometry, handling a rectangular bottomhole card along with the other restrictive assumptions outlined above.

Perhaps the most important advancement in the Gibbs technique over that of any other predictive method was in the determination of peak torque. In the older SRI-API method, peak torque was predicted using an approximation method that considered neither the conventional unit's full torque factor schedule nor its true kinematic polished rod motion*. This created a modest discrepancy in predicting peak and minimum polished rod loads and pump displacement, but with the possibility of a significant error in torque predictions.

On the other hand, Gibbs used the actual kinematic linkage dimensions of the pumping unit geometry in question, which rendered an exact simulation of the polished rod motion—so necessary to determine accurate load and pump displacement values. The profile of the surface dynamometer card for any pumping mode is a complex function of the kinematic arrangement of the particular pumping unit.

Additionally, Gibbs applied the unit's full torque factor schedule to the predicted dynamometer card which had been precisely developed using the unit's true linkage dimensions.

This approach enabled the prediction of torque values with greater accuracy than ever before because the predictive technique duplicated the method used to measure torque from dynamometer cards taken in the field.

When the Gibbs technique was modified to include the inertial forces and prime mover characteristics of the system, the predicted load and pump displacement figures were generally superior to those of any other method.

Following is a typical application example employing the Gibbs' method where the polished rod motion is developed by the actual kinematic linkage dimensions of the unit selected and the torque is determined by applying the full API torque factor method, i.e. API-11E.

*The motion assumed at the polished rod was sinusoidal plus 6% second harmonic.

EXAMPLE PROBLEM #15: SAMPLE DESIGN CALCULATION (AFTER REF. 34)

Data

Unit geometry:	Mark II	Plunger diameter:	2.25 in.
Pump depth:	4175 ft	Rod number:	86
Pumping speed:	10.7 spm	Stroke length:	120 in.

Solution

(1) Determine rod properties
From Table 2.7:

$W_r = 2.315$ lb/ft
$E_r = 0.656 \times 10^{-6}$ in./lb-ft
$F_c = 1.153$

(2) Determine structural unbalance
From Table 2.18:

$B = -3872$ lb

(3) Determine N/N_o' and W_f/SK

$$\frac{N}{N_o'} = \frac{N\,L}{245{,}000\,F_c} = \frac{10.7 \times 4175}{245{,}000 \times 1.153} = 0.1581$$

$$SK = \frac{S}{L\,E_r} = \frac{120.}{4175 \times 0.656 \times 10^{-6}} = 43{,}814.8 \text{ lb}$$

$$W_f = 0.34034\,D_p^2 L = 0.34034 \times (2.25)^2 \times 4175$$

$W_f = 7193.4$ lb

$$\frac{W_f}{SK} = \frac{7193.4}{43814.8} = 0.1642$$

(4) Calculate dimensionless prediction parameters
From double interpolation of tabulated values from Tables 2.20 and 2.21:

$G_1 =$	0.26380	$G_2 =$	0.14753
$G_3 =$	0.87762	$G_4 =$	0.18435
$G_5 =$	−0.00351	$G_6 =$	0.57359
$G_7 =$	0.21217	$G_8 =$	0.40725
$G_9 =$	0.70959		

(5) Calculate weight of rods in fluid

$W_2 = 0.871971\,W_r\,L = 0.871971 \times 2.315 \times 4175$
$W_2 = 8427.7$ lb

(6) Calculate maximum load

$W_{max} = W_2 + 0.99(SK)G_1$
$W_{max} = 8427.7 + (0.99 \times 43814.8 \times 0.26380)$
$W_{max} = 19{,}870$

(7) Calculate minimum load

$W_{min} = W_2 - 0.80(SK)G_2$
$W_{min} = 8427.7 - (0.80 \times 43814.8 \times 0.14753)$
$W_{min} = 3257$ lb = MPRL

(8) Calculate pump stroke and pump displacement

$S_p = S\,G_3 = 120 \times 0.87762 = 105.3$ in.
$PD = 0.1166\,N\,S_p\,D_p^2 = (0.1166)(10.7)(105.3)(2.25)^2 = 665$ B/D

(9) Calculate polished rod horsepower

$$PRHP = \frac{N\,S(SK)}{396{,}000}\,G_4 = \frac{10.7 \times 120 \times 43814.8}{396{,}000}(0.18435)$$

PRHP = 26.19 HP

(10) Calculate counterbalance moment

TABLE 2.18
(After Ref. 23)
AVERAGE STRUCTURAL UNBALANCES FOR THE MARK II

Stroke in.	Unbalance lb
54	−1198
64	−1420
74	−1755
86	−1838
100	−3492
120	−3872
144	−4345
168	−5122
192	−7160
216	−7450

$$\left(\frac{W_2 - B}{SK}\right) = \frac{8427.7 + 3872}{43814.8} = 0.28072$$

$$M_{CB} = S(SK)\left[G_5 + G_6\left(\frac{W_2 - B}{SK}\right)\right]$$

$M_{CB} = 120 \times 43814.8 \times (-0.00351 + 0.57359 \times 0.28072)$
$M_{CB} = 828{,}143$ in.-lb

(11) Calculate peak torque

$$PT = S(SK)\left[G_7 + \left(\frac{W_2 - B}{SK}\right)\right] \times G_8 - M_{CB}\,G_9$$

$PT = 120 \times 43814.8 \times (0.21217 + 0.28072) \times 0.40725 - 828{,}143 \times 0.70959$
$PT = 467{,}749$ in.-lb

(12) Calculate index of torsional effectiveness

$$ITE = 100\,\frac{63{,}025\,PRHP}{(PT)N} = \frac{100 \times 63{,}025 \times 26.19}{467{,}749 \times 10.7}$$

ITE = 32.98%

Note: The index of torsional effectiveness (ITE) is an absolute and accurate measurement of the ability of any beam-type pumping unit geometry (under normal conditions—handling a full barrel of incompressible fluid) to smooth out and idealize the net crankshaft torque load. The ITE (assuming any given torque-work load at the crankshaft of the pumping unit) is simply the ratio of the average net torque to the peak (in-balance) net torque, expressed as a percentage. The ITE is a valuable tool when comparing different pumping unit geometries over the same (or similar) wells, with the same stroke length, pumping speed, and downhole hardware.

The higher the ITE, the more effective is the unit geometry in converting the polished rod load into a smoother and more ideal torsional load. The formula for determining ITE for any sucker rod (beam) pumping unit geometry is:

$$ITE = \frac{63{,}025\,PRHP}{(PT_b)\,N\,(1 - e)}$$

where:

PT_b = In-balance peak torque at the crankshaft, in.-lb

TABLE 2.20 (After Ref. 34)
DIMENSIONLESS PARAMETERS FOR 3 AND 4-TAPER STRING
N/NO = 0.150

WF/SK	G1	G2	G3	G4	G5	G6	G7	G8	G9
.000	.03440	.07180	1.03200	.03730	−.00015	.47100	−.01020	.35300	.52500
.025	.07826	.07889	1.01186	.06370	−.00158	.50714	.01041	.30867	.40298
.050	.11634	.09941	.98761	.08750	−.00194	.51567	.03819	.29175	.35981
.075	.15153	.11229	.96519	.11031	−.00267	.53388	.07152	.29670	.37924
.100	.18209	.12346	.94165	.13198	−.00377	.54351	.10880	.31800	.44500
.125	.21342	.13472	.91569	.15205	−.00414	.55873	.14840	.35011	.54082
.150	.24317	.13736	.88662	.17065	−.00179	.56712	.18871	.38750	.65044
.175	.27176	.14078	.85902	.18808	.00069	.57269	.22812	.42464	.75758
.200	.29944	.14371	.83663	.20495	.00354	.57494	.26500	.45600	.84600
.225	.32625	.14748	.81605	.22042	.00567	.57874	.29774	.47605	.89941
.250	.35515	.14908	.79455	.23394	.00819	.58185	.32474	.47925	.90156
.275	.38413	.15046	.77221	.24664	.01090	.58314	.34436	.46008	.83618
.300	.40896	.15220	.74882	.25814	.01388	.58170	.35500	.41300	.68700
.325	.43356	.15171	.72676	.26863	.01687	.58181	.37386	.39588	.63170
.350	.45806	.15138	.70980	.27876	.01907	.58376	.39305	.38206	.58937
.375	.48649	.15091	.69311	.28879	.02353	.57719	.41347	.37446	.56961
.400	.51443	.14864	.67645	.29786	.05121	.51950	.43600	.37600	.58200
.425	.53413	.14796	.65865	.30670	.05617	.51708	.46984	.38222	.61352
.450	.55695	.14609	.63920	.31495	.06101	.51647	.51312	.39850	.68131
.475	.58326	.14401	.61911	.32238	.06787	.51473	.56784	.42678	.79194
.500	.60898	.14097	.59798	.32831	.07410	.51408	.63600	.46900	.95200

e = Loss in mechanical efficiency of the structural bearings.

From a study of several hundred dynagraph cards, the following ITE rating schedule was determined.[23]

TABLE 2.19
INDEX OF TORSIONAL EFFECTIVENESS

Under 15%	Very Poor
15 - 20%	Poor
20 - 25%	Fair
25 - 30%	Average
30 - 35%	Good
35 - 40%	Very good
40 - 45%	Excellent
45% & Above	Outstanding, exceptional

Proper use and understanding of the concept of the index of torsional effectiveness can help guide the operator to substantial cost savings in the application and operation of a beam and sucker rod pumping system.

(13) Calculate lift efficiency

$$HP_h = \frac{L(PD)}{135{,}771} = \frac{4175 \times 665}{135{,}771} = 20.45$$

$$LE = 100\frac{HP_n}{PRHP} = 100\frac{20.45}{26.19} = 78.1\%$$

(14) Calculate economic index

$$EI = 10^{-7}\left[\frac{W_{max}(PT)PRHP}{LE}\right]$$

$$EI = 10^{-7}\left[\frac{19870 \times 467{,}749 \times 26.19}{78.1}\right]$$

$$EI = 311.7 \text{ lb}^2\text{-in.-hp}$$

Note: Economic index (EI) gives the most economical pumping combination when considering torsional, structural, prime mover and lifting requirements. By selecting the lowest EI number, the most economical pumping system is defined.

(15) Calculate prime mover horsepower

From Figure 2.24:

$$E_S = 72\%$$

$$HP_{PMD} = 100\frac{PRHP}{E_S} = 100 \times \frac{26.19}{72.} = 36.38 \text{ HP}$$

(FOR NEMA "D" motor)

$$HP_{PMC} = 138\frac{PRHP}{E_S} = 138 \times \frac{26.19}{72.} = 50.2 \text{ HP}$$

(FOR NEMA "C" motor)

2.131 Predicting polished rod dynamometer card shapes

One of the most important tools in the prediction of beam and sucker rod pumping loads is the ability to compute a polished rod dynamometer card *profile* accurately. In predicting peak torque, it is not enough to be able to predict peak and minimum rod loads and their positions; it is also imperative that the shape of the dynagraph be computed as well.

Accurate prediction of dynamometer card shapes and load values depend upon (1) the solution of the classic wave equation, (2) a precise determination of the kinematic output of the surface pumping unit, and (3) the general configuration of the bottomhole dynamometer card. When these three variables are accurately known, a dynamometer card can be synthesized with considerable precision.

In Figure 2.68 are two dynamometer cards—one predicted and one measured later. Using the powerful Gibbs method, it can be seen that the predicted dynamometer card has a peak polished rod load of 19,830 lb and a minimum load of 2,860 lb.[32] When the well was later drilled and a dynamometer card taken under the same or similar pumping conditions, the peak polished

TABLE 2.21 (After Ref. 34)
DIMENSIONLESS PARAMETERS FOR 3 AND 4-TAPER STRING
N/NO = 0.175

WF/SK	G1	G2	G3	G4	G5	G6	G7	G8	G9
.000	.04679	.09454	1.04650	.04430	.00047	.46850	−.01023	.37012	.56762
.025	.09370	.09378	1.02235	.07191	−.00359	.51468	.02148	.34784	.51582
.050	.13337	.11616	.99622	.09604	−.00587	.53156	.05536	.33879	.50079
.075	.16428	.13219	.97052	.11925	−.00842	.54957	.09016	.34194	.51799
.100	.19597	.14560	.94689	.14158	−.01019	.56171	.12541	.35412	.55850
.125	.22742	.15409	.92524	.16253	−.00948	.56836	.16061	.37217	.61338
.150	.25542	.16221	.90412	.18198	−.01064	.57723	.19528	.39292	.67373
.175	.28629	.16658	.88200	.20009	−.00959	.58293	.22896	.41321	.73059
.200	.31598	.16882	.86145	.21715	−.00678	.58585	.26112	.42987	.77506
.225	.34455	.17427	.84558	.23455	−.00560	.58833	.29131	.43975	.79821
.250	.37086	.17621	.82974	.25097	−.00355	.59160	.31905	.43967	.79112
.275	.40234	.17666	.81389	.26631	−.00075	.59061	.34383	.42648	.74486
.300	.42726	.17803	.79653	.27952	.02443	.52575	.36519	.39700	.65050
.325	.44902	.17885	.77780	.29033	.02865	.52375	.39010	.38735	.62105
.350	.47280	.17893	.75759	.29991	.03334	.52346	.41561	.38127	.60695
.375	.50023	.17673	.73425	.30681	.04001	.52062	.44222	.38084	.61553
.400	.52459	.17711	.70937	.31216	.04501	.52084	.47044	.38812	.65412
.425	.55372	.17238	.68399	.31775	.05317	.51823	.50753	.39951	.70223
.450	.57984	.16972	.65710	.32337	.05995	.51756	.55177	.41896	.77647
.475	.61033	.16856	.62837	.32801	.06727	.51668	.60456	.44779	.88045
.500	.63803	.17005	.59692	.33106	.07310	.51691	.66731	.48731	1.01781

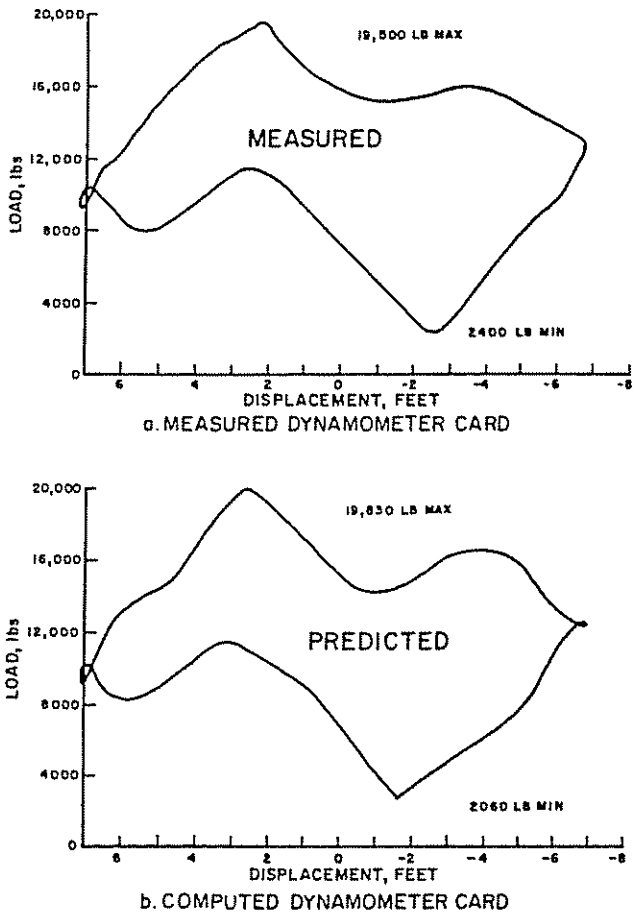


Fig. 2.68 (after Ref. 23)

rod load measured 19,500 lb and the minimum polished rod load measured 2,400 lb, for a very modest discrepancy between predicted and measured values. Visual inspection shows the predicted card to be extremely close, in profile, to the eventually-measured card. If a standard, full 360° torque factor analysis were made of these two cards, the in-balance peak torque would not vary by more than 2-3%. Consequently, when predicting peak torque, an accurate representation of the dynamometer card profile must be computed along with peak and minimum rod load values and their positions.

In Figure 2.69, two other comparative dynamometer cards are shown—one predicted and one measured. The synthetic card had a peak polished rod load prediction of 22,100 lb, and a minimum predicted polished rod load of 12,900 lbs. When the well was eventually drilled and a dynamometer card taken, the peak polished rod load actually measured 21,800 lb, while the minimum load measured 13,300 lb. Production was predicted at 195 B/D and eventually measured 200 B/D. If a standard full torque factor analysis is made of both of these cards, the predicted peak torque is seen to be almost identical to the peak torque value eventually measured.

Without knowing the exact kinematic output of the

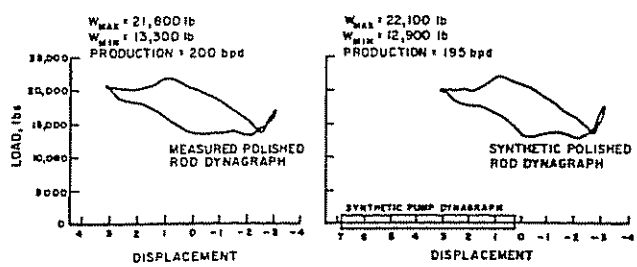


Fig. 2.69 (after Ref. 23)

surface equipment, assuming a particular bottomhole card profile, and using the classic solution of the wave equation, it is difficult to predict peak torque in a precise manner. It is important to recognize that not only the peak and minimum values of the dynamometer card must be accurately predicted, but an accurate prediction of the card profile must also be generated by the means described above.

2.14 THE DIAGNOSTIC TECHNIQUE

As early as the mid-1930's, it was recognized that polished rod dynamometer cards did not afford complete information about the performance of the beam and sucker rod pumping system. Though this method was particularly valuable for determining rod, structural, and torque loads on the surface unit and prime mover, bottomhole pump information still had to be inferred from visual inspection of the dynagraph.

In shallow to medium depth wells, intepretation of the surface dynamometer card was reasonably effective in inferring bottomhole pump performance. In deeper wells, however, the complex nature of the elastic rod string made determination of bottomhole pump performance by observation of the surface card virtually impossible.

Recognizing this fact, a bottomhole pump dynamometer was developed that was run in on the rods and afforded a fairly accurate representation of the load-displacement relationship at the bottomhole pump. Obviously, this type of bottomhole dynamometer, though furnishing valuable information, was cumbersome, expensive, and impractical to operate since the rod string had to be pulled to cut each individual dynamometer card. As valuable as the instrument was in determining bottomhole pump performance, a more practical method of obtaining this information was needed.

In the early 1960s, Dr. S. G. Gibbs of Shell Research reasoned that the sucker rod string acting as a "transmission line" between the bottomhole pump (transmitter) and the surface dynagraph (receiver) was in effect a reversible system. Gibbs reasoned that if a given bottomhole dynagraph configuration could be transmitted upward to the surface resulting in a "scrambled" surface dynagraph, why couldn't this process work in reverse?

Thus, by cutting a precise surface dynamometer card and knowing most of the important parameters involved (depth, pumping unit geometry, pumping mode, downhole hardware, etc.) he reapplied the solution of the wave equation to unscramble the complex surface pattern, thereby synthesizing a precise and reliable bottomhole dynamometer card shape. The ability to synthesize this bottomhole dynamometer card and interpret its various configurations became the basis of the *diagnostic technique.*

Since harmonics were not directly involved in the profile of the bottomhole card, this new technique reduced the dynagraph spectrum to a relatively few typical card shapes that could be easily and accurately interpreted by a competent observer. With an accurate load displacement diagram (dynagraph) of the bottomhole pump as well as a precise surface dynamometer card, for any given application practically all of the information necessary to optimize a beam and sucker rod system would then be available.

Following (Fig. 2.70) are two series of bottomhole dynagraphs—one group (on the left) with anchored tubing and a second group (on the right) with unanchored tubing. These are generalized representations of synthetic bottomhole cards, with the pumping system experiencing some of the more common bottomhole problems.

2.15 INERTIAL TORQUE IN A BEAM AND SUCKER ROD PUMPING SYSTEM

If the power driving a correctly balanced beam and sucker rod pumping system were suddenly cut off, the unit would probably not stop immediately but would coast for at least a portion of the crank cycle. This is because the kinetic energy stored in the moving parts of the system is being given up as inertial torque at the crankshaft. This inertial torque is a function of the total amounts of the kinetic energies stored (or given up) in the unit's moving parts during the crank cycle. As the power is cut off and the unit components come to rest, the amount of inertial torque work given up is proportional to the sum of the individual polar moments of inertia of all the moving components times their change in angular velocity, across a given time interval.

In order that the law of conservation of energy be satisfied, individual (instantaneous) inertial torques for each moving component must sum to zero each revolution of the crank. Obviously, then, the total inertial torque from all components must also sum to zero each crank cycle. When the pumping unit is decelerating under load, part of this intertial energy feeds back into the system; when the system is accelerating as the load decreases, energy is being stored in the moving parts to be given up later in the cycle.

Another item in the consideration of inertial torque is the complex nature of the loading in an elastic rod string. When these irregular, fluctuating harmonic stress waves are felt at the surface, the unit tries to respond, attempting to speed up when the harmonic wave has a low value, while slowing down when it is high. Thus, the inertial torque is considerably more complex than if a differential, concentrated mass type polished rod load were applied to the pumping unit.

The change of angular velocity of the system is sensitive to changes in torque load due to the harmonic stress waves in the rod string and to the speed-torque relationship of the prime mover. Obviously, there is a complex relationship between the change of velocity of the individual components during the crank cycle, the speed-torque relationship of the prime mover, and the involved harmonic stress wave pattern at the polished rod.

In the ordinary pumping system, three principal types of inertial torque exist: (1) the inertial torque resulting from all of the rotating parts between the crankshaft up to and including the prime mover (this is often referred to as *driving* rotary inertial torque), (2) the inertial torque developed by the rotating members between the crankshaft and the polished rod, referred to as *driven* rotary inertial torque, and (3) articulating inertial torque resulting from the oscillation of the

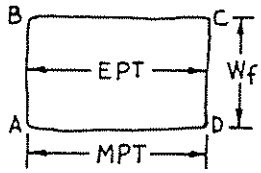
TYPICAL COMPUTER PUMP CARDS

These cards represent typical computed pump dynagraphs, illustrating different pumping conditions and malfunctions of downhole equipment.

MPT = Maximum Plunger Travel EPT = Effective Plunger Travel

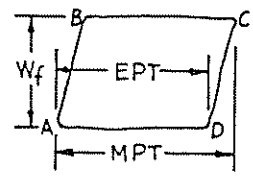
W_f = Differential Fluid Load on Plunger

WITH TUBING ANCHOR



Normal Pumping-Full liquid stroke (no gas). All pump components functioning properly. With tubing anchored, EPT= MPT, pump performance indices all equal to 1.00. Without anchor EPT < MPT, pump performance indices equal but less than 1.00. More well fluid available.

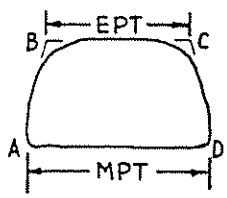
WITHOUT TUBING ANCHOR



The successive steps in the recording of such a pump operation are:

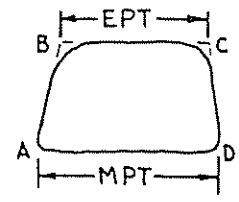
1. At the start of the upstroke (point A), traveling valve closes.
2. From A to B, the fluid load is picked up by rods as gas in clearance space expands from static tubing pressure (P_t) to pump intake pressure (PIP).
3. Standing valve opens at B, allowing fluid to enter pump when pressure in pump drops below intake pressure (PIP).
4. From B to C, fluid load is carried by rods as fluid is drawn into pump.
5. At C, standing valve closes as plunger starts down, traveling valve remains closed.
6. From C to D, gas in the pump (if present) is compressed and fluid load is transferred from rods to tubing.
7. At D, pump discharge pressure (Pd) equals static tubing pressure (P_t), and traveling valve opens.
8. From D to A, fluid is displaced through traveling valve into tubing.

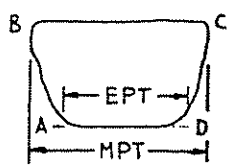
WITH TUBING ANCHOR



Leaking traveling valve or excessive plunger slippage causes delay in picking up fluid load from A to B and premature unloading from C to D, (standing valve closes during upstroke).

WITHOUT TUBING ANCHOR





Leaking standing valve causes premature loading of rods from A to B, and delay in unloading from C to D, (traveling valve closes during downstroke).

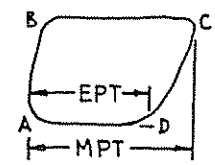
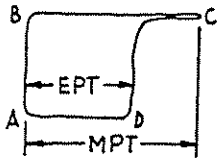
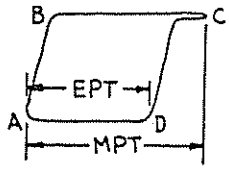
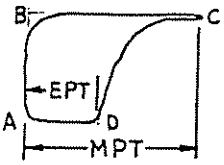


Fig. 2.70 (after Ref. 23)

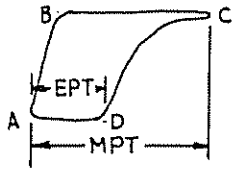


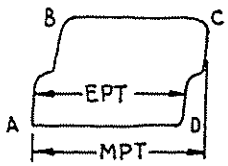
Severe fluid pound, well being pumped off. Pump components functioning properly.



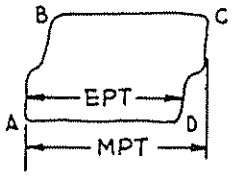


Gas interference causing loss of EPT. Pump components functioning properly. Pump performance indices of little value. Unstable well conditions--EPT different length every stroke.



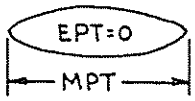


Malfunctioning tubing anchor, or partial stuck tubing.





Worn out pump. No tubing movement in either instance.

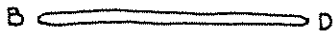


Practically any combination of the malfunctions shown above can exist in any well. The effects of these malfunctions will be superimposed on one another.

SPECIAL CONDITIONS



Gas Locked Pump---Both valves remain closed because static tubing pressure (P_t) > pump discharge pressure > (Pd) pump intake pressure (PIP).



Agitating Well---Both valves remain open because static tubing pressure (Pt) ≦ pump discharge pressure ≧ (Pd) pump intake pressure (PIP). Also deep rod part.

Fig. 2.70 (cont.)

walking beam, pitmans, horsehead, etc. These three principal inertial torques can be either plus or minus, and during one pumping cycle the algebraic sum of the instantaneous inertial torque moments must be zero.

Though it is difficult to generalize, when the load is applied and the system slows down, the driving rotary inertial torque adds to the net torque furnished by the prime mover. Across this same slow-down interval, the driven rotary inertial torque reduces the net torque delivered by the prime mover. Consequently, when these two rotary inertial torques are considered, the driven must exceed the driving torque in order that a resultant negative inertial torque moment is created which may help reduce normal net prime mover torque. If the driving rotary inertial torque exceeds the driven then the inertial torque will add to the net prime mover torque.

Often, articulating inertial torque has little bearing on prime mover (or net) torque peaks; for normally articulating torque values, whether negative or positive, occur over top and bottom of the stroke rather than at mid up and downstrokes where peak prime mover net torque normally occurs.

In using rotary inertial torque to minimize prime

mover net torque, another problem arises. It is necessary to develop a substantial negative torque moment and to position it in the very narrow area directly beneath the net prime mover torque peaks. Reduction of net torque any other place in the cycle is of little benefit in reducing prime mover torque. Consequently, to effect beneficial, negative, inertial torque moments to help reduce peak net prime mover torque: (1) the driven rotary inertial torque must exceed the driving rotary inertial torque, and (2) the application of the resulting negative inertial torque moment must fall directly beneath both the up and downstroke peak net prime mover torques. To develop a beneficial negative inertial torque (because of the complexity of the harmonic rod stresses and speed-torque characteristics of the prime mover), positioning these beneficial torques at the proper place in the cycle often becomes difficult to accomplish by simply changing pumping mode and/or prime mover speed—torque characteristics.

Another problem in attempting to reduce net prime mover torque by inertial means is to reduce driving torque by using smaller sheaves, etc., so it will be of less value than driven torque and will make the system more sensitive to the placement of the negative inertial torque moment at or near the peak net prime mover torque. On the other hand, the driven torque must be relatively large to create a significantly sizeable negative inertial torque moment. When this occurs, the entire system is less sensitive to delivering negative inertial torque at the two proper instants in the crank cycle.

The diagram in Figure 2.71 shows a typical pumping application and how inertial torque may affect net or prime mover torque. The figure illustrates the effect of inertial torque on the net (crankshaft) torque pattern of a beam pumping unit operating over a typical well. Positive inertial torque values add to net crankshaft torque, while negative values reduce it. The inertial torque was rigorously developed by weighing each moving component of the sucker rod system, computing each individual moment of inertia, and determining the component speed variation and its inertial torque.

The nature of the rod string with its complex harmonic stress waves may, if the pumping characteristics are in the correct mode, develop inertial forces that can reduce peak net torque. In other instances, inertial torque does little to reduce crankshaft torque and can occasionally actually increase its peak net value. In this example, inertial torque slightly increases peak net torque at mid-downstroke, while having little effect at mid-upstroke.

To further illustrate the complexity of this system, Table 2.22 and Figure 2.72 show an inertial study made on another typical pumping unit application, showing what part the various inertial torques played in modifying net prime mover torque peaks. In this study, the algebraic sum of the driven and driving rotary torques was used rather than their individual values. The algebraic sum of the two rotary inertias (driving and driven) plus the articulating inertia is designated total inertial torque.

Column 4 in Table 2.22 shows the net torque disregarding inertial forces, while Column 8 shows the net torque including inertial forces. In this example, the

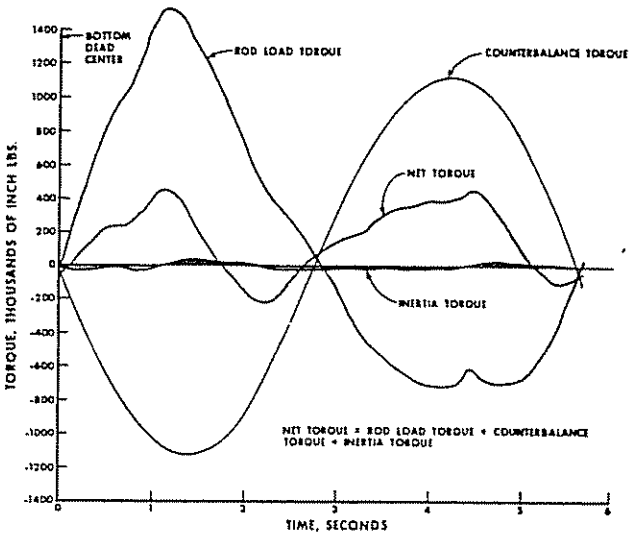


Fig. 2.71 (after Ref. 23) The Effect of Inertial Forces on the Net Torque Peaks of a Typical Beam Pumping Unit

peak in-balance net torque, momentarily forgetting inertial torque, is 478,600 in.-lb, while the peak in-balance net torque considering inertia is 441,900 in.-lb. This is a reduction of some 8% in peak torque. This is attributable to consideration of inertial torques. These values are plotted in curve form in Figure 2.72, and inertial torque effect on peak torque can be readily observed in this typical example.

2.151 A performance comparison between two beam and sucker rod pumping units of dissimilar geometry considering the influence of inertial torque

A sophisticated and precise comparison was recently made between two beam and sucker rod geometries for the same application by using a computer and a powerful mathematical model. This mathematical relationship (GIBBS) using a computerized solution of the wave equation probably simulates beam, rod, and pump motion more accurately than any other type of analytical device.

The comparisons in Examples 16 and 17 were made by Dr. S. G. Gibbs of the NABLA Corporation (p. 92). These comparisons are unique, because the solution of the wave equation was used and because the motion of the pumping unit and sucker rod string was developed by using the true kinematics of the pumping unit geometries and the true inertias involved.

In Example 16, unit A used an API 640 transmission with a 144-in. stroke. Unit B was equipped with the same transmission and stroke length. Both of these units were powered by 75 HP NEMA "D" motors, driving an API No. 86 rod string with a 1½-in. plunger and pumping at a speed of 11.4 spm in a 7000-ft well. The bottomhole card had a 20% gas pound, making the volumetric efficiency approximately 80%. Using the unit's own kinematic dimensions and the same bottomhole dynamometer card for both units, equivalent (comparative) surface cards were then synthesized.

Determining the moments of inertia of all the moving parts of the beam and sucker rod system and their

TABLE 2.22 (AFTER REF. 23)
TORQUE HISTORY—TORQUE (1000 in-lbs.)

Pos.	Rod torque	C. B. torque	Net torque disr. inert.	Rotary inert. torque	Art. inert. torque	Total inert. torque	Net torque inc. total inert.	S.P.M.
0°	78.3	0	+78.3	−6.8	8.2	+1.4	79.6	11.3
10°	314.9	−121.3	+193.6	−23.7	28.8	+5.1	198.7	11.18
20°	483.7	−239	+244.7	−36.1	35.5	−0.6	244.1	10.85
30°	695.7	−349.4	+346.3	−44.7	30.5	−14.2	332	10.39
40°	961.8	−449.3	+512.5	−58.6	18.4	−40.2	472.2	9.76
50°	1021	−535.4	+485.6	−59.3	4.2	−55.1	430.4	8.95
60°	1083.8	−605.3	+478.5	−22.3	5.2	−17.1	461.4	8.33
70°	892.2	−656.8	+235.4	16.9	18.9	+35.8	271.2	8.33
80°	954.1	−688.3	+265.8	21.5	15.1	+36.6	302.2	8.62
90°	933.5	−699	+234.5	13.3	−1.7	+11.6	246.1	8.85
100°	986.8	−688.3	+298.5	10	−11.3	−1.3	297	8.99
110°	876.8	−656.8	+220.0	13.1	−13.5	−0.4	219.6	9.13
120°	737.2	−605.3	+131.9	27.6	−16.1	+11.5	143.4	9.37
130°	564.9	−535.4	+29.5	50.2	−19	+31.2	60.6	9.84
140°	378.3	−449.3	−71.0	59.6	−19.5	+40.1	−30.9	10.42
150°	260.4	−349.5	−89.1	49.8	−17.3	+32.5	−56.5	11.01
160°	142.1	−239	−96.9	30.2	−10.9	+19.3	−77.7	11.41
170°	31.6	−121.3	−89.7	22	−2.5	+19.5	−70.2	11.53
180°	−61.4	0	−61.4	6	4.7	+10.7	−50.7	11.71
190°	−144.1	121.4	−22.7	−1.6	9.1	+7.5	−15.2	11.74
200°	−211.1	239.1	+28.0	−14.2	12.5	−1.7	26.3	11.66
210°	−267.1	349.5	+82.4	−26.5	15.9	−13.7	71.7	11.45
220°	−329.8	449.4	+119.6	−30.7	16.9	−13.8	105.7	11.15
230°	−366.8	535.5	+168.7	−26.3	15.2	−11.1	157.6	10.84
240°	−422.9	605.4	+182.5	−23.5	15.2	−8.3	174.1	10.58
250°	−356.5	656.9	+300.4	−32.2	17.5	−14.7	285.7	10.26
260°	−243.7	688.4	+444.7	−48.2	15.1	−33.1	411.6	9.78
270°	−319.4	699.1	+379.7	−49.8	9.8	−40.0	339.6	9.13
280°	−409	688.4	+279.4	−16.8	18.4	+1.6	281	8.65
290°	−603.8	656.9	+53.1	32.4	31.9	+64.3	117.3	8.75
300°	−602.1	605.4	+3.3	56.8	20.9	+77.7	81	9.33
310°	−525.2	535.5	+10.3	47.9	−8.7	+39.2	49.5	9.94
320°	−449.9	449.4	−0.5	30.3	−34	+3.7	−4.4	10.38
330°	−345.1	349.6	+4.5	26.1	−43.2	−17.1	−12.7	10.67
340°	−280.9	239.1	−41.8	28.4	−37.9	−9.5	−51.4	10.97
350°	−144.5	121.4	−23.1	19.6	−18.2	+1.4	−21.8	11.22

SUMMARY OF GEARBOX PERFORMANCE

	Existing	In balance
Peak torque (1000 in-lbs.):	472.2	448.3
Counterbalance (1000 in-lbs.):	699	735.9
Percent of gearbox rating:	103.5	98.3

Note: Torques are shown each 10 degrees of crank rotation starting from zero which is vertical position nearest bottom of stroke. In balance computations assume inertia effects will be relatively unchanged.

instantaneous angular velocities, six inertial torques were then correctly applied to the dynamics of each unit. The six initial torques included rotary driving and driven (positive and negative), and articulating (positive and negative).

In Example 16, unit A has an in-balance peak net torque of 580,800 in.-lb.; unit B has a peak in-balance net torque of 250,200 in.-lb. Inertial torques were considered in both units. The speed variation for unit A is 17.6% and for unit B is 5.1%.

The cyclic load factor is directly proportional to the nameplate horsepower requirement of the prime mover. In unit A, it is approximately 2.4, and in unit B, about 1.1.

Ideally, power should be delivered by the prime mover to the pumping system at a uniform rate with the lowest peak power and the lowest power range possible. In this example, the peak power for unit A is 102.9 HP, while it is only 54.4 HP for unit B. The nameplate horsepower required for unit A is 63.7, but only 34 HP is required for unit B. The load range of the rod string is 12,843 for unit B, and 14,256 for unit A.

An important parameter is regenerative power. For unit A, it is −87.7 HP, while for unit B it is zero. Regenerative, or backdriven, power indicates an undesirable negative torque condition which increases peak torque and torque range, and tends to increase prime mover thermal losses as well. Finally, the peak polished rod

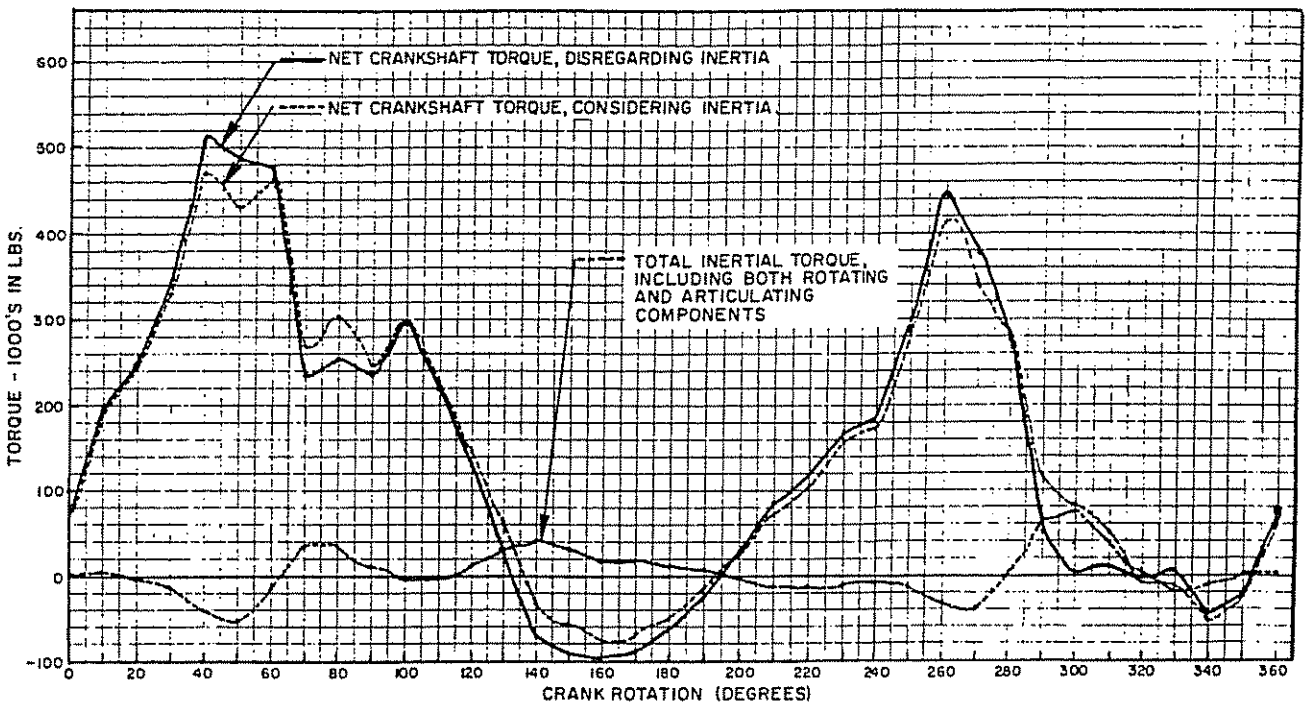


Fig. 2.72 (after Ref. 23)

load for unit A is 22,559 lb, while for unit B it is 20,886 lb.

In Example 17, a second comparison was made between unit A and unit B. Both machines are 640D-304-144 units, powered by 75 HP NEMA "D" motors, driving API No. 86 rod strings with 2.75 in. plungers at 9403 ft. The pumping speed for both units is 12.6 spm.

Again, inertia, both rotating and articulating, is considered. As in Example 16, both the surface and subsurface dynamometer cards are shown along with the permissible load diagrams for both units; a volumetric efficiency of 100% is also assumed.

In Example 16, identical bottomhole dynamometer cards were assumed for both units, along with a volumetric efficiency of 80%. In Example 17, regular surface cards were assumed for both units, which resulted in a greater net plunger travel for unit B.

In this application, unit A has a 19% speed variation, while unit B has a 9.2% speed variation. Again, the cyclic load factor for unit A is in excess of 2.0, and for unit B is about 1.2.

The peak power for unit A is 116.3 HP, but only 82.9 HP for unit B. The regenerative power (backdriven) is −73.7 for unit A and only −8.4 for unit B.

The nameplate horsepower required for unit A is 88.4 vs. 50.7 for unit B. Unit A has a peak polished rod load of approximately 20,140 lb, while unit B has only 18,527 lb.

The peak in-balance torque for unit A is 581,000 in.-lb, while it is only 393,400 in.-lb for unit B. Furthermore, the pump capacities of units A and B, respectively, are 633 and 662.7 B/D.

The index of torsional effectiveness (ITE) for units A and B is 28% and 43%, respectively. This index indicates the ability of the pumping unit geometry to smooth out and reduce peak torque for this particular application.

These two examples are precise comparative studies, which employ the most advanced predictive techniques available, while considering inertial torques, both rotating and articulating. Bottomhole and surface cards are shown for both examples, as are the comparative permissible load diagrams.

In comparing the relative performance characteristics of these two beam pumping units, nothing was changed except the geometry of the units themselves.

NOMENCLATURE

A_p = cross-sectional area of pump plunger, sq in
A_r = rod average cross-sectional area, sq in
A_t = cross-sectional area of metal in tubing, sq in
A_{top} = cross-sectional area of top section of rods, sq in
c/p = crank-to-pitman ratio
B = structural unbalance, lbs
C_i = "ideal" counterbalance effect, lbs
CLF = cyclic load factor
C_s = counterbalance effect from structural imbalance of pumping unit, lbs
C_t = total counterbalance effect at the polished rod, lbs
C_w = counterbalance effect produced by counterweights, lbs
d = distance from crankshaft to the center of gravity of the counterweight
d_e = diameter of engine sheave, in.
d_u = diameter of unit sheave, in.
D = depth of working fluid level in the casing annulus (net lift), ft
D_p = plunger diameter, in
e = loss in mechanical efficiency of structural bearings
e_p = plunger overtravel, in
e_r = rod stretch, in

EXAMPLE #16 (after Ref. 23)
PUMPING UNIT PERFORMANCE COMPARISON
UNIT A VS. UNIT B

	Unit A	Unit B
1. Type of unit geometry	640D-304-144	640D-304-144
2. Type of prime mover (h.p.)	75 NEMA "D"	75 NEMA "D"
3. Type of rod string	API 86	API 86
4. Pump diameter (inches)	1.5	1.5
5. Depth (ft.)	7,000	7,000
6. Gas interference	See Bottomhole Card	See Bottomhole Card
7. Volumetric efficiency (approx)	80%	80%
8. Anchored or unanchored tubing	Anchored	Anchored
9. Maximum and minimum prime mover speed (rpm)	1291.6 1064.4	1195.6 1134.7
10. Speed variation (%)	17.6	5.1
11. Cyclic load factor	2.4038	1.1249
12. Peak power (h.p.)	102.9	54.4
13. Regenerative power (h.p.)	−87.7	0
14. Nameplate power required	63.7	34.0
15. Maximum and minimum structural load (lbs.)	22,559 8,303	20,886 8,043
16. Maximum load as percentage of rating (%)	74.2	68.7
17. Average pumping speed (spm)	11.4	11.4
18. Polished rod horsepower	23.7	24.0
19. Peak net torque with existing counterbalance (in. lbs.)	601,800 1,260,800	289,300 1,300,000
20. Peak net torque and counterbalance required for perfect balance	580,800 1,287,200	250,200 1,393,900
21. Percent of gearbox rating (existing; in-balance)	94.0 90.8	45.2 39.1
22. Rotary inertia torque (slug ft.-ft.)	40,000 15,000	45,000 20,000
23. Articulating inertia torque (slug ft.-ft.)	19,000	38,000
24. Fluid load on pump; buoyant rod weight (lbs.)	3,500 12,690	3,500 12,690
25. Maximum and minimum surface rod stresses (psi)	28,723 10,572	26,592
26. Surface rod loading (lbs.)	22,559	20,886
27. Gross pump stroke	147.1	147.4
28. Net pump stroke	117.1	117.4
29. Rod string damping factor (up and downstroke)	.05 .15	.05 .15
30. Pump capacity based on gross and net pump stroke (bpd)	438.0 348.8	441.9 352.0
31. Tubing stretch (inches)	1	1
32. Lost pump displacement caused by tubing stretch (bpd)	3	3
33. Computed surface stroke (inches)	145.3	144.1
34. Rotation	C.W.	C.C.W.
35. Drive train efficiency (assumed)	90%	90%
36. Overall speed ratio	102	102
37. Index of torsional effectiveness (average torque/peak torque)	25.07%	58.92%

EXAMPLE #17 (after Ref. 23)
PUMPING UNIT PERFORMANCE COMPARISON
UNIT A VS. UNIT B

	UNIT A	UNIT B
1. Type of unit geometry	640D-304-144	640D-304-144
2. Type of prime mover (h.p.)	75 NEMA "D"	75 NEMA "D"
3. Type of rod string	API 86	API 86
4. Pump diameter (inches)	1.75	1.75
5. Depth (ft.)	9,403	9,403
6. Gas interference	See Bottomhole Card	See Bottomhole Card
7. Volumetric efficiency (approx)	100%	100%
8. Anchored or unanchored tubing	Anchored	Anchored
9. Maximum and minimum prime mover speed (rpm)	1277.9 1035.0	1209.2 1097.5
10. Speed variation (%)	19	9.2
11. Cyclic load factor	2.06	1.156
12. Peak power (h.p.)	116.3	82.9
13. Regenerative power (h.p.)	−73.7	−8.4
14. Nameplate power required	88.4	50.7
15. Maximum and minimum structural load (lbs.)	30,140 11,347	28,527 10,495
16. Maximum load as percentage of rating (%)	99.1	61.5
17. Average pumping speed (spm)	12.6	12.6
18. Polished rod horsepower	32.6	33.8
19. Peak net torque with existing counterbalance (in. lbs.)	599,000 1,300,000	424,500 1,497,000
20. Peak net torque and counterbalance required for perfect balance (in. lbs.)	581,000 1,281,400	393,400 1,465,600
21. Percent of gearbox rating (existing; in-balance)	93.6 90.8	66.3 61.5
22. Rotary inertia torque (slug ft.-ft.)	30,000 10,000	35,000 15,000
23. Articulating inertia torque (slug ft.-ft.)	25,000	45,000
24. Fluid load on pump; buoyant rod weight (lbs.)	4,200 18,100	4,200 18,100
25. Maximum and minimum surface rod stresses (psi)	38,376 14,448	36,322 13,363
26. Surface rod loading (lbs.)	30,140	28,527
27. Gross pump stroke	140.2	146.8
28. Net pump stroke	139.8	146.3
29. Rod string damping factor (up and downstroke)	.12 .12	.12 .12
30. Pump capacity based on gross and net pump stroke (bpd)	633.0 631.1	662.7 660.6
31. Tubing stretch (inches)	0	0
32. Lost pump displacement caused by tubing stretch (bpd)	0	0
33. Computed surface stroke (inches)	120.7	120.0
34. Rotation	C.W.	C.C.W.
35. Drive train efficiency (assumed)	90%	90%
36. Overall speed ratio	91	91
37. Index of torsional effectiveness (average torque/peak torque)	28.07%	42.8%

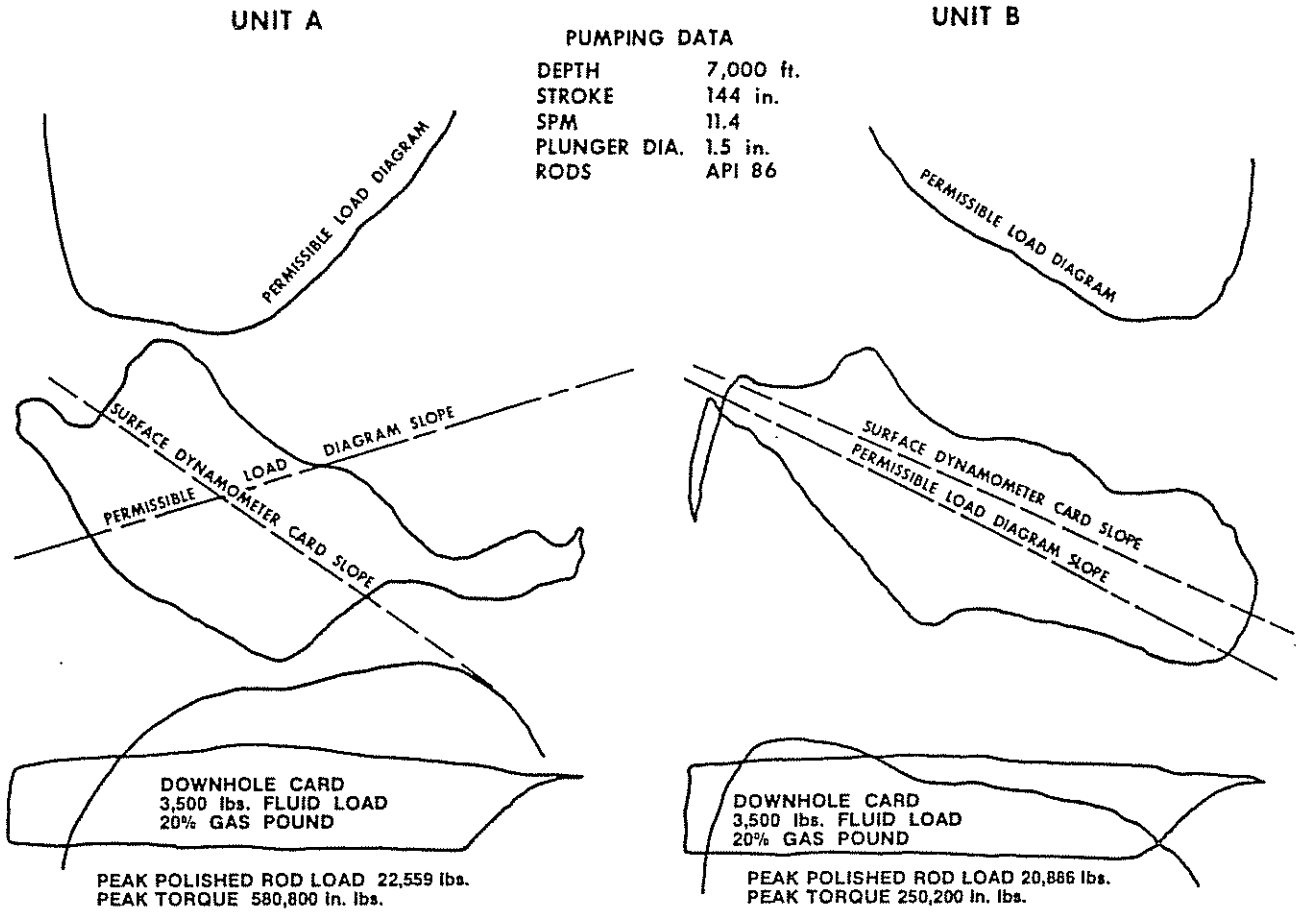


Fig. 2.73 (after Ref. 23) Dynagraphs and Permissible Load Diagrams for Example 2.16

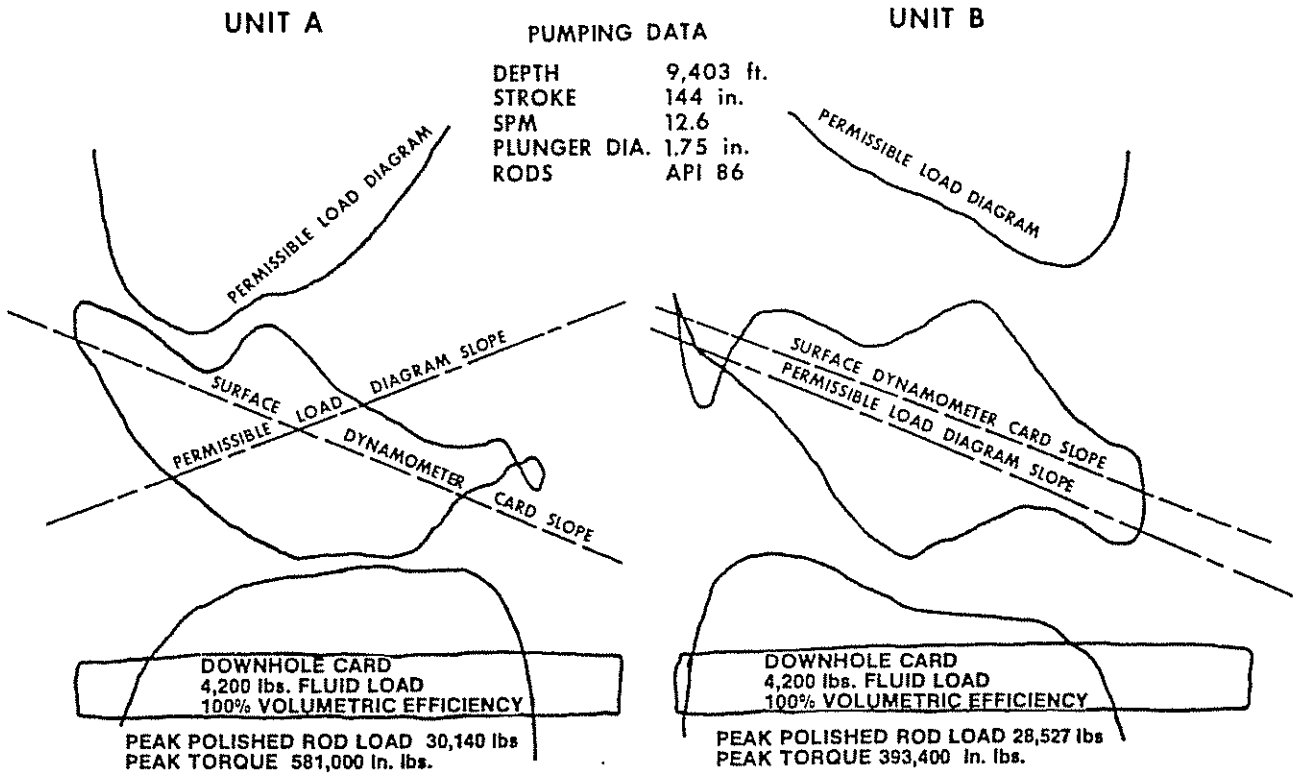


Fig. 2.74 (after Ref. 23) Dynagraphs and Permissible Load Diagrams for Example 2.17

e_t = tubing stretch, in
E = modulus of elasticity for steel, psi
EI = Economic Index
EBHP = engine brake horsepower
E_r = elastic constant of rod string, inches per pound-foot
E_s = surface efficiency of pumping system
E_t = elastic constant of tubing, inches per pound-foot
E_v = pump volumetric efficiency
G = fluid specific gravity
HP_f = frictional (subsurface) horsepower
HP_h = hydraulic horsepower
HP_{np} = nameplate motor horsepower
ITE = Index of Torsional Effectiveness
I = motor current
K = pump constant, Bbl/day/inch/spm
l_1 = distance from saddle bearing to the tail bearing
l_2 = distance from saddle bearing to the bridle
L = length of sucker rod string, ft
LE = lift efficiency
L_n = net lift, ft
M = max. moment of cranks & counterweights about the crankshaft
M_{CB} = counterbalance moment
MPRL = minimum polished rod load, lbs
N = pumping speed, strokes per minute (spm)
N_e = speed of engine sheave, rpm
N_u = speed of unit sheave, rpm
PD = pump displacement, B/D
PPRL = peak polished rod load, lbs
PRHP = polished rod horsepower
P_t = tubing pressure, psi
PT = peak torque, in-lbs
PT_b = in-balance peak torque, in.-lb
PT_D = peak torque on downstroke, in.-lbs
PT_u = peak torque on upstroke, in.-lbs
Q = actual surface production rate, B/D
R = percentage of a rod size in tapered rod string
r = distance from crankshaft to the pitman bearing
S = polished rod stroke length, in
S_A = max. allowable stress of rods, psi
S_p = effective plunger stroke, in
S.F. = rod service factor
TF = torque factor
T_r = gearbox torque rating, in.-lb
TF_{max} = maximum torque factor, wherever it occurs in the pumping cycle
TF_1 = maximum torque factor during the upstroke
TF_2 = maximum torque factor during the downstroke
T_{wn} = torque due to net well load
W_c = weight of counterweights, lb
W_f = weight of fluid, lb
W_n = net well load, pounds
W_p = permissible load, pounds
W_r = weight of rods in air, lb
W_l = weight of rods and fluid – W_r + W_f, lb
$W_{rf} = W_2$ = weight of buoyant rods, lb
Z = gear ratio of gear reducer
α = acceleration factor
Θ = crank position, degrees

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Chapter 3

GAS LIFT

by Joe Mach and Kermit E. Brown

3.1 INTRODUCTION

3.11 Definitions

Gas lift is a method of lifting fluid where relatively high-pressure (250 psi minimum) gas is used as the lifting medium through a mechanical process. This is accomplished by one of the two following methods:

(1) In *continuous flow* a continuous volume of high-pressure gas is introduced into the eductor tube to aerate or lighten the fluid column until reduction of the bottom hole pressure will allow a sufficient differential across the sand face, causing the well to produce the desired rate of flow. To accomplish this, a flow valve is used that will permit the deepest possible one point injection of available gas lift pressure in conjunction with a valve that will act as a changing or variable orifice to regulate gas injected at the surface depending upon tubing pressure. This method is used in wells with a high productivity index and a reasonably high bottom hole pressure relative to well depth.

In this type of well, fluid production can range from 200-20,000 B/D through normal size tubing strings. On casing flow it is possible to lift in excess of 80,000 B/D. The internal diameter (ID) of the pipe governs the amount of flow—provided the well productivity index, bottom hole pressure, gas volume and pressure, and the mechanical conditions are ideal. Smaller volumes can be efficiently lifted using continuous flow where small "macaroni" tubing is used. As low as 25 B/D may be produced efficiently through 1-in. tubing by continuous flow.

(2) *Intermittent flow* involves expansion of a high-pressure gas ascending to a low-pressure outlet. A valve with a large port permits complete volume and pressure expansion control of gas entering into the tubing, thus either regulating lift of the accumulated fluid head above the valve with a maximum velocity to minimize slippage or controlling liquid fall back, fully ejecting it to the tank with minimum gas. Generally used in conjunction with a surface time cycle controller (intermitter), intermittent lift is used on wells with fluid volumes that are relatively low, or wells that have the following characteristics: (1) high PI with low bottom hole pressure or (2) low PI with low bottom hole pressure.

In intermittent lift, gas is injected at regular intervals by the intermitter (a motor valve operated by a connecting timing device that permits selective cycling with controlled gas injection into the casing annulus). The cycling is regulated to coincide with fluid fill-in rate from the producing formation into the well bore.

Sometimes intermittent lift may be accomplished without using a surface intermitter. This type of installation requires a valve that is somewhat more fluid sensitive than normal and uses a built-in "spread." Use of this type of valve can present problems when wells must be lifted into long flow-lines against surface chokes or under other conditions where the sensitivity of the valve will become a disadvantage. However, there are other instances in which a valve of this type is an advantage, e.g. in some rotative compressor systems.

Intermittent lift may also be accomplished by the multi-point injection of gas through more than one gas lift valve. The installation must be designed so the lowest gas lift valves open just as the bottom of the slug passes each valve. Normally, this type of lift is used with fluid-operated valves which, by design, require a smaller port size to assure closure of the valve.

The percentage of wells going on gas lift has increased considerably in recent years—particularly since the end of World War II. This popularity is due to improved equipment and other technical advances. Considerable research was done from 1952–1977 to determine more accurately the pressure loss occurring in two-phase vertical and horizontal flow. Research work has been done for all types of flow occurring in vertical tubing strings. This has taken gas lift from "seat-of-the-pants" engineering to a precise engineering science.

Gas waste has greatly increased the use of gas compressors, and rotative gas lift systems are being utilized more and more. Air is also being used by first removing the oxygen, leaving mostly nitrogen and CO_2

as products of combustion. This allows wells to be gas lifted that previously were not considered. In rare cases unprocessed air is being used.

Gas lift is a means of lifting liquids (generally oil and/or water) from a vertical or inclined well by the injection of gas into the well. For example, in a well containing 5½-in. casing and producing through 2-in. tubing, the gas may be injected down the tubing-casing annular space and up the tubing string. The gas entering the string aerates the liquids and causes the well to flow (continuous flow) or enters beneath a slug of accumulated liquid and propels this slug to the surface in a piston form (intermittent flow).

The question arises: "What must gas lift accomplish for a well to produce?"

Assume that a 5000-ft well is standing full of salt water (γ_o = 1.07, gradient = 0.465 psi/ft) and that it must produce 100% salt water against 100 psi flowing tubing pressure at the surface. Assuming flowing frictional resistance is zero, this well must have flowing bottom hole pressure equivalent to tubing pressure plus the pressure exerted by the static column of salt water. This must be 100 + 5000 (0.465) = 100 + 2325 = 2425 psi.

Next, assume this well is making 100% 42° API oil but that it still produces against the 100 psi flowing tubing pressure. This 42° API oil has a γ_o of 0.815 or it exerts a gradient of 0.354 psi/ft. Again, neglecting flowing friction, the total flowing bottom hole pressure required would be 100 + (0.354) (5000) = 100 + 1770 = 1870 psi. Therefore, it is easier for the well to flow 100% oil than 100% water. A difference in flowing pressure of 555 psi exists for these two conditions. Further assume that the well is making solution gas along with the 100% oil and that this lightens the gradient to an average of 0.25 psi/ft. The flowing bottom hole pressure now required will be 100 + (5000)(0.25) = 100 + 1250 = 1350 psi. A further decrease in the required flowing bottom hole pressure is experienced.

The principle of continuous flow gas lift is now apparent. The flowing gradients of 0.465 psi/ft for 100% salt water, 0.354 psi/ft for 100% oil, and 0.25 psi/ft for oil with solution gas can all be decreased by the artificial injection of gas into the flowing fluid column. For example, a sufficient amount of gas might be injected to reduce the fluid pressure gradient to as low as 0.10 psi/ft. Gas is much lighter than either water or oil; by mixing it with either oil or water, a mixture results with a much smaller fluid gradient.

For example, if the average flowing gradient is 0.10 psi/ft with gas being injected at 5000 ft and the well is still flowing against 100 psi tubing pressure, the total flowing bottom hole pressure is now 100 + (5000)(0.10) = 100 + 500 = 600 psi. A comparison of these flowing bottom hole pressures shows 2425 psi for 100% salt water, 1870 psi for 100% oil, 1350 psi for 100% oil with solution gas, and 600 psi when gas is artificially injected at 5000 feet. Thus, gas lift creates the desired flowing bottomhole pressure (BHP) by injecting lift gas at the desired point in the tubing string.

The purposes of intermittent lift and continuous flow are identical—to create the necessary flowing bottom hole pressure to allow the well to produce a desired rate. One important difference, however, is that in intermittent lift a liquid slug must first build up, then gas is injected underneath the slug. The gas propels the slug to the surface similar to the firing of a gun. Flowing bottom hole pressure varies greatly during the period between the injection of gas underneath the slug and the escape of the slug at the surface.

The use of one large-ported valve to accomplish the lift cycle will result in maximum fluid recovery and minimum gas usage. This valve, the *operating valve*, is the deepest valve exposed to gas in the casing annulus. However, recent emphasis on multi-point injection (having several valves open as the slug passes) has also proven to be an efficient means of intermittent lift. Valves above the operating valve—*unloading valves*—facilitate unloading well fluids with the existing gas supply pressure.

In continuous flow installations and for intermittent installations in water drive reservoirs, the lowest valve is not always the operating valve. This and other valves are installed to be available when the well pressure declines and the point of operation moves down. An increase in the water content of the produced fluid can also necessitate working farther down the well.

An often overlooked feature of gas lift is its flexibility. Well characteristics are often unknown, and it is entirely possible that a well may be lifted by either continuous or intermittent flow. A well that has been designed properly may produce 1000 B/D of fluid or only 50 B/D. Most other types of artificial lift would require different types and sizes of equipment to offer this flexibility, whereas one gas lift installation could produce efficiently at either rate.

3.2 GAS LIFT VALVE CHARACTERISTICS

3.21 Introduction

Before a successful gas lift system can be designed, the designer must select a gas lift valve with proper characteristics. An installation cannot be designed properly or operated efficiently unless the gas lift valve is thoroughly understood. This chapter shows how various types of commonly-used gas lift valves operate under both static and dynamic conditions.

3.22 Gas lift valve nomenclature

Common oil field terminology has been adopted in which casing pressure (P_c) refers to gas pressure and tubing pressure (P_t) refers to flowing or static liquid pressure, regardless of the completion configuration. For the purpose of this chapter, this terminology will apply. This distinction is made since in annular flow wells the gas is injected down the tubing and liquid is produced up the casing tubing annulus. The valve mechanics principles are identical in both cases; only the nomenclature changes.

The gas lift industry has categorized gas lift valves depending on which pressure, (P_c) or (P_t) the valve is most sensitive to. Generally, they are classified by which pressure has the greater effect on the opening of the valve. This sensitivity is determined by mechanical construction of the gas lift valve. The pressure which is exposed to the largest area in the valve is the dominant pressure for that valve. Following is a list of the valves discussed in this chapter:

(1) *Casing pressure operated valve* (usually called a pressure valve). This valve is from 50% to 100% sensitive to casing pressure in the closed position and 100% casing pressure sensitive in the open position. It requires a buildup in casing pressure to open and a reduction in casing pressure to close.

(2) *Throttling pressure valve.* This valve is also called a proportional valve or continuous flow valve. It is the same as the pressure valve in the closed position. However, once in the open position this valve is tubing sensitive. It requires a build-up in casing pressure to open and a reduction in tubing pressure or casing pressure to close.

(3) *Fluid operated valve.* The fluid valve is 50% to 100% tubing pressure sensitive in the closed position and 100% tubing pressure sensitive in the open position. This valve requires a build-up in tubing pressure to open and a reduction in tubing pressure to close.

(4) *Combination valve.* This valve is also called a fluid open-pressure closed valve. It requires a build-up in fluid pressure to open and a reduction in casing or tubing pressure to close.

There are several types gas lift valves designed for a specific valve action. However, each valve will fall under one of these categories.

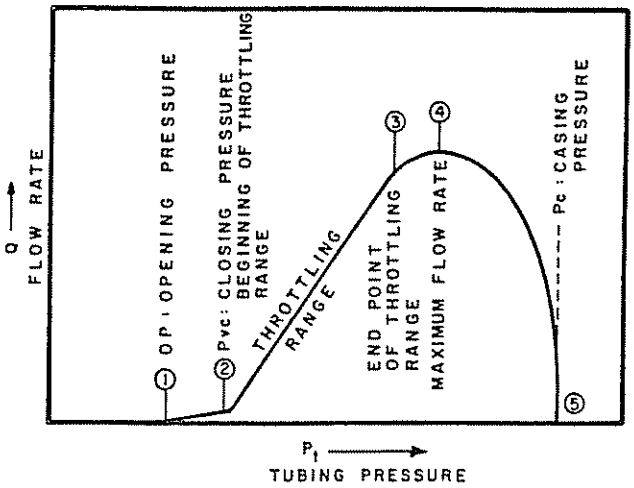


Fig. 3.21 Gas Passage Response Curve

3.23 Design considerations

Before selecting the type of valve to be used for a given installation, the designer must decide whether the well should be placed on intermittent or continuous

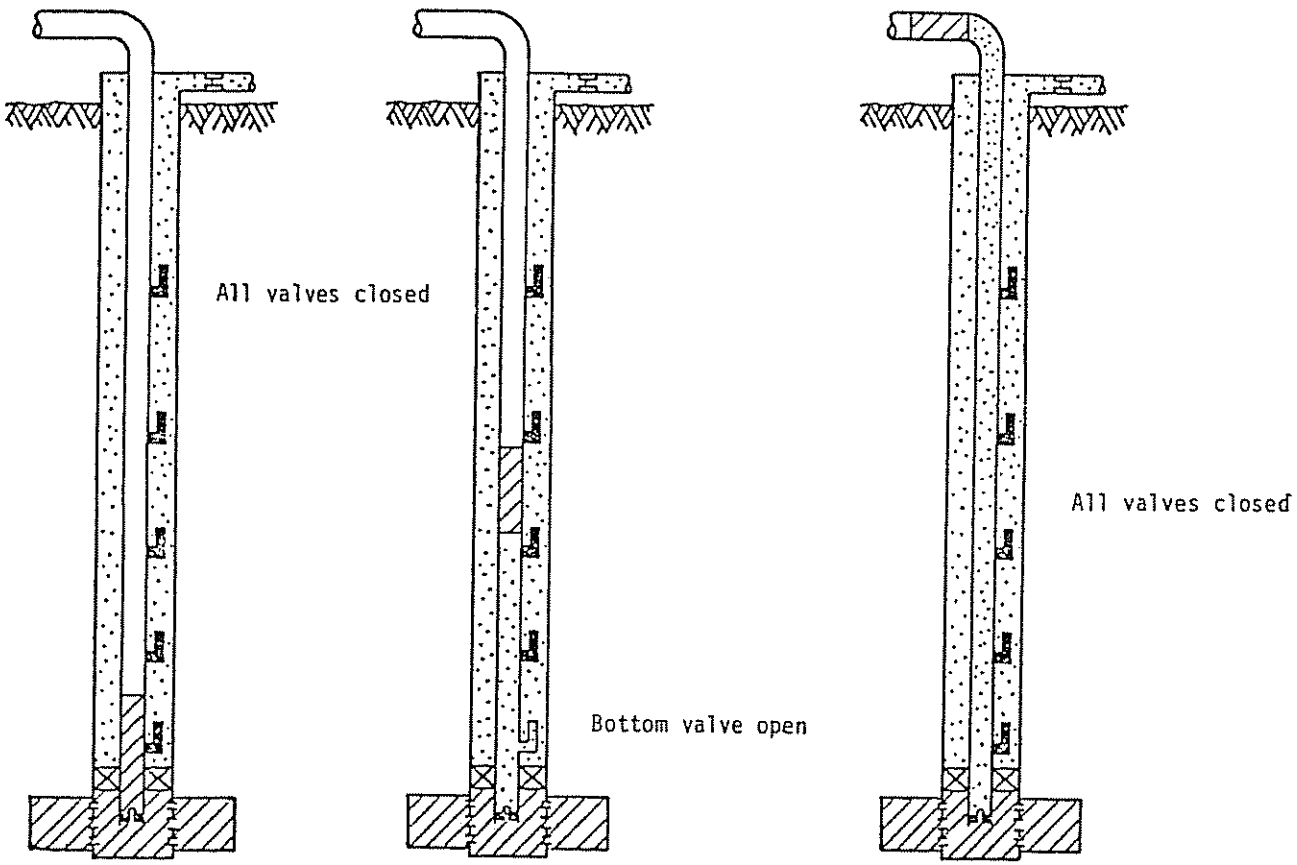


Fig. 3.22 Intermittent Single Point Injection

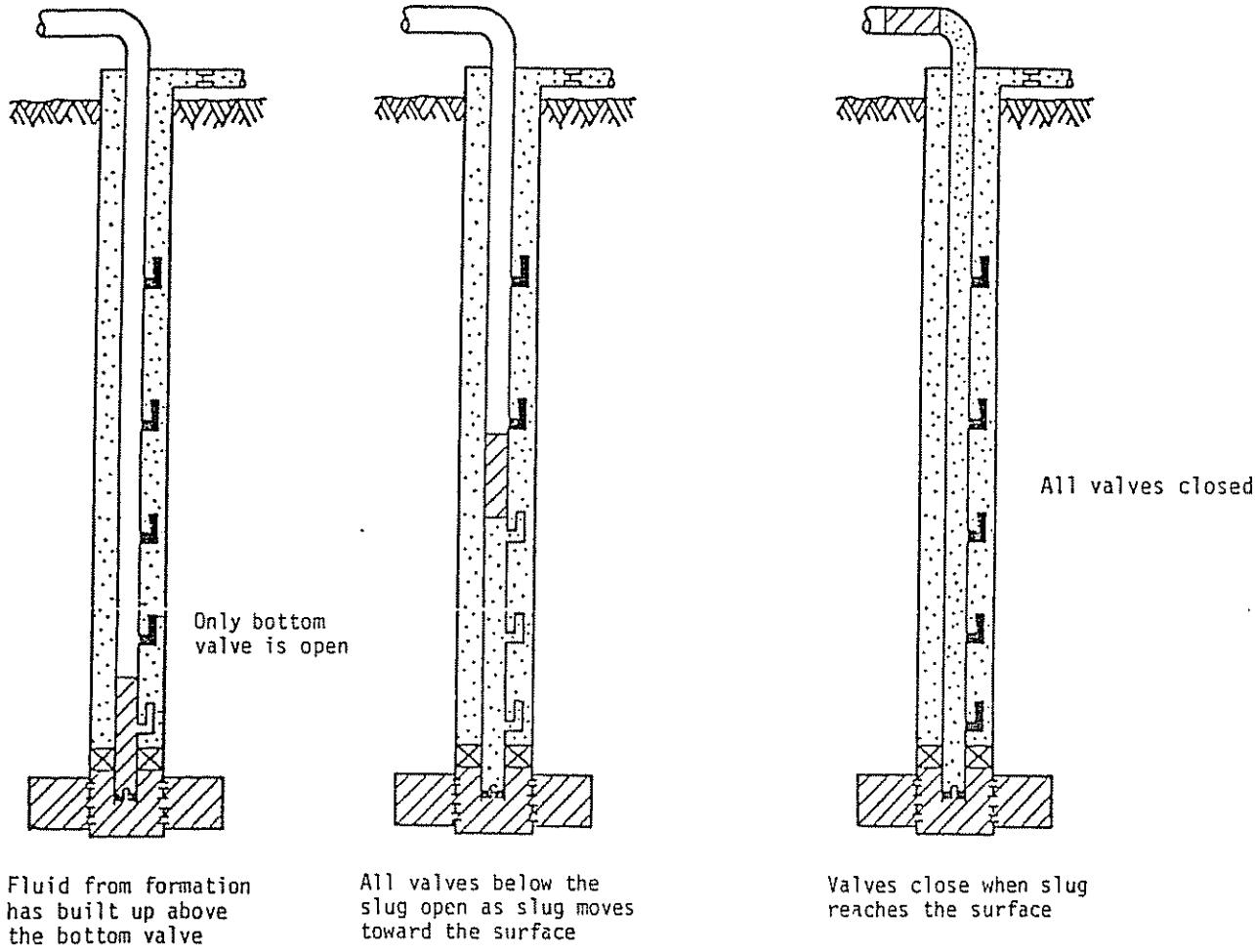


Fig. 3.23 Intermittent Multipoint Injection

flow gas lift. When information is lacking or when the well is borderline between continuous flow and intermittent lift, it is unknown at the time of the design whether the well will do better on continuous flow or intermittent lift. Valves are available that are suitable for both continuous flow and intermittent lift, depending on valve spacing and surface control of injection gas. In general, these valve characteristics are desired.

3.231 Continuous flow

A valve used for continuous flow application should be sensitive to tubing pressure when in the open position. The valve should respond proportionately to increase and decrease of tubing pressure. As the tubing pressure decreases, the valve should begin to throttle closed to decrease gas throughput; as the tubing pressure increases, the valve should begin to throttle open which will increase gas throughput. This proportional response maintains the established flowing tubing pressure or tends to keep a constant pressure inside the tubing.

Figure 3.21 shows the desired gas throughput response of a continuous flow gas lift valve. This characteristic can be achieved with a throttling pressure valve or a fluid valve. The opening mechanism is of little consequence for continuous flow application as long as the valve can be opened under operating casing and tubing pressure combinations.

3.232 Intermittent lift

Successful intermittent installations can be designed with almost any type of gas lift valve if designed properly. There are basically two types of intermittent lift, single point injection and multipoint injection. In single point intermittent lift all of the gas necessary to expand the slug to the surface is injected through the operating valve (see Fig. 3.22). This valve should

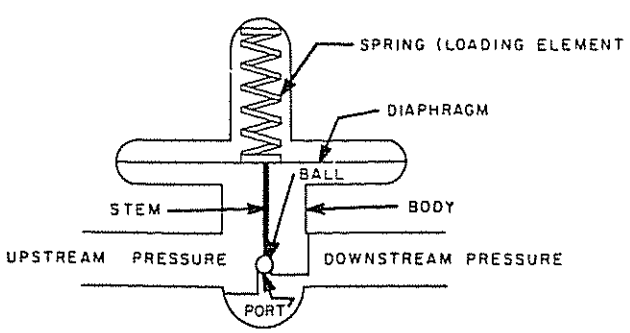


Fig. 3.24 Pressure Regulator

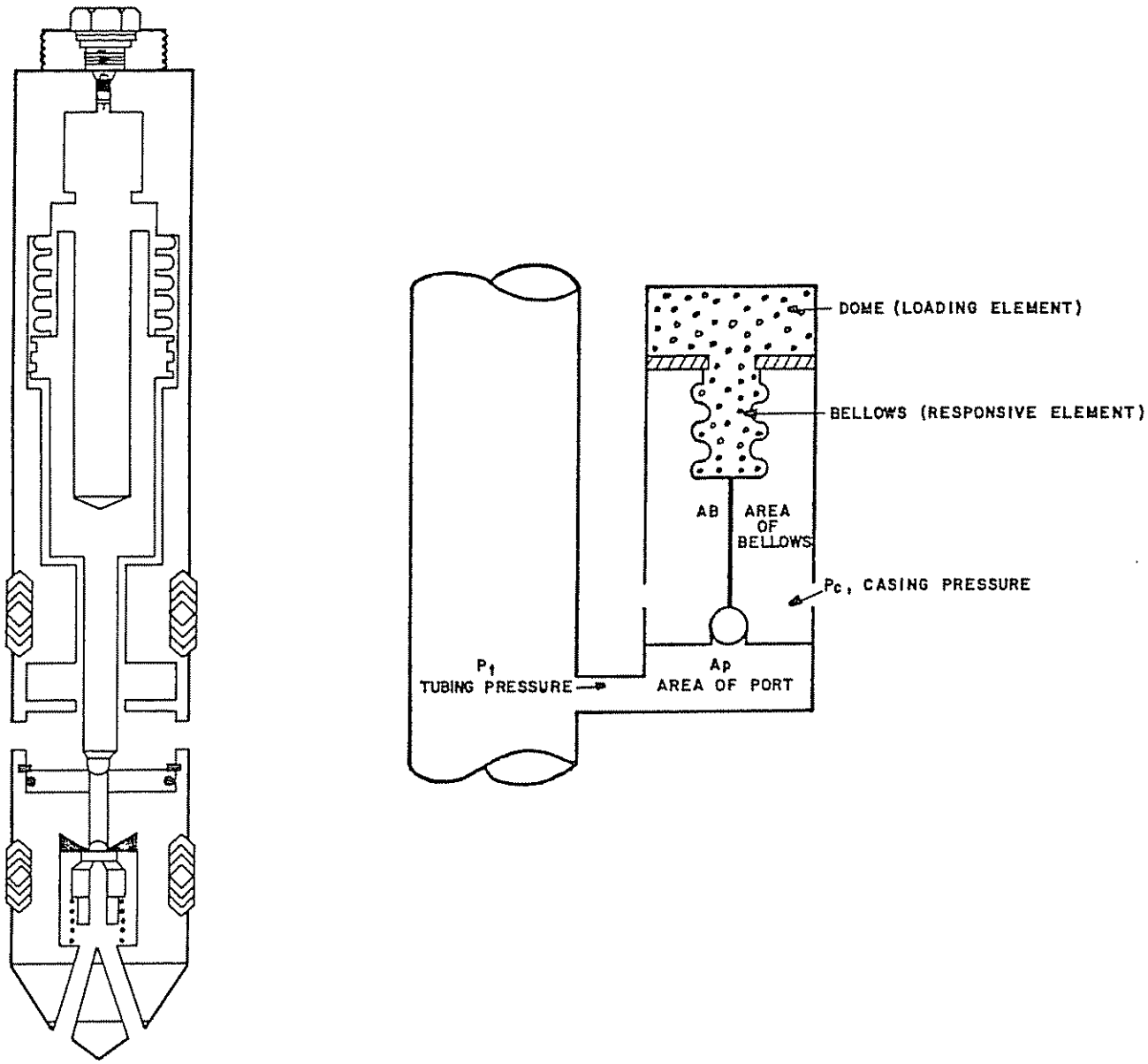


Fig. 3.25 Pressure Valve Schematic

expand to a port size as large as possible simultaneously with opening and should remain in the full open position until closing. The valve port sizes will range from ¼ inch to one inch in diameter, depending on the completion configuration which often limits the physical dimension of a valve which can be installed in a well.

Figure 3.23 shows the sequence of events for mutlipoint intermittent lift. The operating valve must pass enough gas to expand the slug to the next valve up the hole. At this time the pressure under the slug opens that valve which supplements the gas passage of the operating valve. The operating valve and the valve(s) opened as the slug moves up the hole should remain open until the slug reaches the surface.

All valves in the string need not be opened in multipoint lift. The number of opened valves depends on the type of valve used. gas lift design, and completion configuration. Any of these four types of valves can be used for this type of lift if designed properly. The design principles and type valves used are discussed in section 3.4.

3.24 Pressure, area and force relationships

Pressure is force per unit area. It is equal to the force acting perpendicular to an area divided by the area over which the force is distributed. The common oil field units are pounds force per square inch of area (psi).

$$\text{Pressure (psi)} = \frac{\text{F (force, lbs}_f\text{)}}{\text{A (area, sq in)}} \qquad (3.21)$$

If a pressure and area are known, the resultant force created by the pressure acting on an area can be calculated by multiplying the pressure by the area.

$$F = PA \qquad (3.22)$$

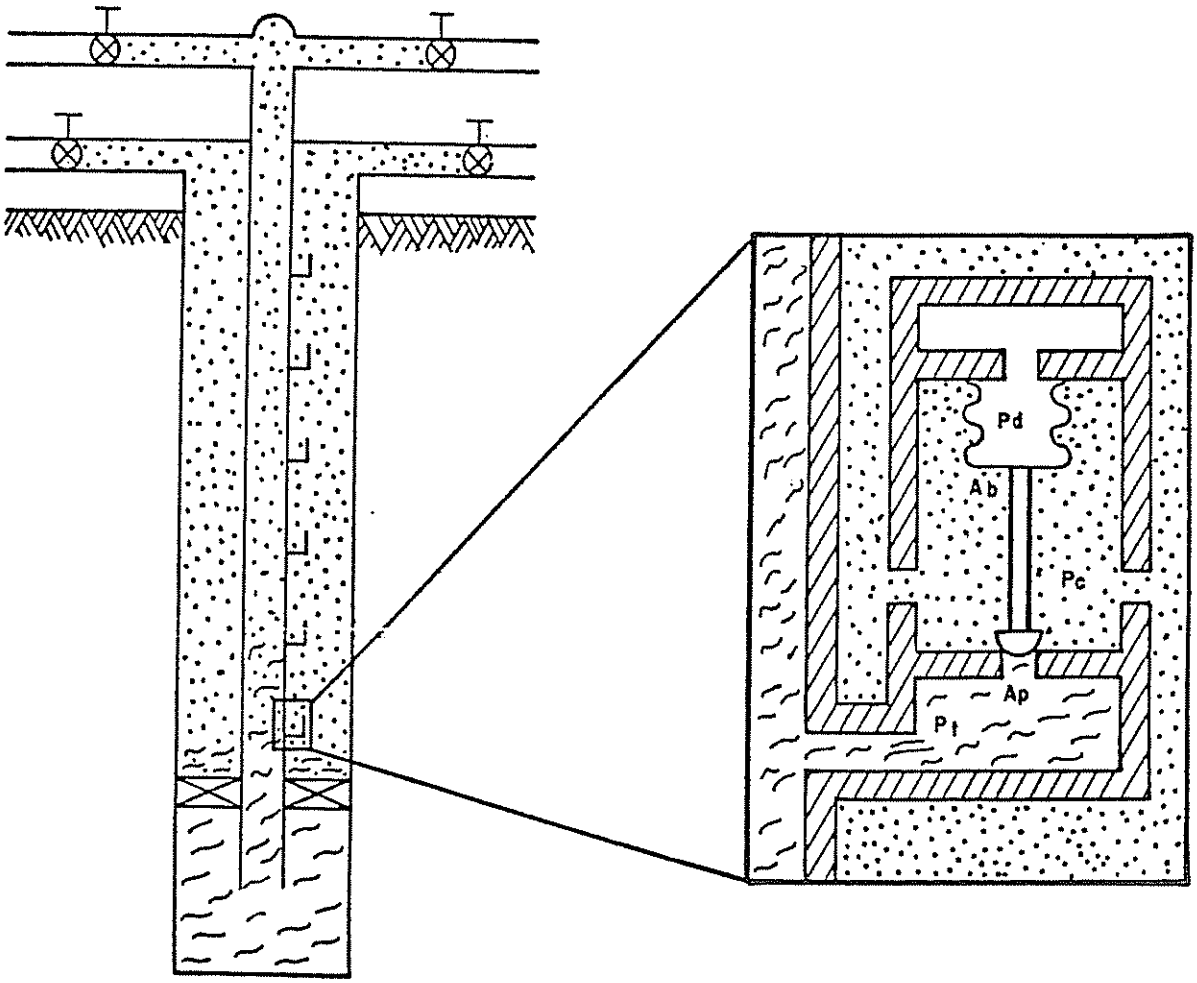


Fig. 3.26 Pressure Valve Under Actual Operating Conditions

Figure 3.24 shows a pressure regulator and nomenclature for each part of the regulator valve. This same nomenclature is used for gas lift valves. The spring supplies a downward force which keeps the ball on seat. The diaphragm acts as a responsive element of area for the upstream pressure to act on. This upstream pressure results in an upward force equal to $P_d A_b$ which tends to push the ball off the port seat. The port area serves as an area for gas passage and as an element of area for the downstream pressure to act on. The downstream pressure will also result in an upward force ($P_d A_b$) tending to push the ball off the port seat.

Since the diaphragm area is several times larger than the port area, each psi of upstream pressure change will alter the resultant force more than the same pressure change on the downstream side of the valve. For example, suppose the effective area of the diaphragm is 10 sq in. and the effective port area is 0.1 sq in. Using Equation 3.22, each psi of pressure change in upstream or downstream pressure will affect the upward force on this regulator valve in the closed position:

Upstream pressure:

$$F = PA$$

$$F = \frac{1 \text{ lb}}{\text{sq in.}} \times 10 \text{ sq in.} = 10 \text{ lb}_f$$

Downstream pressure:

$$F = P A$$

$$F = \frac{1 \text{ lb}_f}{\text{sq in.}} \times 0.1 \text{ sq in.} = 0.1 \text{ lb}_f$$

This valve is 100 times more sensitive to upstream pressure than downstream pressure when in the closed position. If this regulator were a gas lift valve and the upstream pressure were casing pressure, this would be a "pressure valve." If the upstream pressure were tubing pressure this would be a "fluid valve."

3.25 Casing pressure operated valve

3.251 Unbalanced bellows valve with pressure-charged dome as loading element

Figure 3.25 shows a pressure valve with a schematic showing the nomenclature. This is a single element valve since the only loading element is a nitrogen-charged dome. The responsive element is a bellows with convolutions which allow the stem to move the ball on and off seat. Since the tubing pressure acts on the area of the port, it supplies an opening force. Consequently, it is called an unbalanced pressure valve. In a bal-

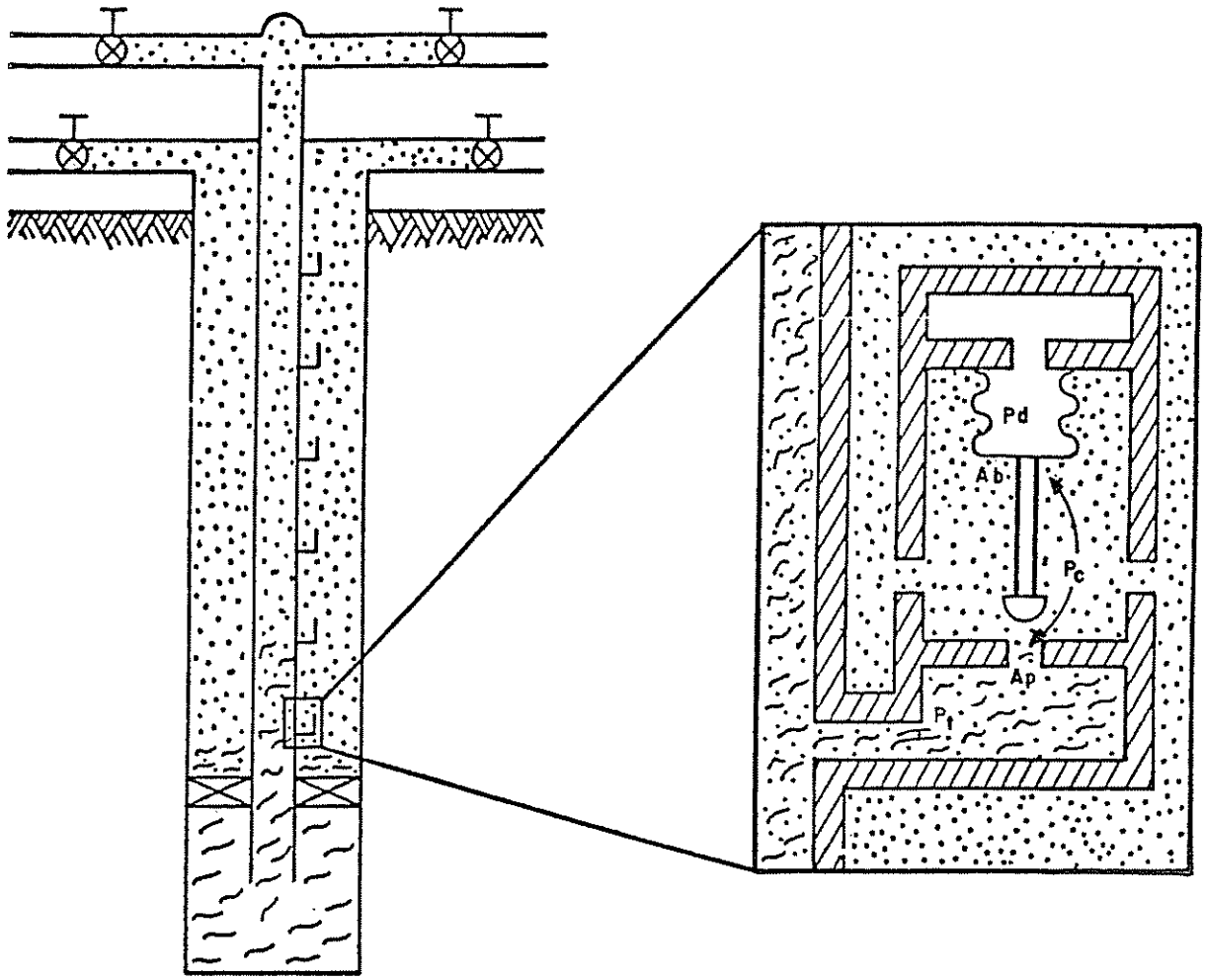


Fig. 3.27 Pressure Valve Under Actual Operating Conditions

anced pressure valve, the tubing pressure has no effect on opening or closing the valve.

3.2511 Opening pressure of valve under operating conditions

Since the pressure valve is mostly sensitive to casing pressure, the opening pressure is defined as the casing pressure required to open the valve under actual operating conditions. Since several pressures and areas are involved, a force balance equation must be derived to ascertain the opening pressure.

Figure 3.26 shows a valve under actual operating conditions. It illustrates forces that are acting on the valve by showing which pressures act on which areas.

To derive the force balance equation, it will be investigated in the closed position at the instant before opening. At this instant the forces which are trying to open the valve are equal to the forces trying to hold it closed. At this moment the following equation can be written:

$$F_o = F_c \tag{3.23}$$

where:

F_o = Summation of all forces trying to push the ball off seat.

F_c = Summation of all forces trying to hold the ball on seat.

(1) Forces trying to hold the valve closed:

$$F_c = P_d A_b \tag{3.24}$$

(2) Forces trying to open the valve:

$$F_o = P_c (A_b - A_p) + P_t A_p \tag{3.25}$$

Substituting into equation 3.23 gives:

$$P_c(A_b - A_p) + P_t A_p = P_d A_b \tag{3.26}$$

where:

P_d = Pressure in the dome, psig
A_b = Total effective bellows area, sq in.
P_c = Casing pressure at valve depth required to open the valve under operating conditions, psi
A_p = Seat area, sq in.
P_t = Tubing pressure at valve depth when the valve opens, psi

Solving for P_c:

$$P_c(A_b - A_p) = P_d A_b - P_t A_p$$

Divide each term by A_b:

$$P_c(1 - A_p/A_b) = P_d - P_t A_p/A_b$$

let $A_p/A_b = R$

then:

$$P_c(1 - R) = P_d - P_tR$$

Divide both sides by $1 - R$:

$$P_c = \frac{P_d - P_tR}{1 - R} \tag{3.27}$$

Equation 3.27 defines the casing pressure at depth required to open a pressure valve under operating conditions.

EXAMPLE PROBLEM #1

Suppose a pressure valve is located at 6000 ft. The pressure in the dome is 700 psi and the tubing pressure is 500 psi at 6000 ft. Find the casing pressure at 6000 ft required to open the valve if $A_b = 1.0$ sq in. and $A_p = 0.1$ sq in. Placing values in Equation 3.27:

$$P_c = \frac{P_d - P_tR}{1 - R}$$

when:

$P_d = 700$ psi
$P_t = 500$ psi
$A_b = 1$ sq in.
$A_p = 0.1$ sq in.
$R = 0.1/1 = 0.1$

then:

$$P_c = \frac{700 - 500\,(0.1)}{1 - 0.1} = \frac{650}{0.9} = 722 \text{ psi}$$

Under these operating conditions, when the casing pressure is built up to 722 psi at 6000 ft the valve will open.

Now investigate how much effect the tubing pressure has in opening the pressure valve. To do this rewrite Equation 3.27 as follows:

$$P_c = \frac{P_d}{1 - R} - \frac{P_t\,(R)}{(1 - R)} \tag{3.28}$$

Notice from Equation 3.28 that the tubing pressure term subtracts from the required casing pressure to open the valve. That is, as the tubing pressure increases, the casing pressure required to open the valve decreases. The tubing pressure term in Equation 3.28 is called the *tubing effect* (T.E.).

$$\text{T.E.} = P_t \frac{R}{(1 - R)} \tag{3.29}$$

The ratio $\frac{R}{1 - R}$ is called the *tubing effect factor* and is reported by most manufacturers as a percentage.

$$\text{T.E.F.} = \frac{R}{1 - R} \tag{3.210}$$

$$\text{T.E.} = P_t(\text{T.E.F.}) \tag{3.211}$$

EXAMPLE PROBLEM #2

Calculate the tubing effect caused by the tubing pressure of 500 psi in Example #1.

$$\text{T.E.F.} = \frac{0.1}{1 - 0.1} = 0.1111$$

$$\text{T.E.} + 500\,(0.1111) = 56 \text{ psi}$$

If the tubing pressure (P_t) were zero psi at valve depth it would require $722 + 56 = 778$ psi in the casing to open the valve. This 778 psi is sometimes called the maximum opening pressure. The tubing pressure (500 psi in this example) reduces the required opening pressure from the 778 psi maximum to 722 psi.

3.2512 Closing pressure of valve under operating conditions

The same force balance approach taken with the valve in the open position will be taken at the instant before it closes. Figure 3.27 shows the valve under actual operating conditions. All pressures and areas are identical, except now the pressure under the ball is assumed to be casing pressure, not tubing pressure. At the instant before closure the following equation can be written:

$$F_o = F_c \tag{3.212}$$

(1) Forces trying to close the valve:

$$F_c = P_d\, A_b$$

(2) Forces trying to hold the valve open:

$$F_o = P_c\,(A_b - A_p) + P_c\, A_p \tag{3.213}$$

Notice P_c has replaced P_t in Equation 3.25. Substituting into Equation 3.212 gives:

$$P_c\,(A_b - A_p) + P_c\, A_p = P_d\, A_b$$

Since we are looking for the pressure in the casing at valve depth where the valve closes, the following substitution will be made:

$$P_{vc} = P_c$$

where:

P_{vc} = Pressure in the casing at which the valve will close under operating conditions, (psi).

Therefore: Solving for P_{vc}:

$$P_{vc}\,(A_b - A_p) + P_{vc}\, A_p = P_d A_b \tag{3.214}$$

$$P_{vc}\, A_b - P_{vc}\, A_p + P_{vc}\, A_p = P_d A_b$$

or

$$P_{vc} = P_d \tag{3.215}$$

Equation 3.215 shows that the casing pressure at depth must be reduced to the dome pressure at depth for the valve to close. Based on Equation 3.215, the valve which opened at 722 psi in Example Problem #1 will close when the casing pressure at depth is reduced to 700 psi.

3.2513 Spread

The difference between the opening and closing pressure of a pressure valve is called *spread*. To calculate the spread, the closing pressure must be subtracted from the opening pressure, or Equation 3.27 minus Equation 3.215:

$$\text{Spread} = \Delta P = \frac{P_d - P_tR}{1 - R} - P_{vc} \tag{3.216}$$

Rearrange terms as follows:

$$\Delta P = \frac{P_d - P_tR}{1 - R} - \frac{(1 - R)\,P_d}{1 - R}$$

$$\Delta P = \frac{P_d - P_t R - P_d + P_d R}{1 - R}$$

$$\Delta P = \frac{R}{1 - R}(P_d - P_t)$$

Since $\frac{R}{1-R}$ = TEF:

$$\Delta P = TEF\ (P_d - P_t) \qquad (3.217)$$

EXAMPLE PROBLEM #3

Calculate the spread of the valve described in Example Problem #1.

$P_d = 700$ psi
$P_t = 500$ psi
$A_b = 1$ sq in.
$A_p = 0.1$ sq in
$R = 0.1/1 - 0.1$

$$TEF = \frac{R}{1-R} = \frac{0.1}{1-0.1} = 0.1111$$

$$\Delta P = 0.1111\ (700 - 500) = 22 \text{ psi}$$

Opening pressure $= P_{vo} = P + \Delta P = 700 + 22 = 722$ psi

Notice the results from this approach are the same as Example Problem #1.

Equation 3.217 shows that spread is controlled by two factors, TEF and P_t. For a given port size or R valve, the minimum spread (0) will occur where the tubing pressure = dome pressure. The maximum spread will occur where tubing pressure = zero, from Equation 3.217:

Maximum spread = ΔP max = TEF (P_d).

Spread can be controlled by designing for high tubing pressure loads. However, a high tubing pressure load may require unrealistically close valve spacing.

Figure 3.28 shows how the TEF will increase sharply as port size (R valve) increases. Therefore, if limited spread is desired the port size will necessarily be limited.

Spread can be important in continuous flow installations but is particularly important in intermittent gas lift installations where unbalanced pressure valves are used. The spread controls the minimum amount of gas used for each cycle. As pressure reduction (spread) required to close the operating valve increases, the amount of gas injected during that cycle increases.

Obviously, spread must be limited to prevent too much gas passage per cycle. However, horsepower and lift efficiency are sacrificed if the size of the valve's port through which the gas must flow during the lift cycle is restricted. For this reason pilot valves are made which can have a small R valve to control spread yet do not restrict the gas passage port.

3.2514 Gas pressure at depth

The opening and closing pressures of these pressure valves have thus far been at valve depth. In actual operation the surface pressure is the controlled pressure; thus, the surface (0 ft) is usually used as a datum to compare and relate gas lift valve settings. To correct from depth to surface or vice versa, the designer must predict the pressure change caused by the gas head and frictional losses above the gas lift valve under both static and dynamic (flowing) conditions.

The difference between static and dynamic pressure

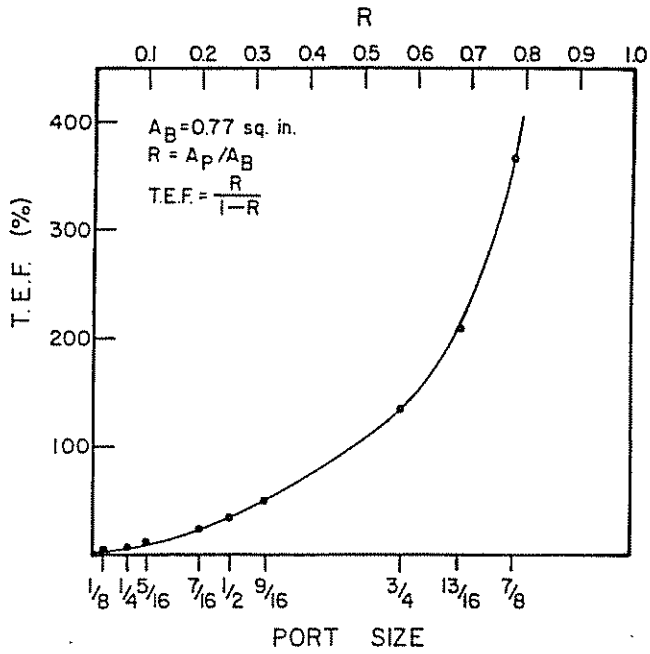


Fig. 3.28 Effect of Port Size on Tubing Effect Factor (TEF)

change is the frictional losses for the flowing gas. If the flowing conduit is small or the gas flow rate is relatively high, these losses must be accounted for. Frictional loss from downward flowing gas in the casing annulus of most installations is very small and can be neglected without noticeable error. For this reason most gas lift installations are designed using static gas pressure increases with depth.

The calculation of pressure increase due to gas column weight is based on a statement of energy balance on the gas flowing between any two points of the system; that is:

$$\int_1^2 vdP + \Delta \frac{v^2}{2\,g_c} + \frac{g}{g_c}\Delta X + lw + w = 0 \qquad (3.218)$$

For a static gas column with no velocity, friction, or work, Equation 3.218 reduces to:

$$\int_1^2 vdP + \frac{g}{g_c}\Delta X = 0 \qquad (3.219)$$

Solving Equation 3.219 for a non-ideal gas and assuming average temperature and Z factors give the following equation:

$$P_{depth} = P_{surface} \exp\left(\frac{0.01877\gamma_g L}{Tz}\right) \qquad (3.220)$$

where:

P_{depth} = Pressure at depth of gas column, psi
$P_{surface}$ = Surface pressure at wellhead, psi
e = Napierian Logarithm base (2.71828)
γ_g = Specific gravity of gas (air = 1.000)
L = Depth, ft
T = Average temperature, °R
z = Compressibility factor based on $\overline{T}$ and $\overline{P}$

Equation 3.220 involves a trial and error solution, since $\bar{z}$ depends on P_{depth} and vice versa. For this reason charts have been developed which give good results and are usually based on Equation 3.220. One such chart is shown in Figure 3A.1, Appendix 3A. For normal pressures and temperatures, the pressure increase with

depth (pressure gradient) is approximately constant for a given surface pressure. Therefore, the pressure in the casing can be described graphically by a straight line from the surface pressure to the pressure at depth.

EXAMPLE PROBLEM #4

Graphically describe the casing pressure for a well given the following:

Depth of packer = 8000 ft
Average temperature = 120°F.
$P_{surface}$ = surface casing pressure = 800 psi (P_{so})
γ_g = gas gravity = 0.65

Solution:

From Appendix 3A.1, F = 0.1979 for a 0.65 gravity gas at 8000 ft. Since the basis of the chart is 80°F., F must be corrected for temperature.

$$F_{corr} = F\left(\frac{80+460}{120+460}\right) = 0.1979\left(\frac{540}{580}\right) = 0.1843$$

$$P = 800\,(1 \div F_{corr}) = 800\,(1.1843) = 947 \text{ psi}$$

$$\text{Pressure gradient} = \frac{947-800}{8000} = 0.018 \text{ psi/ft}$$

or:

$$= \frac{18 \text{ psi}}{1000 \text{ ft}}$$

(See Fig. 3.29)

EXAMPLE PROBLEM #5

Suppose the surface opening pressure (psi) of a pressure valve is 750 psi. Find the opening pressure at depth, the closing pressure at depth, and the surface closing pressure given the following:

Valve depth = 6000 ft
Average temperature = 110°F.
$\gamma_g = 0.60$
$P_t = 500$ psi
R = 0.1

Solution:

Find the pressure increase due to gas weight for 750 psi surface pressure:

From Appendix 3A.1
F = 0.1330

$$F_{corr} = 1330\left(\frac{80+460}{110+460}\right) = 0.126$$

$$P_{vo} = 750\,(1 + F_{corr}) = 750\,(1.126 = 845 \text{ psi @ 6000 ft}$$

$$\text{Gas gradient} = \frac{845-750}{6} = 16 \text{ psi/1000 ft}$$

Pressure increase due to gas weight = 95 psi at 6000 ft
Opening pressure at depth:

$$P_{vo} = P_{so} + \Delta P_c = 750 + 95 = 845 \text{ psi}$$

where:

P_{so} = Surface opening pressure
ΔP_c = Pressure increase due to gas column weight.

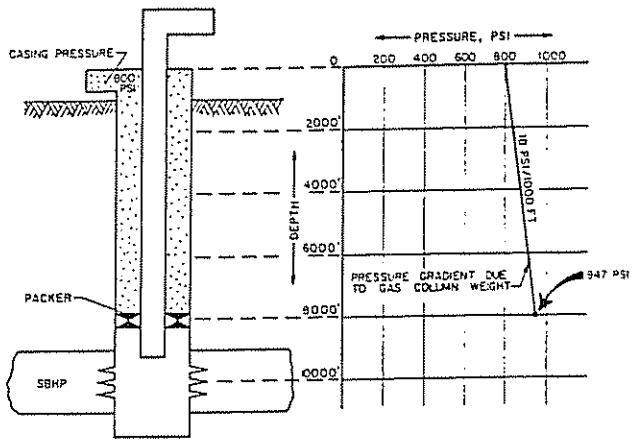


Fig. 3.29 Relationship Between Surface Pressure and Pressure at Depth

Closing pressure at depth:
Solving Equation 2.7 for P_d and remembering:

$P_{vc} = P_d$ = Valve closing pressure at depth
$P_{vc} = P_d = P_{vo}(1 - R) + P_t R$
$P_{vc} = 845\,(0.9) + 500\,(0.1) = 811$ psi

Surface closing pressure:

$$P_{sc} = P_{vc} - \Delta P_c = 811 - 95 = 716 \text{ psi}$$

where:

P_{sc} = Surface closing pressure

Some error is introduced by using the same gas pressure gradient for both surface pressures of 750 psi and 716 psi. In this example the spread is 34 psi at depth; however, if a smaller gradient is used at surface for the surface closing pressure calculation, the surface spread will be smaller. If the actual lower gradient is used, the surface close will be 720 psi or 4 psi higher.

For gas usage calculations in intermittent wells, the average of surface spread and down hole spread should be used if the difference is significantly large.

3.2515 Test rack opening pressure

Once the opening and closing pressures are established by the designer, the valve must be set at an opening pressure in the shop which corresponds to the desired opening pressure in the well. This is the test rack opening pressure (P_{tro}) (See Fig. 3.210). Notice that in a test rack the tubing pressure is 0 psig. If P_t is set equal to zero in Equation 3.27 the results are:

$$P_{vo} = \frac{P_d}{1-R} \qquad (3.221)$$

The dome of a gas lift valve is assumed to have a constant volume. Therefore, the pressure in a nitrogen-charged dome increases as temperature increases. The pressure in the dome (P_d) is known at valve depth. This means if Equation 3.221 were used for test rack opening pressure, each valve would have to be heated to a temperature equal to that at which it would operate in the well at depth.

Of course this is impractical. Most manufacturers cool their gas lift valves to 60°F. prior to setting them. This is done by submerging the valves in a water bath controlled at 60°F. To calculate the test rack opening

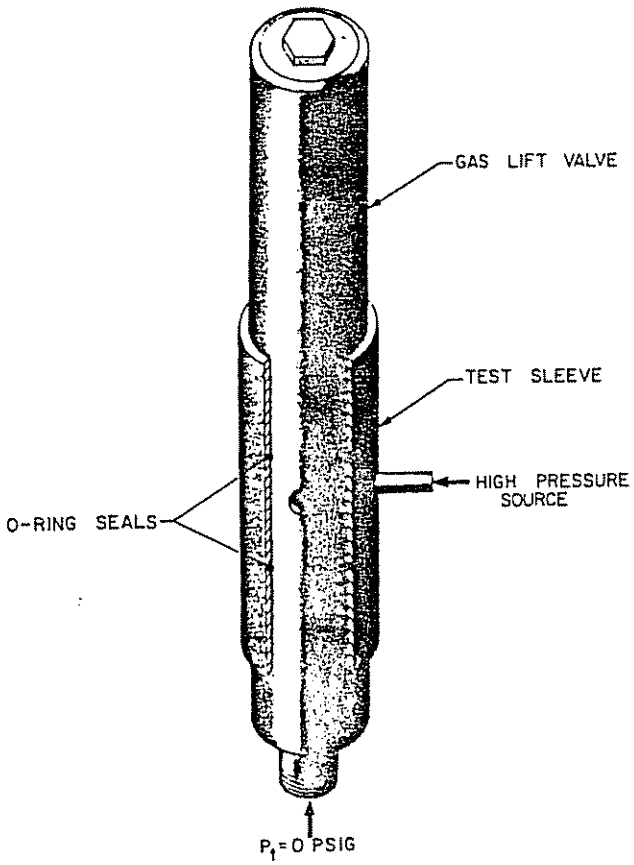


Fig. 3.210 Typical Test Facility for a Pressure Valve

pressure, the dome pressure at depth must be corrected to 60°F. Therefore, the equation used for test rack opening (P_{tro}) is:

$$P_{tro} = \frac{P_d @ 60°F}{1 - R} \tag{3.222}$$

To correct P_d to P_d @ 60°F. the non-ideal gas law is used.

$$\frac{P_d}{Z_d T_d} = \frac{P_d @ 60°F.}{Z_{60°F} 520}$$

or

$$P_{d\,@\,60°F} = \frac{P_d Z_{60°F} 520}{Z_d T_d} \tag{3.223}$$

Obviously, any base temperature could be used. Equation 3.221 would have to be changed for the new base temperature. Some manufacturers use a base temperature of 80°F. Since the solution of Equation 3.223 is trial and error, charts must be developed which are easy to use and are based on Equation 3.223. Appendix 3D.1 shows a temperature correction chart for a nitrogen-charged dome at a base temperature of 60°F.

EXAMPLE PROBLEM #6

Find the test rack opening pressure for the valve of Example Problem #5. Assume T = 70 + 1.3°F/100 ft.

Solution:

P_d @ 6000 ft = 811 psi

$$T_d @ 6000 \text{ ft} = 70 + \frac{1.3\,(6000)}{100} = 148°F.$$

From Appendix 3D (Table 3D.1), $C_t = 0.841$

$$P_d @ 60°F = C_t (P_d @ 6000 \text{ ft}) = 0.841\,(811) = 682 \text{ psi}$$

$$P_{tro} = \frac{P_{d\,@\,60°F}}{1 - R} = \frac{682}{0.9} = 758 \text{ psi}$$

3.2516 Standard pressure operated gas lift valve summary example given data:

Depth of valve = 8000 ft
Surface operating gas pressure = 800 psi
Surface opening pressure = 800 psi = P_{so}
Specific gravity of injection gas = 0.7
Temperature at surface = 100°F.
Temperature at 8000 ft = 180°F.
Pressure in tubing at valve location = 655 psi
Valve size = standard 1½" diameter
Port size = ½" diameter (R = 0.2562, 1 − R = 0.7438
Valve charge = Nitrogen at 60°F. in shop

Find:

(1) Opening pressure at valve depth, P_{vo}
(2) Closing pressure at valve depth, $P_{vc} = P_d$
(3) Spread at valve depth, ΔP_d
(4) Surface closing pressure, P_{sc}
(5) Spread at surface, ΔP_s and average spread
(6) Test rack opening @ 60°F, P_{tro}

Solution:

(1) The opening pressure at valve depth is equal to surface opening (P_{so}) plus the increase in casing pressure to 8000 ft due to gas column weight, from Appendix 3A.1.

$$F = 0.2145\ T_{avg} = 100 + 180 = 140°F$$

$$F_{corr} = 0.2145 \frac{540}{600} = 0.1931$$

$$P_{depth} = 800\,(1.1931) = 955$$

P_{vo} = 955 psi opening pressure at valve depth

2. The valve closing pressure at depth is equal to the dome pressure, P_d at 180°F.

$$P_{vc} = P_d = P_{vo}(1 - R) + P_t R$$

$$P_{vc} = 955\,(0.7438) + 655\,(.02562) = 879 \text{ psi}$$

3. Spread at depth is equal to P_{vo} minus P_{vc} or:

$$\Delta P_d = TEF\,(P_{vc} - P_t)$$
$$\Delta P_d = 955 - 879 = 76 \text{ psi}$$

or

$$TEF = \frac{R}{1 - R} = \frac{0.2562}{0.7438} = 0.344$$
$$\Delta P_d = 0.344\,(879 - 655) = 76 \text{ psi}$$

4. To calculate the surface closing pressure, P_{sc}, the surface pressure must be found which corresponds to the closing pressure at depth (P_{vc}) of 879 psi. The correction factor from surface to depth has already been determined in Step 1.

Therefore:

$$P_{vc} = P_{sc}\,(1 + F_{corr})$$
$$P_{sc} = \frac{P_{vc}}{1 + F_{corr}}$$

From Step 1: $1 + F_{corr} = 1.1931$

$P_{sc} = 879/1.1931 = 737$ psi

5. The spread at surface is the surface opening pressure (P_{so}) minus the surface closing pressure, P_{sc}:

$$\Delta P_s = P_{so} - P_{sc} = 800 - 737 = 63 \text{ psi}$$

$$\Delta P \text{ avg.} = \frac{\Delta P_d + \Delta P_s}{2} = \frac{76 + 63}{2} = 70 \text{ psi}$$

6. To calculate the test rack opening pressure using Equation 3.222, the dome pressure @ 60°F. must be determined. Using Table 3D.1 (Appendix 3D) for a temperature at the valve of 180°F:

$$C_t = 0.795 = \frac{P_d \text{ @ } 60°F.}{P_d \text{ @ } 180°F.}$$

or:

$$P_d \text{ @ } 60°F. = 0.795\ P_d \text{ @ } 180°F.$$

P_d @ 180°F. was found in Step 2:

$$P_d \text{ @ } 180°F. = P_{vc} = 879 \text{ psi}$$

$$P_d \text{ @ } 60°F. = 0.795\ (879) = 699 \text{ psi}$$

$$P_{tro} = \frac{P_d @ 60°F}{1 - R} = \frac{699}{0.7438} = 940 \text{ psi}$$

Summary tabulation

Depth	Port size	Temp. at valve	$R = A_p/A_b$	P_{so}	P_{vo}-opening pressure at depth	P_{vc}	P_{sc}	ΔP Avg. spread
8000	½	180°F.	0.2562	800	955	879	737	70

P_{tro} @ 60°F = 940 psi

3.252 Unbalanced bellows valve with pressure-charged dome and spring as loading element

This type of pressure valve is called a double element valve since it has two loading elements: the spring and a pressure-charged dome. Figure 3.211 shows the valve under actual operating conditions, illustrating the forces that are acting on the valve by showing which pressures are acting on which areas. Numerous spring loaded pressure valves are utilized for both intermittent and continuous flow gas lift. Often the dome charge is 0 psig, so all of the closing force is supplied by the spring.

3.2521 Opening pressure of valve under operating conditions

As with a single element valve, a force balance equation can be written for the valve in the closed position at the instant before opening. At this instant the forces which are trying to open the valve are exactly equal to the forces trying to hold it closed.

$$F_o = F_c$$

where:

F_o = summation of all forces trying to push the ball off seat

F_c = summation of all forces trying to hold the ball on seat

(1) forces trying to hold the valve closed:

$$F_c = P_d\ A_b + S_t\ (A_b - A_p) \tag{3.224}$$

(2) forces trying to open the valve:

$$F_o = P_{vo}\ (A_b - A_p) + P_t\ A_p \tag{3.225}$$

Substitution into Equation 3.223 gives:

$$P_{vo}\ (A_b - A_p) + P_t\ A_p = P_d\ A_b + S_t\ (A_b - A_p)$$

where:

P_d = pressure in the dome (psig)

A_b = total effective bellows area (sq in.)

P_{vo} = casing pressure at valve depth required to open the valve unde. operating conditions (psig)

A_p = seat area (sq in.)

P_t = tubing pressure at valve depth when the valve opens (psi)

S_t = an equivalent pressure caused by spring tension. Assumed to be acting on the area: $A_b - A_p$.

Solving for P_{vo}:

$$P_{vo}(A_b - A_p) + P_t\ A_p = P_d\ A_b + S_t\ (A_b - A_p)$$

Divide each term by A_b:

$$P_{vo}(1 - A_p/A_b) + P_t(A_p/A_b) = P_d + S_t(1 - A_p/A_b)$$

Let $A_p/A_b = R$, then:

$$P_{vo}(1 - R) + P_t\ (R) = P_d + S_t(1 - R)$$

Divide by $(1 - R)$ and solve for P_c:

$$P_{vo} = \frac{P_d}{1 - R} + S_t - \frac{P_t R}{1 - R} \tag{3.226}$$

Equation 3.226 is similar to Equation 2.7 except for the addition of the S_t term due to the spring. Notice if the dome is not charged, all of the closing force will be supplied by the spring. Equation 3.226 reduces to:

$$P_{vo} = S_t - \frac{P_t R}{1 - R} \tag{3.227}$$

If the dome pressure is needed to supplement the spring force, the designer will usually choose an S_t which will ensure the spring will not "stack" under operating conditions. Once the S_t is selected, P_d can be calculated.

3.2522 Closing pressure of valve under operating conditions

Figure 3.212 shows the valve under actual operating conditions. When the valve is open, the pressure under the ball is assumed to be casing pressure (not tubing pressure). At the instant before the valve closes, the following force balance equation can be written:

$$F_o = F_c \tag{3.228}$$

(1) forces trying to close the valve:

$$F_c = P_d\ A_b + S_t\ (A_b - A_p)$$

(2) forces trying to hold the valve open:

$$F_o = P_c\ (A_b - A_p) + P_c\ A_p \tag{3.229}$$

(P_c has replaced P_t in Equation 2.25)

Substitution into Equation 3.228 gives:

$$P_c(A_b - A_d) + P_c\ A_p = P_d\ A_b + S_t\ (A_b - A_p)$$

or

$$P_c A_b = P_d A_b + S_t (A_b - A_p)$$

replace P_c with P_{vc} and solve for P_{vc}:

$$P_{vc} = P_d + S_t (1 - R) \qquad (3.230)$$

3.2523 Spread

The spread of this valve is found by subtracting Equation 3.230 (closing pressure) from Equation 3.226 (opening pressure).

$$\text{Spread} = \Delta P = P_{vo} - P_{vc}$$

$$\Delta P = \left[\frac{P_d}{1-R} + S_t - \left(\frac{P_t R}{1-R} \right) \right] - [P_d + S_t (1 - R)]$$

$$\Delta P = \frac{R}{1-R} [P_d + S_t (1 - R) - P_t]$$

or

$$\Delta P = [\text{TEF}] [P_d + S_t (1 - R) - P_t] \qquad (3.231)$$

Notice that if there is no spring ($S_t = 0$), Equation 3.231 reduces to the same equation for spread in the single element pressure-charged valve.

3.2524 Summary examples

SUMMARY EXAMPLE #1: double element pressure valve without dome charge:

Given data:
- Depth of valve = 8000 ft
- Surface operating gas pressure = 800 psi
- Surface opening pressure = P_{so} = 800 psi
- Specific gravity of injection gas = 0.7
- Temperature at surface = 100°F.
- Temperature at 8000 ft = 180°F.
- Pressure in tubing at valve location = 655 psi
- Valve size = standard 1½" diameter
- Port size = ½" diameter (R = 0.2562, 1 − R = 0.7438)
- Valve charge = spring only

Find:
(1) Opening pressure at valve depth, P_{vo}
(2) Closing pressure at valve depth, P_{vc}
(3) Spread at valve depth, ΔP depth
(4) Surface closing pressure, P_{sc}
(5) Spread at surface, (ΔP surface) and average spread
(6) Test rack opening

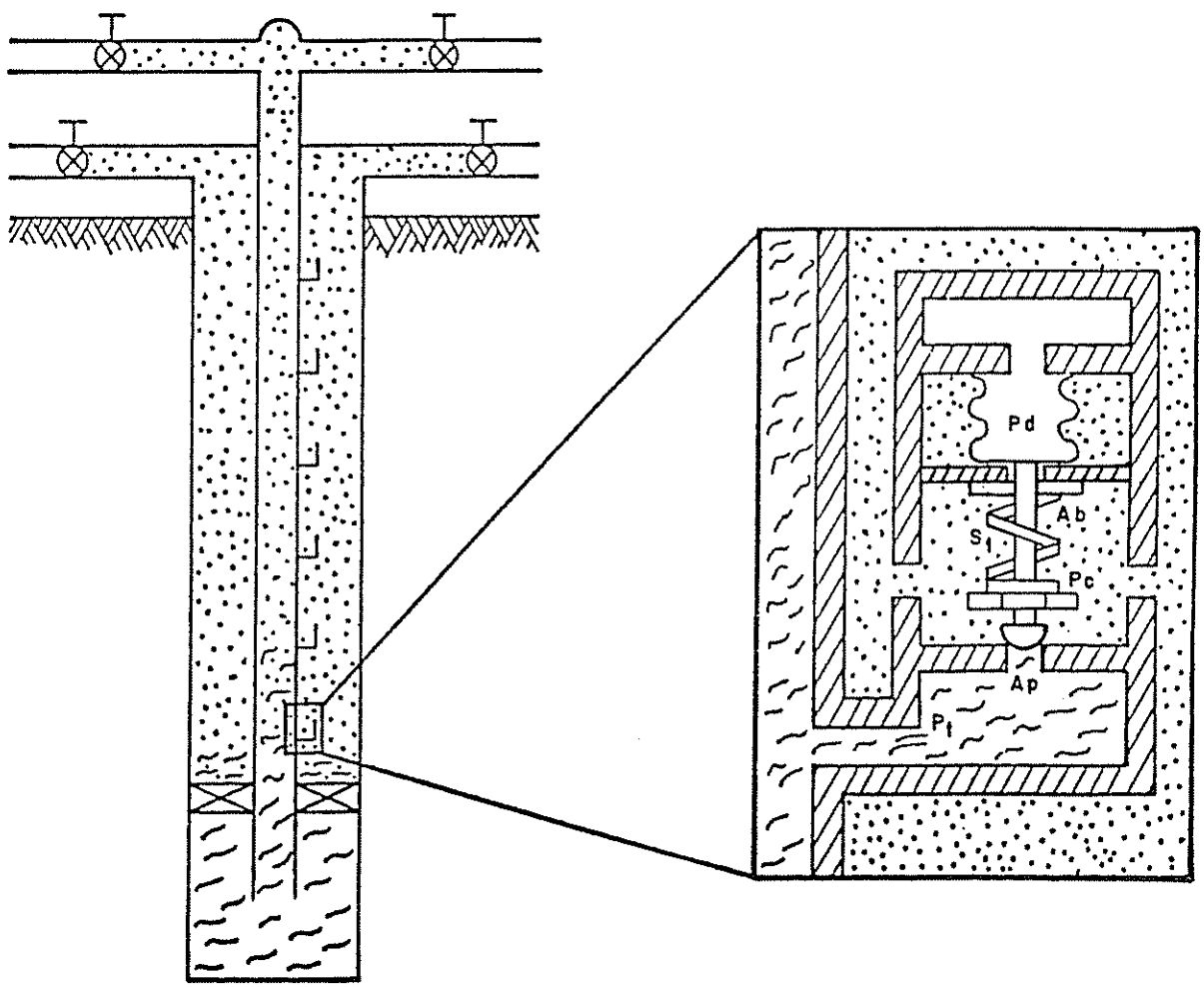


Fig. 3.211 Unbalanced Bellows Valve with Pressure Charged Dome and Spring as Loading Element

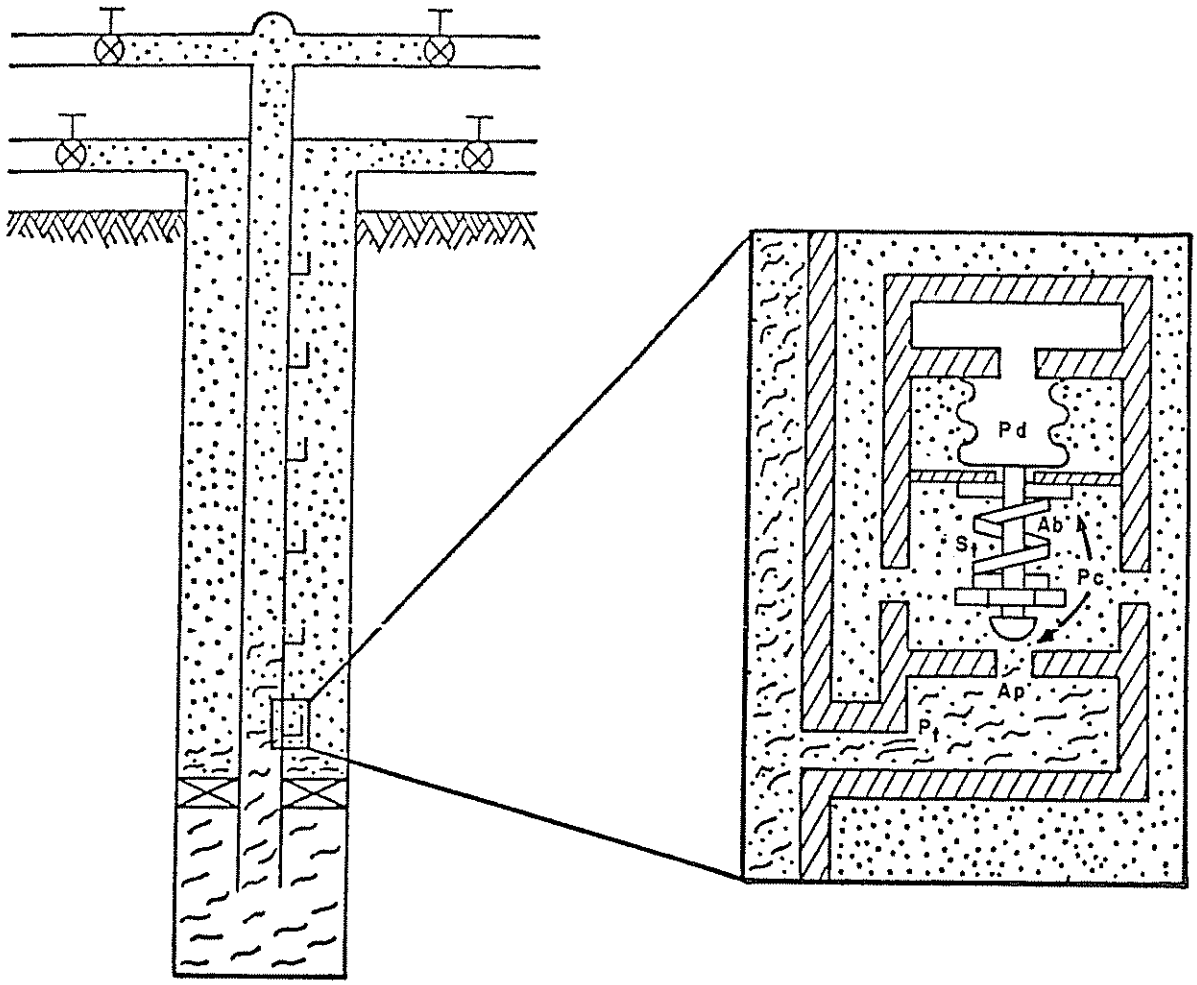


Fig. 3.212 Valve Under Operating Conditions

Solution:

(1) Opening pressure at depth:
From Appendix I

$$F = 0.2145\,T\,avg = \frac{100 + 180}{2} = 140°F.$$

$$F_{corr} = 0.2145\frac{540}{600} = 0.1931$$

$P_{c\ depth} = 800\,(1.1931) = 955$ psi
$P_{vo} = 955$ psi opening pressure at valve depth

(2) Setting the dome pressure equal to 0 psig in Equation 3.230, the closing pressure at depth is:

$$P_{vc} = 0 + S_t\,(1 - R) = S_t\,(1 - R)$$

S_t must be found by solving Equation 3.226 for S_t using the P_{vo} calculated in step 1 and setting $P_d = 0$ psig:

$$S_t = P_{vo} + \frac{P_t R}{1 - R} = 955 + 655\left(\frac{0.2562}{0.7438}\right)$$

$S_t = 1181$
$P_{vc} = 1181\,(0.7438) = 879$ psi

(3) Spread at depth is equal to P_{vo} minus P_{vc}, or
$\Delta P_{depth} = TEF\,[S_t\,(1 - R) - P_t]$
$\Delta P_{depth} = 955 - 879 = 76$ psi

or:

$$TEF = \frac{R}{1 - R} = \frac{0.2562}{0.7438} = 0.344$$

$\Delta_{depth} = 0.344\,(879 - 655)\gamma = 76$ psi

(4) Surface closing pressure:

$P_{vc} = P_{sc}\,(1 + F_{corr})$
$P_{sc} = P_{vc}/(1 + F_{corr})$

from Step 1: $1 + F_{corr} = 1.1931$

from Step 2: $P_{vc} = 879$

$$P_{sc} = \frac{879}{1.1931} = 737 \text{ psi}$$

(5) Surface spread:

$P_{surf} = P_{vc} - P_{sc} = 800 - 737 = 63$ psi

$$P_{avg.} = \frac{\Delta P_{surf} + \Delta P_{depth}}{2} = \frac{76 + 63}{2} = 70 \text{ psi}$$

(6) Since there is 0 psig in the dome and temperature has essentially no effect on spring tension, it is unnecessary to account for temperature when calculating the test rack opening pressure of this valve. In the test rack $P_t = 0$. Since $P_d = 0$, Equation 3.226 is the equation for P_{tro} when $P_t = 0$:

$P_{tro} = S_t$
$P_{tro} = 1181$ psi

SUMMARY EXAMPLE #2: double element pressure valve with pressure-charged dome and spring tension

Given data: Same as previous example except valve charge. Set $S_t = 600$ psi.

(1) Opening pressure at depth is found in the same manner as the previous example.

$P_{vo} = 955$ psi opening pressure at valve depth

(2) Closing pressure at depth. Since $P_{vc} = P_d + S_t(1 - R)$; Equation 3.226 can be expressed as follows:

$$P_{vo} = \frac{P_{vc} - P_t R}{1 - R}$$

solving this equation for P_{vc} gives:

$$P_{vc} = P_{vo}(1 - R) + P_t R$$

since P_{vo} is known from Step 1

$$P_{vc} = 955(.743) + 655(0.2562) = 879 \text{ psi}$$

(3) Spread at depth:

$$\Delta P_{depth} = P_{vo} - P_{vc} = 955 - 879 - 76 \text{ psi}$$

or if Equation 2.31 is used P_d must be determined.

From Equation 3.230:

$$P_d = P_{vc} - S_t(1 - R)$$
$$= 879 - (600)(0.7438) = 433 \text{ psi}$$

therefore:

$$\Delta P_{depth} = TEF[P_d = S_t(1 - R) - P_t]$$
$$= 0.344[433 + 600(0.7438) - 655] =$$
$$76 \text{ psi}$$

(4) Surface closing pressure:

$P_{sc} = 737$ psi

(5) Surface spread:

$\Delta P_{surf} = 63$ psi
$\Delta P_{avg} = 70$ psi

(6) Since the dome will be charged with nitrogen, the valve must be set at 60°F. or some other base temperature. Setting $P_t = 0$ in Equation 3.226 the test rack opening pressure is:

$$P_{tro} = \frac{P_d @ 60°F}{1 - R} + S_t$$

The dome pressure (P_d) was found in Step 3 to be 433 psi

Using Appendix 3D and Table 3D.1 for a temperature at the valve of 180°F:

$$C_t = 0.795 = \frac{P_d @ 60°F}{P_d @ 180°F}$$

$$P_d @ 60°F = 0.795(433) = 344 \text{ psi}$$

$$P_{tro} = \frac{344}{0.7438} + 600 = 1062 \text{ psi}$$

Setting procedure:

Equation 3.227 shows that in a test rack where $P_t = 0$, the test rack opening pressure is equal to S_t.

$P_{tro} = S_t$ when dome pressure equals 0 psi.

(A) Install valve in test rack. Adjust spring setting until the valve opens with 600 psig applied pressure. This sets

$S_t = 600$ psig in the valve

(B) Pressure up the dome. Cool valve to 60°F.

(C) Bleed pressure off of dome until valve opens with 1062 psig applied pressure.

3.253 Balanced pressure valve

A balanced pressure valve is not influenced by tubing pressure when in the closed or open position (see Fig. 3.213). Notice that casing pressure acts on the entire area of the bellows at all times. This means the valve opens and closes at the same pressure-dome pressure. The spread is zero regardless of the port size.

Figure 3.214 shows a completely different construction for a gas lift valve. The main element of this valve is a resilient flexible sleeve which seals the dome of the valve. The dome is charged with a gas at a predetermined pressure. The flexible sleeve seals off the injection gas flow courses through the valve when the valve is closed and flexes to open these flow courses when the pressure acting on the outside of the sleeve exceeds the dome pressure.

This is a balanced valve—it is not influenced by tubing pressure. If the dome is charged to 700 psig, then the valve opens and closes at 700 psig. A further example would be to have a 700 psig valve with a 400

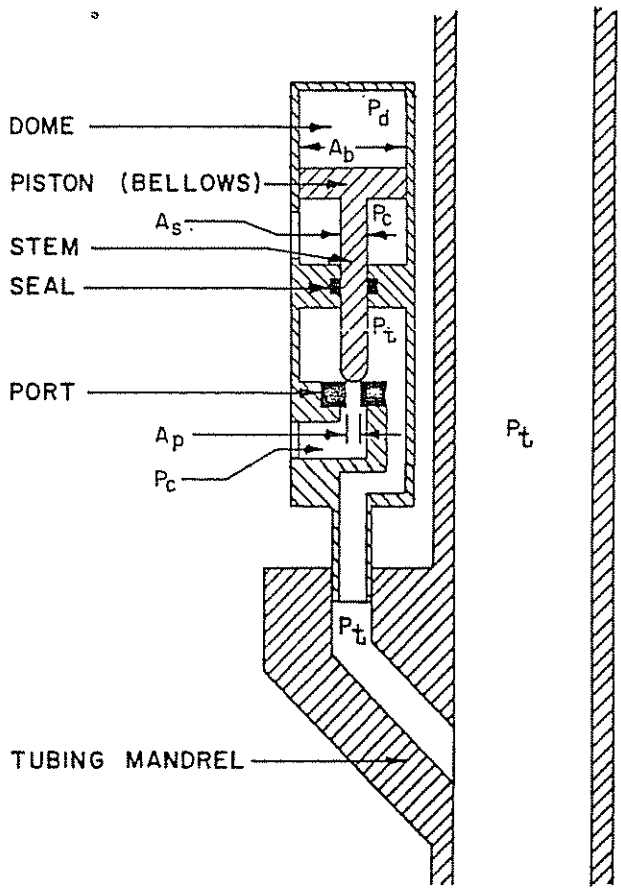


Fig. 3.213 Balanced Casing Pressure Operated Valve

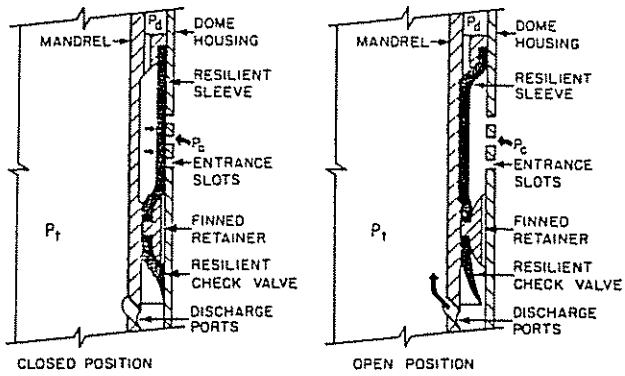


Fig. 3.214 Balanced Flexible Sleeve Valve

psig tubing load. The 400 psig tubing load has no effect on the opening or closing pressures of this valve. The valve still opens and closes at the same pressure.

Therefore, this valve has no spread. It acts as an expanding orifice regulator, opening to pass whatever gas is injected from the surface. If additional gas is needed, it is injected at the surface and the valve opens to take care of it. If less gas is needed, an adjustment is made at the surface and the valve partially closes to control the lower gas rate.

3.254 Pilot valves

Since for intermittent lift it is desirable to have a gas lift valve with a large port yet have close control over the spread characteristics, a pilot valve was developed. Two different types of pilot valves are shown in Figures 3.215 and 3.216. In both of the arrangements the effective A_p used for opening calculations is the small port (inner port in Fig. 3.215). The larger port is used for gas passage when the valve opens. The large port can be as large as mechanically possible and does not have to be changed to alter the spread. This allows the designer to achieve maximum intermittent lift efficiency. The valve mechanic's calculations are exactly the same as already discussed. Often the small port is called the "control port" and the larger gas passage port is the "power port."

For choke control intermittent lift, the control port should be selected so the actual spread is about 10% smaller than the desired spread. For time cycle controlled intermittent lift, the control port should be selected so the actual spread is from 30% to 60% smaller than the required spread. The proper amount of gas can be provided by adjusting the gas injection time on the controller. Remember that the spread sets the minimum amount of gas injected each cycle. If desired, more gas can be injected by overriding the valve with the time cycle controller.

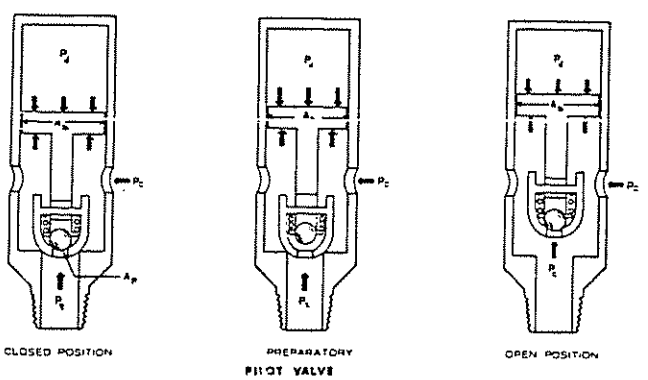


Fig. 3.215 Pilot Operated Valve

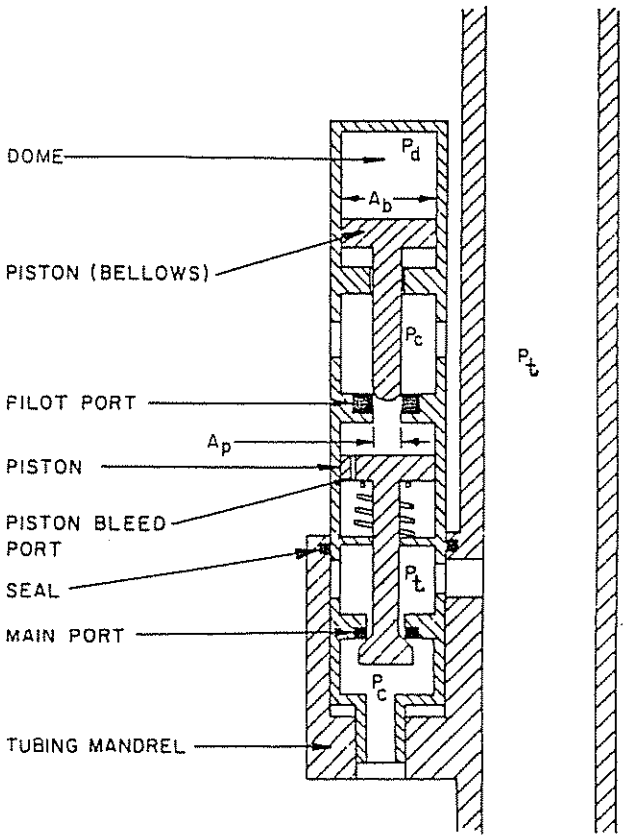


Fig. 3.216 Pilot Operated Valve

EXAMPLE PROBLEM #7

An intermittent installation is to be designed to lift from 8000 ft. Select the control port for both choke and time cycle controlled intermittent lift given the following:

Surface opening pressure = 800 psi
Gas gravity = 0.65
Tubing size = 2⅜ EUE 8rd
Casing size = 7" – 23#/ft
Average temp. = 120°F.
Separator pressure = 50 psi
Valve type: standard 1½" O.D. pilot valve
Power port: 9/16
A_b = 0.77 sq in.
Design tubing pressure = 600 psi

Solution:

The gas pressure at depth corresponding to 800 psi at the surface is 947 psi. Therefore, the opening pressure of the valve at depth will be 947 psi.

From Figure 3G.7 (Appendix 3G), the required input gas per cycle is 6400 scf.

From Figure 3H.6 (Appendix 3H), the required spread to achieve this 6400 scf is about 60 psi.

Choke control:

$$\Delta P = 60 \text{ psi} = TEF\,(P_d - P_t)$$
$$P_{vc} = P_d = 947 - 60 = 887 \text{ psi}$$

Therefore:

$$TEF = \frac{\Delta P}{P_d - P_t} = \frac{60}{887 - 600} = 0.21$$
$$TEF = \frac{R}{1 - R} = 0.21$$

or

$$R = \frac{0.21}{1.0 + .21} = 0.17$$
$$A_p = 0.17\,(0.77) = 0.13 \text{ sq in.}$$

A 24/64" port has an area of 0.1134 sq. in.; (select the ⅜" port)

$$\text{Actual } R = \frac{0.1134}{0.77} = 0.1473 \qquad TEF = 0.173$$

$$\text{Actual spread} = 0.173\,(986 - 600) = 51 \text{ psi}$$

This is close enough to desired spread; use a 24/64" port.

Time cycle controller:

Try a ¼" port.

$$A_p = 0.0511 \text{ sq in.}$$
$$R = \frac{0.0511}{0.77} = 0.066$$
$$1 - R = 0.934$$
$$TEF = 0.071$$
$$P_d = P_{vo}(1 - R) + P_t R = 947(.934) + 600(.066) = 924 \text{ psi}$$
$$\text{Spread} = 0.071\,(924 - 600) = 23 \text{ psi}$$

This is sufficient; use a ¼" port.

3.255 Gas passage

The pressure valves discussed in this chapter have the same gas flow characteristics as a square-edged orifice (see Fig. 3.217).

The critical flow pressure ratio for natural gas will generally occur where:

$$\frac{P_t}{P_c} = 0.55$$

Gas flow through a choke can be calculated from the following equation as given by Craft, Holden & Graves:[1]

$$Q = \frac{155.5\,C_d A P_1 \sqrt{2g\dfrac{K}{K-1}\left(R^{2/K} - R^{(K+1)/K}\right)}}{\sqrt{\gamma T}}$$

where:

Q = gas flow in Mcfd @ 14.65 psi and 60°F.
C_d = discharge coefficient
A = area of choke, sq in.
P_1 = upstream pressure, psi
P_2 = downstream pressure, psi
g = 32.17 ft/sec²
$K = C_p/C_v$ (approx. 1.31 for natural gases)
$R = P_2/P_1 > R_o$
$R_o = \left(\dfrac{2}{K+1}\right)^{K/(K-1)}$ = critical flow ratio
γ = specific gravity (1.00 for air)
T = inlet temperature, °R

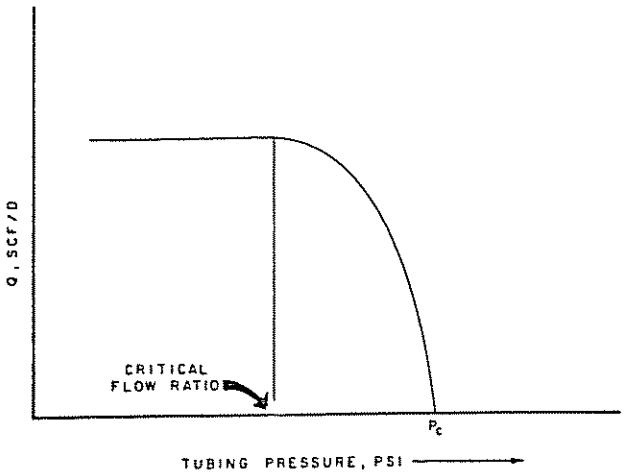


Fig. 3.217 Gas Passage Characteristics for Orifice

Charts are available (Appendix 3C) which are easily used and are usually based on some form of this equation. Since a gas lift valve is actually a series of chokes, the gas flow through a valve will be less than predicted by most charts. This should be considered when using these charts for gas flow estimates.

3.26 Throttling pressure valve

This valve is often called a continuous flow valve. The basic components in the valve are the same as the pressure-operated valve. However, modifications are made which make the valve more tubing sensitive when in the open position. This can be done by choking the entrance port of the valve which drops the casing pressure to tubing pressure or by using a tapered stem or seat which allows the port area to sense tubing pressure when the valve is open. Remember that an intermittent pressure valve senses casing pressure only when in the open position, which means the casing pressure must drop to dome pressure for the valve to close. The casing pressure does not have to drop to dome pressure for the continuous flow valve to close; in fact, this valve will close on a reduction in tubing pressure with the casing pressure held constant.

Figures 3.218 and 3.219 show two configurations for continuous flow pressure valves. The force balance equations already derived can be used to calculate the opening pressure of this type valve. However, dynamic testing must be accomplished to show how these valves react to tubing pressure when in the open position. Figure 3.220 shows the gas flow characteristics of a continuous flow valve with a tapered seat. Notice that the valve closes when the tubing pressure drops to 300 psi, even though the casing pressure is held constant at 800 psi. The linear portion of the gas flow curve is called the throttling range of the valve. The valve responds to a change in tubing pressure, throttling open for heavier loads and throttling closed as the tubing load lightens. This is sometimes called proportional response.

Dynamic testing shows that some larger ported standard pressure valves exhibit this throttling response. Figure 3.221 shows a standard pressure valve with a ¼" port. Notice the valve closes with a reduction in

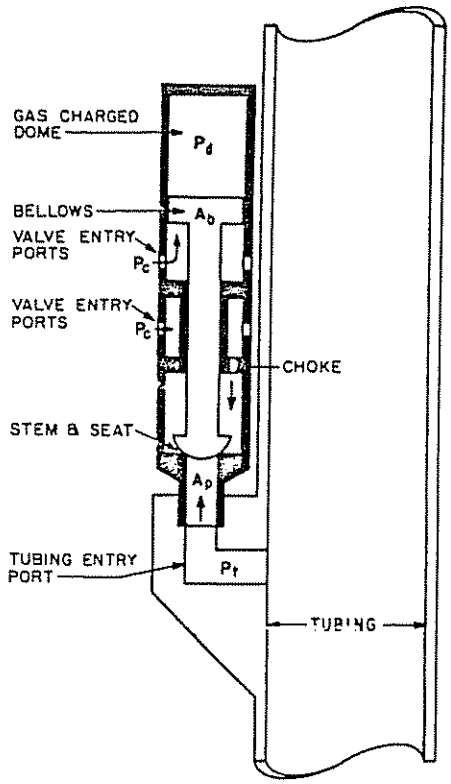


Fig. 3.218 Continuous Flow Valve

tubing pressure even though the casing pressure is held constant well above the dome pressure of the valve. Under operating conditions with gas choked at the surface or regulator controlled, there should be no drop in casing pressure to close the valve.

3.27 Fluid-operated valve

Figure 3.222 shows a fluid valve and a schematic with nomenclature. The nomenclature is the same as for a pressure valve. The valves are identical except the tubing pressure now acts on the larger area of

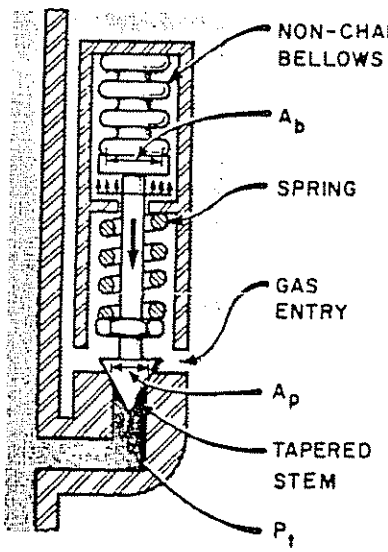


Fig. 3.219 Continuous Flow Valve

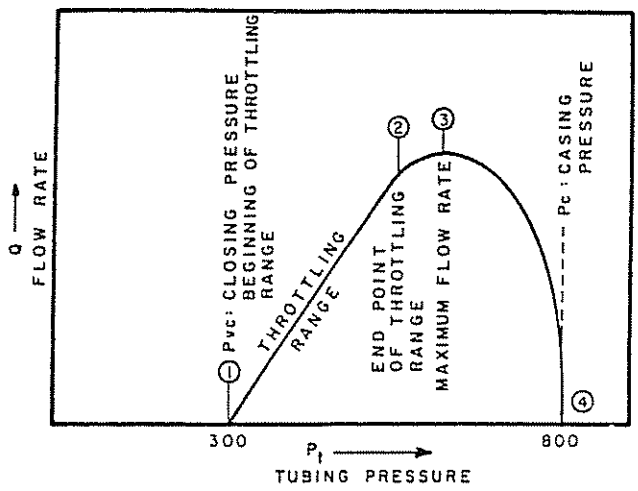


Fig. 3.220 Gas Passage Characteristics for a Continuous Flow Valve

the bellows and the casing pressure acts on the area of the port.

This is a double element valve with both a spring and dome charge supplying the closing force. Most manufacturers of the fluid valve charge the dome only when high set pressures require supplementing the spring force. General force balance equations will be derived which can be used for:

1. Spring with no dome charge
2. Dome charge with no spring
3. Combination spring and dome charge

3.271 Opening pressure of fluid valve under operating conditions

Since the fluid valve is mostly sensitive to tubing pressure, the opening pressure is defined as the tubing pressure required to open the valve under actual operating conditions. Figure 3.223 shows a valve under operating conditions and the pressures or forces which are acting on it. As with the pressure valve, the opening pressure will be investigated when the valve is in the closed position at the instant prior to the valve opening. At this instant the following equation can be written:

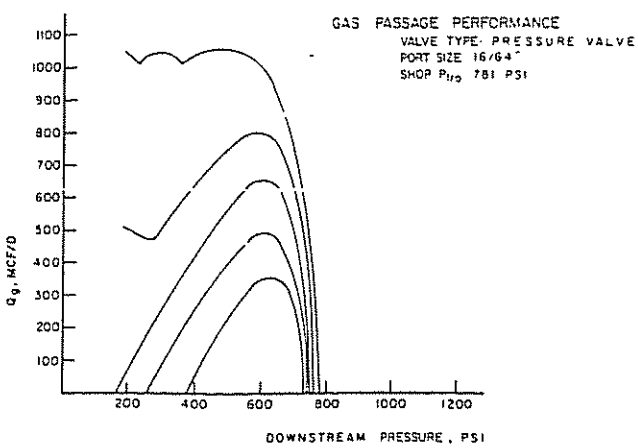


Fig. 3.221 Gas Passage Performance Curves

$$F_o = F_c \tag{3.232}$$

where:

F_o = sum of all forces trying to push the ball off seat.
F_c = sum of all forces trying to hold the ball on seat.

(1) Forces trying to hold the valve closed:

$$F_c = P_d A_b + S_t(A_b - A_p) \tag{3.233}$$

(2) Forces trying to open the valve:

$$F_o = P_c A_p + P_t(A_b - A_p) \tag{3.234}$$

substitution into Equation 2.30 gives:

$$P_c A_p + P_t(A_b - A_p) = P_d A_b + S_t(A_b - A_p)$$

solving for P_t in the manner as already shown gives:

$$P_t = \frac{P_d}{1-R} + S_t - P_c \frac{R}{1-R} \tag{3.235}$$

Equation 3.235 can calculate the tubing pressure required to open the fluid valve under actual operating conditions. Notice that Equation 3.235 for fluid valves is the same as Equation 3.226 for pressure valves except that P_c and P_t have replaced each other. The casing pressure term in Equation 3.235 $\left(P_c \frac{R}{1-R}\right)$ subtracts from the required tubing pressure to open the valve. That is, as the casing pressure increases, the tubing pressure required to open the valve decreases. The casing pressure term in Equation 3.233 is called the casing effect (C.E.).

$$C.E. = P_c \frac{R}{1-R} \tag{3.236}$$

The ratio $\frac{R}{1-R}$ in a fluid valve is called the *casing effect factor* (C.E.F.) and is reported by most manufacturers as a percentage.

$$C.E.F. = \frac{R}{1-R} \tag{3.237}$$

$$C.E. = P_c(C.E.F.) \tag{3.238}$$

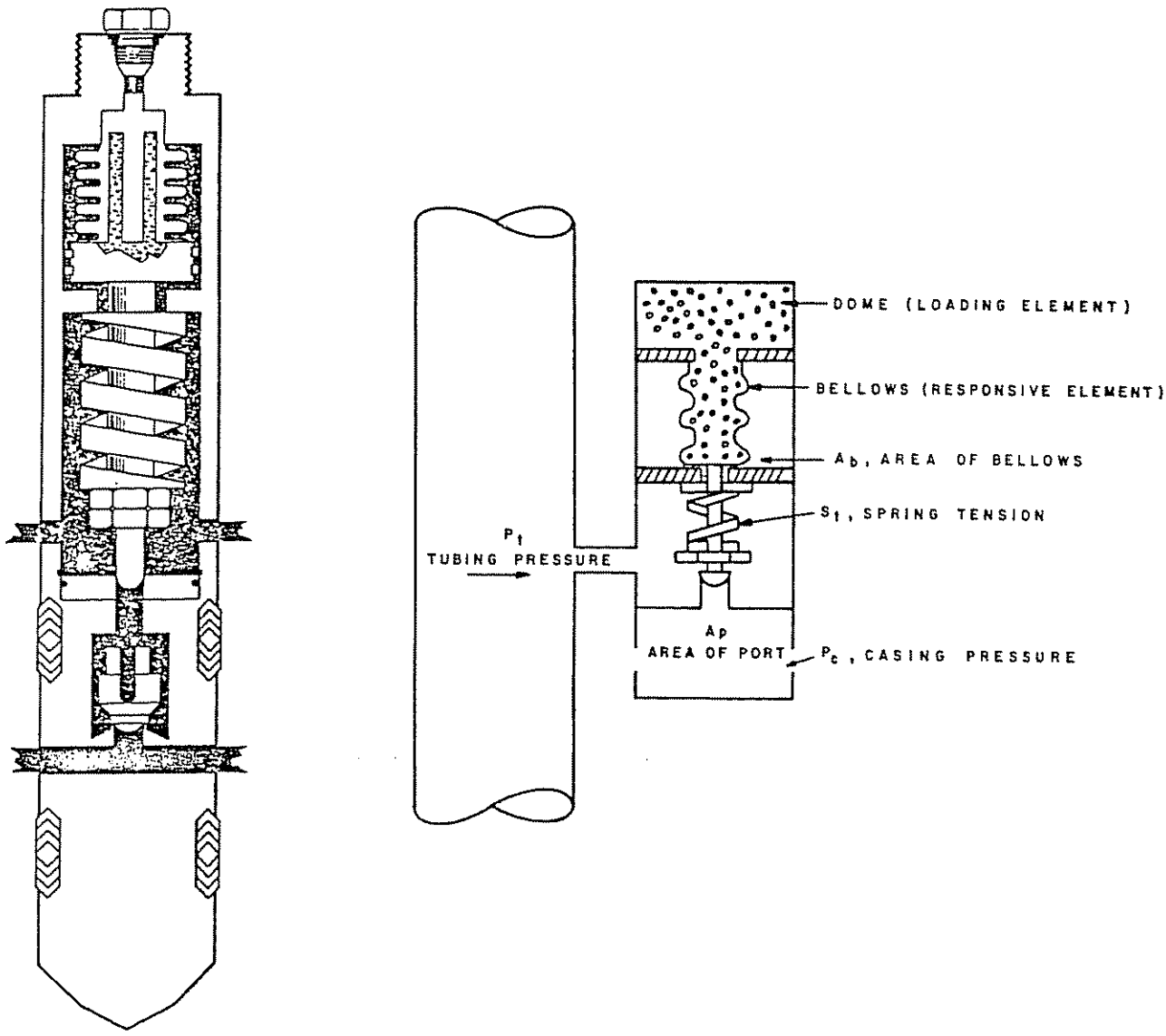


Fig. 3.222 Fluid Operated Valve

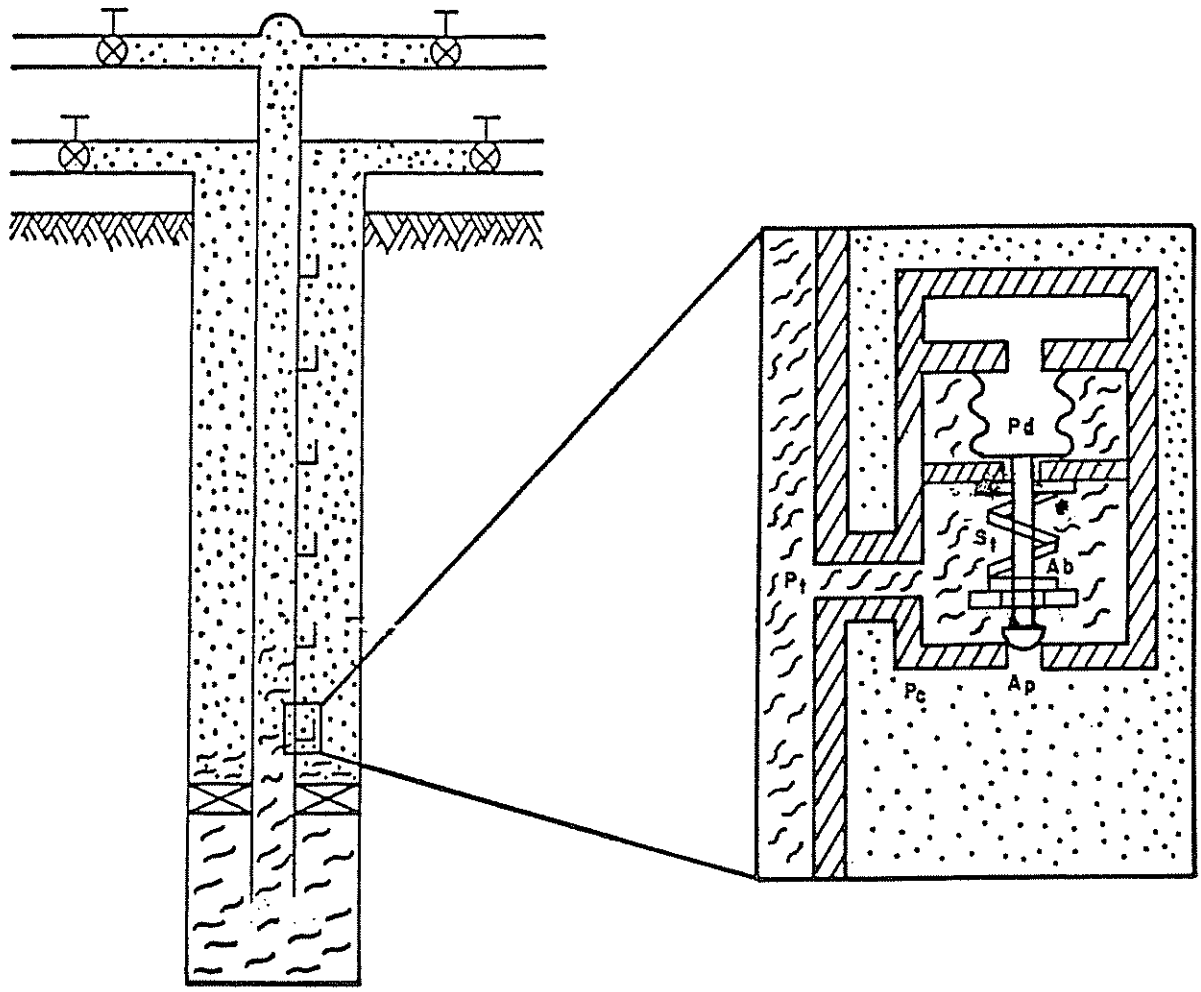


Fig. 3.223 Fluid Valve Under Operating Conditions (Valve Closed)

3.272 Closing pressure of valve under operating conditions

Figure 3.224 shows a fluid valve in the open position under operating conditions. When the valve is in the open position, the pressure under the ball is assumed to be tubing pressure, not casing pressure. This means the port in a fluid valve must be small enough to drop the casing pressure to tubing pressure as the gas flows through the valve.

Since pressure drop across an orifice is related to gas flow volume, larger ports can be used if casing to tubing differential is large enough to allow relatively high gas volumes to be passed (see "Dynamic Conditions").

At the instant before the valve closes, the following force balance equation can be written:

$$F_o = F_c \tag{3.239}$$

(1) Forces trying to close the valve:

$$F_c = P_d\, A_b + S_t (A_b - A_p) \tag{3.240}$$

(2) Forces trying to hold the valve open:

$$F_o = P_t\, A_p + P_t (A_b - A_p)$$

or

$$F_o = P_t\, A_b \tag{3.241}$$

substitution into Equation 3.239 gives:

$$P_t\, A_b = P_d\, A_b + S_t (A_b - A_p)$$

solving for P_t gives:

$$P_t = P_d + S_t (1 - R)$$

To determine the pressure in the tubing at which the valve closes, let:

$$P_{vc} = P_t$$
$$P_{vc} = P_d + S_t (1 - R) \tag{3.242}$$

The designer will estimate a value for P_{vc} and calculate all other values based on that estimate.

3.273 Test rack opening pressure.

Once the opening and closing pressures are established, the fluid valve must be set at an opening pressure in the shop which corresponds to the desired opening pressure in the well. As with the pressure valve, this is called the test rack opening pressure (P_{tro}). Since there is no casing pressure applied to a fluid valve when in a test rack, P_c can be set equal to zero in Equation 3.235 which results in:

$$P_t = \frac{P_d}{1 - R} + S_t \tag{3.243}$$

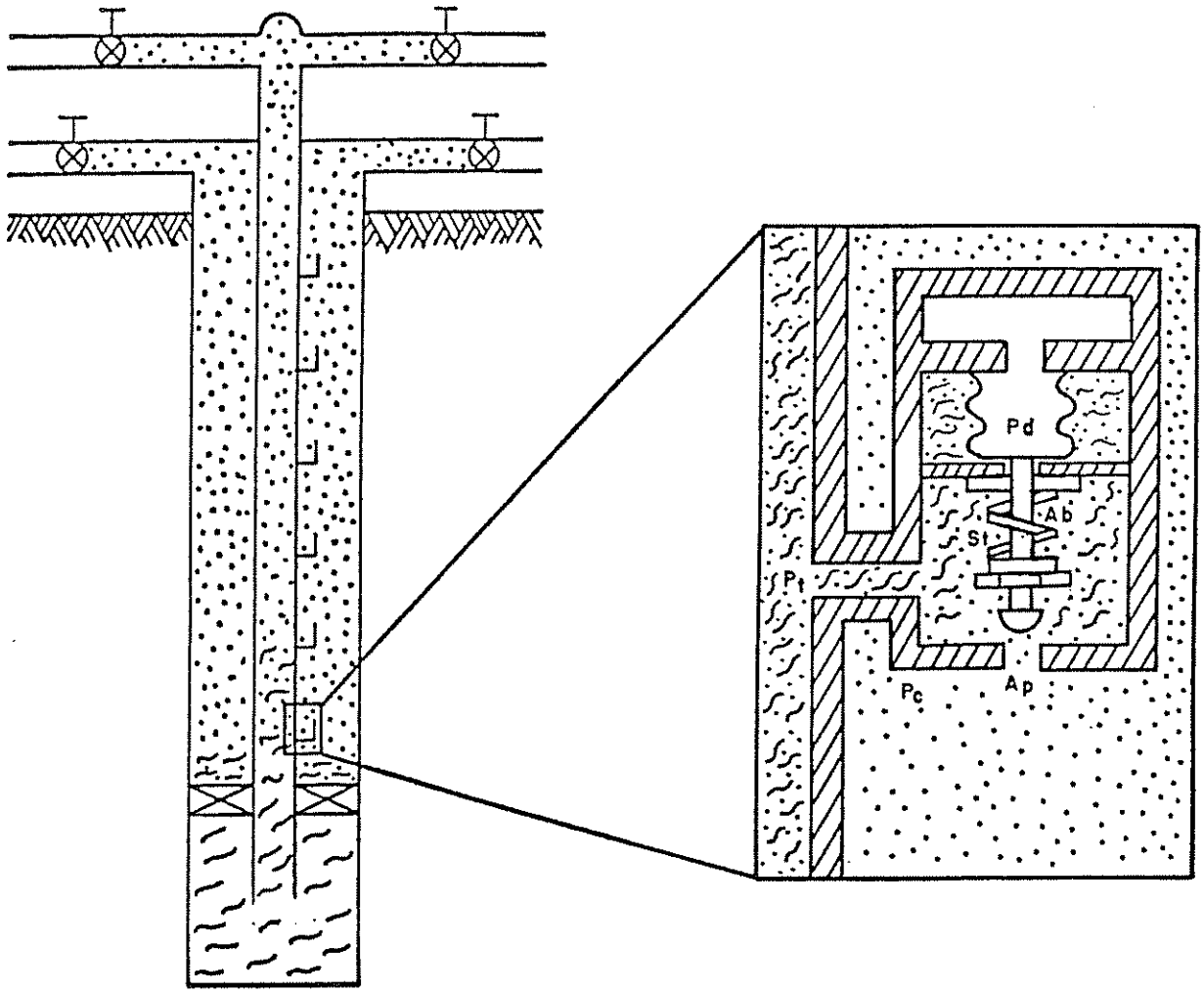


Fig. 3.224 Fluid Valve Under Operating Conditions (Valve Open)

If the pressure in the dome is corrected to 60°F. (or any other temperature for shop basis) Equation 3.243 becomes the equation for "test rack opening" pressure.

$$P_{tro} = \frac{P_d @ 60°F}{1 - R} + S_t \qquad (3.244)$$

Notice that when the dome charge is zero:

$$P_{tro} = S_t$$

When a dome charge is needed to supplement the spring setting, some constant S_t is selected by the designer. This selection depends on the spring constant of the valve being used. Once S_t is selected, the valve is opened at that pressure in a test rack with zero dome charge. This sets S_t in the valve. No temperature correction is needed for S_t.

EXAMPLE PROBLEM #8

Double element fluid valve with dome charge and $S_t = 600$ psi.
Given:

Depth of valve = 8000 ft
Temperature @ valve = 180°F.
$P_{vc} = 1800$ psi
$R = 0.0407$
$1 - R = 0.9593$

Find:
(1) P_d, dome pressure @ depth
(2) P_d @ 60°F.
(3) Test rack opening pressure, P_{tro}

Solution:

(1) Dome pressure @ depth from Equation 2.42:
$P_d = P_{vc} - S_t(1 - R)$
$= 1800 - 600\,(0.9593)$
$= 1224$ psi

(2) Using Appendix 3D (Table 3D.1) for a temperature at the valve of 180°F:

$$C_t = 0.795 \frac{P_d @ 60°F}{P_d @ 180°F}$$

P_d @ 60°F $= 0.795\,(1224) = 973$ psi

(3) Test rack opening pressure:

$$P_{tro} = \frac{973}{0.9593} + 600 = 1614 \text{ psi}$$

Setting procedure is the same as explained for a dual element pressure valve.

3.274 Summary examples: fluid valve

SUMMARY EXAMPLE: double element fluid valve with no dome charge.

Given data:

Depth of valve = 8000 ft
Surface operating gas pressure = 800 psi
Specific gravity of injection gas = 0.7
Temperature at surface = 100°F.
Temperature at 8000 ft = 180°F.
P_{vc} = 800 psi
R = 0.0407
1 − R = 0.9593
Pressure in tubing at valve depth = P_t = 750 psi

Find: (A) Opening pressure @ valve depth
(B) Test rack opening pressure

(1) The casing pressure at depth corresponding to 800 psi at surface is calculated as before to equal 955 psi. Since there will be no dome charge:

$P_d = 0$ psi

From Equation 2.40:

$$S_t = \frac{P_{vc}}{1-R} \text{ when } P_d = 0$$

Therefore, for opening pressure at depth:

$$P_t = \frac{P_{vc}}{1-R} - \frac{P_c R}{1-R}$$

$$P_t = \frac{800}{0.9593} - 955\left(\frac{0.0407}{0.9593}\right) = 793 \text{ psi}$$

(2) When $P_d = 0$

$$P_{tro} = S_t = \frac{P_{vc}}{1-R}$$

$$P_{tro} = \frac{800}{0.9593} = 834 \text{ psi}$$

SUMMARY EXAMPLE: double element fluid valve with dome charge and no spring

Given data:
Same as previous example

Find:
(A) Opening pressure @ valve depth
(B) Test rack opening pressure

(1) The casing presure at depth corresponding to 800 psi at surface is 955 psi.
Since there is no spring:
$S_t = 0$ psi

From Equation 3.235:

$$P_t = \frac{P_d}{1-R} - P_c \frac{R}{1-R}$$

From Equation 3.24:
$P_{vc} = P_d$

Therefore, for opening pressure at depth:

$$P_t = \frac{P_d}{1-R} - P_c \frac{R}{1-R}$$

$$P_t = \frac{800}{0.9593} - 955\,\frac{0.0407}{0.9593} = 793 \text{ psi}$$

(2) When $S_t = 0$:

$$P_{tro} = \frac{P_d @ 60°F}{1-R}$$

The dome pressure @ 60°F must be determined. Using chart 3D.1 for a temperature at the valve of 180°F: (Appendix 3D)

$$C_t = 0.795 = \frac{P_d @ 60°F}{P_d @ 180°F}$$

Therefore:

$$P_d @ 60°F. = (0.795)\, P_d @ 180°F.$$
$$= 0.795\,(800) = 636 \text{ psi}$$

$$P_{tro} = \frac{636}{0.9593} = 663 \text{ psi}$$

SUMMARY EXAMPLE: double element fluid valve with dome charge and spring.

Given data:
Same as previous example
Set S_t = 300 psi

Find:
(A) Opening pressure @ valve depth
(B) Test rack opening pressure
(C) Setting procedure

Solution:

(1) The casing pressure at depth corresponding to 800 psi at surface is 955 psi

From Equation 3.242

$P_d = P_{vc} - S_t(1 - R)$
$P_d = 800 - 300(0.9593) = 512$ psi

Therefore the opening pressure at depth from Equation 3.23 is:

$$P_t = \frac{512}{0.9593} + 300 - 955\,\frac{0.0407}{0.9593} = 793 \text{ psi}$$

(2) For P_{tro} the dome pressure @ 60°F must be determined:
using chart 3D.1 for a temperature at the valve of 180°F:

$$C_t = 0.795 = \frac{P_d @ 60°F}{P_d @ 180°F}$$

Therefore:

$$P_d @ 60°F = 0.795\ P_d @ 180°F.$$
$$= 0.795\,(512) = 407 \text{ psi}$$

Therefore:

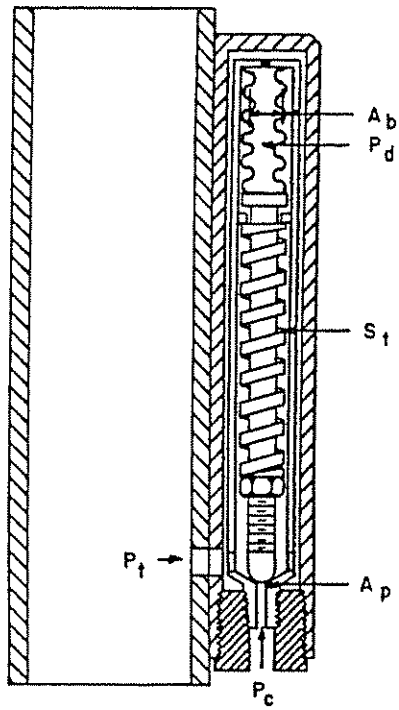
$$P_{tro} = \frac{407}{0.9593} + 300 = 724 \text{ psi}$$

(3) Setting procedure.
Remember that $P_{tro} = S_t$ when $P_d = 0$.

Step 1: Install valve in test rack. Adjust spring setting until the valve opens with 300 psi applied pressure. This sets S_t = 300 psi.

Step 2: Pressure up the dome and cool valve to 60°F.

Step 3: Bleed pressure off dome until valve opens with 724 psi applied pressure.



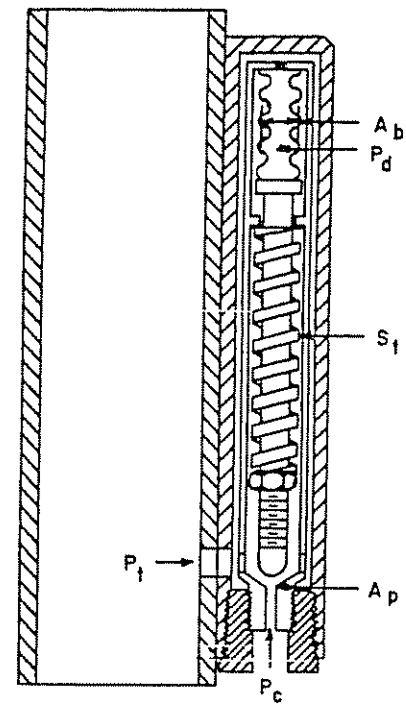


Fig. 3.225 Typical Fluid Operated Valve

3.275 Gas passage characteristics

The design techniques used for fluid valves are based on the gas passage characteristics of the fluid valve (see Fig. 3.225). The port size must be small enough to create a pressure drop across the valve. This results in the entire bellows area sensing only tubing pressure when the valve is in the open position.

Figure 3.226 illustrates the flow rate response of the valve as a function of tubing pressure. If the tubing pressure is within the throttling range of the valve, the pressure drop across the port coupled with the spring effect causes the valve to respond proportionally to increases and decreases in tubing pressure. Gas flow tests indicate that in the interval between Points 1 and 2 of Figure 3.226 some gas flow occurs, but the quantity is negligible in relation to the overall capacity of the valve. In fact, dynamic testing indicates that for larger port sizes the throttling range actually starts at the opening pressure.

The fluid valve of Figure 3.225 acts as a variable orifice with the orifice size increasing with increasing tubing pressure and decreasing with decreasing tubing pressure. This proportional response maintains the established flowing tubing gradient.

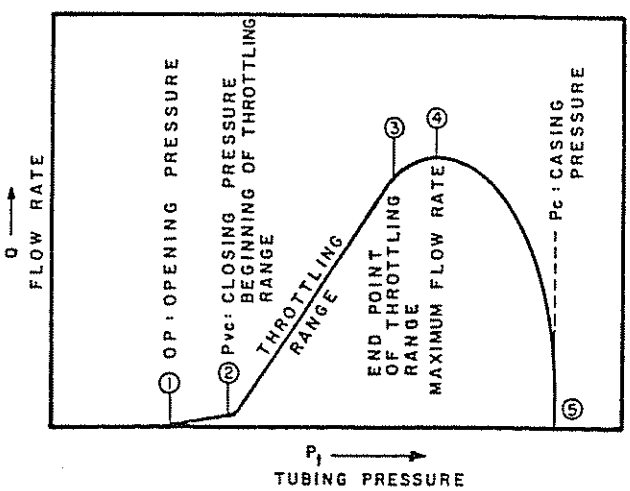


Fig. 3.226 Flow Rate Response Curve for a Fluid Valve

3.276 Differential valve

The differential valve shown schematically in Figure 3.227 was the first widely-used gas lift valve designed to unload a well. As with the fluid valve, the opening and closing pressure will be the tubing pressure defined in terms of the casing pressure.

3.2761 Opening pressure of differential valve under actual operating conditions

Figure 3.227 shows a valve under operating conditions and the pressures or forces acting on it. The following force balance equation can be written for the valve in the closed position at the instant just prior to the valve opening:

$$F_o = F_c \qquad (3.245)$$

where:

F_o = sum of all forces trying to push the ball off seat.
F_c = sum of all forces trying to hold the ball on seat.

Note: The spring in the differential valve is trying to push the ball off of the seat.

Forces trying to hold the valve closed:

$$F_c = P_c A_p \qquad (3.246)$$

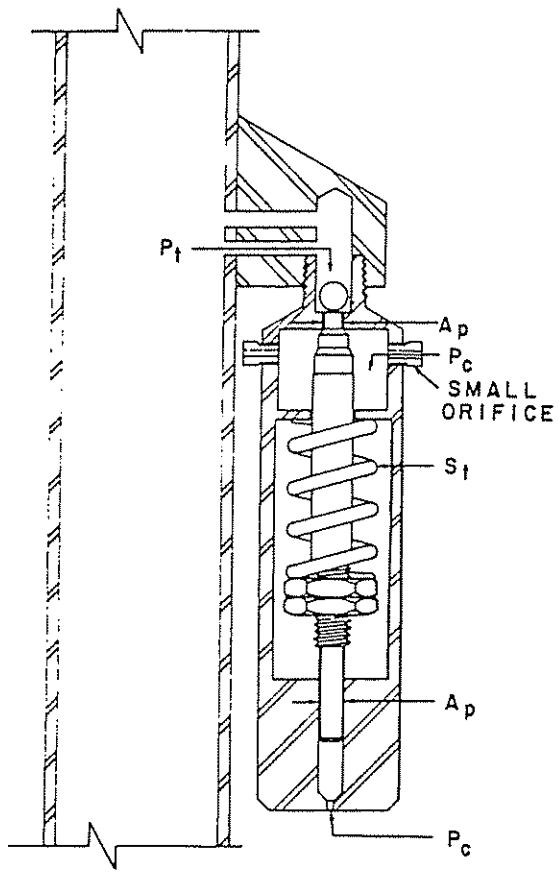


Fig. 3.227 Differential Valve

Forces trying to open the valve:

$$F_o = S_t A_p + P_t A_p \quad (3.247)$$

Substituting into Equation 3.243 gives:

$$P_t A_p + S_t A_p = P_c A_p \quad (3.248)$$

where:

A_p = valve seat area and area of piston at top of valve sq in.
P_c = casing pressure at valve depth when the valve opens, psi
P_t = tubing pressure at valve depth required to open the valve under operating conditions, psi
S_t = An equivalent pressure caused by spring tension assumed to be acting on the area A_p

solving for P_t, divide each term by A_p:

$$P_t + S_t = P_{vo} \quad (3.249)$$

or:

$$P_t = P_{vo} - S_t \quad (3.250)$$

Equation 3.248 shows that the tubing pressure required to open the valve will be the casing pressure minus the constant spring pressures effect. The spring pressure effect is called the differential setting.

If the spring setting of a differential valve is 150 psi and the casing pressure is 850 psi, the valve will open when the tubing pressure increases to 700 psi. Notice that the differential valve is a normally open valve.

3.2762 Closing pressure of differential valve under operating conditions

When the valve is in the open position, the pressure under the ball is assumed to be tubing pressure not casing pressure. For this to be true, the chokes installed in the housing must be small in relation to the seating area for the ball. At the instant just prior to the valve closing, the following force balance equation may be written:

$$P_o = F_c \quad (3.251)$$

Since the pressure and areas are assumed to be the same as when the valve is closed, the pressure in the tubing required to close the valve will be the same as the opening pressure:

$$P_t = P_c - S_t \quad (3.252)$$

If the choke in the housing is too large to drop P_c to P_t as gas passes through the valve, the valve will not close. For a ¼" port, a housing choke no larger than $^{10}/_{64}$" can be used. Notice that both the opening and closing pressure will increase with increased casing pressure.

When running differential valves with a packer, some provision should be made to U-tube the "kill" fluid out of the annular space. If this is not done, all the valves may lock closed and manual adjustments will be necessary to unload the well. Some type of orifice or other type of valve can be set at the bottom location to accomplish this goal.

To explain why this is necessary, assume that 800 psi is available for the operation and that the well is standing full of 0.5 psi/ft kill fluid. If the first valve is at 1880 ft and 800 psi is placed on the casing side at the surface, the casing pressure at the depth of the first valve will be 800 + 0.5 (1880) = 1740 psi. The tubing pressure at this point with no back pressure is 940 psi or 1040 psi with 100 psi back pressure. A differential valve with a 150 psi spring cannot open until the tubing pressure is within 150 psi of the casing pressure. Therefore, it would take 1740 − 150 = 1590 psi in the tubing to open the valve allowing gas entry or U-tubing of liquids. The tubing pressure would need to be boosted or the casing pressure bled off. Although this well can eventually be unloaded, it is better to provide communication between the tubing and casing in another manner.

3.2763 Test rack setting procedure for differential valve

The valve is installed in a closed test rack with the seat exposed to atmospheric pressure. The closed volume is then pressured up rapidly, and the differential caused by the expanding gas closes the valve. Once the valve is closed, the pressure under the seat is atmospheric (psig). The pressure trapped in the enclosed volume is then bled off slowly until the valve begins to pass gas. This pressure is the spring setting of the valve. Adjustments can be made with the lock nuts shown in Figure 3.227. The setting does not have to be corrected for temperature at well conditions.

3.28 Combination valves

Figure 3.228 shows a type of fluid-operated valve that is opened by tubing pressure and closed by a reduc-

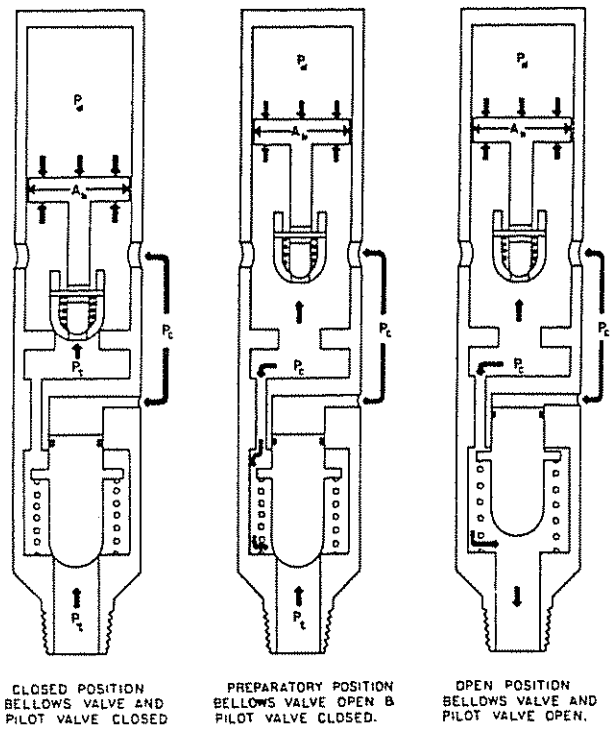


Fig. 3.228 Fluid-Operated Valve with Special Pilot

tion in casing or tubing pressure. This particular device eliminates having to place a restricted entry port on the injection gas inlet. The upper portion of this valve is essentially the same as found in Section 3.25; the bottom pilot device will be described here.

The fluid pilot is a differential pressure device incorporating a stem and spring. The upper end of the stem has the same cross-sectional area as the lower end that seats on the pilot seat when the valve is closed. The upper end of the stem is sealed from pressure within the pilot by means of an O ring.

Holes in the pilot housing admit casing pressure (P_c) to act on the area of the stem at the upper end. The spring arrangement is such that the force exerted by the spring is compression and acts to hold the stem in the upward position. This is the open position for the pilot. Casing pressure acts to move the stem to the down or closed position when the force it exerts exceeds the force of the spring compression.

In the well, under operating conditions the force of the casing pressure always exceeds the spring compression. The fluid pilot will only open when tubing pressure (P_t) acting on the pilot port area is sufficient (in conjunction with the spring compression) to overcome the casing pressure force and move the stem up to the open position.

At the instant the opening casing pressure (P_c) replaces tubing pressure (P_t) beneath the seat, the pilot opens completely providing instantaneous operation for intermittent lift. Should the well feed-in rate maintain a sufficiently high flowing pressure, the valve will remain open. This is especially true when the bellows valve is equipped with a tapered stem.

Depending upon the design problem, different spring tensions may be selected. Spring tension determines the size of the tubing fluid load to be lifted. The fluid load is the difference between the casing pressure at the valve and the spring compression pressure equivalent. Spring ratings are categorized by pressure equivalents; thus, a 250 psig spring is the equivalent of 250 psig of well pressure acting on the pilot seat.

For unloading valves and for operational intermittent lift, a spring with 150-120 psig range difference between P_c and P_t gives best results. Stronger springs may be installed to lift lighter loads if desired. While higher differentials than 250 psig will give greater sweep efficiency (greater surface recovery of total fluid head started at point of lift) the GLR's (gas-liquid ratios) are also considerably higher because of an increased number of cycles necessary to lift the same quantity of fluid. In a low BHP well, it may still be desirable to lift lighter loads to maintain sufficient drawdown.

Valve spring compression equivalents in the continuous flow range are not selected on the same basis as for intermittent flow. They will normally be set in a decreasing manner, i.e. 350, 300, 250, 200, 150 psi, etc., going down the well. This allows the valve near the point of gas injection to open on a low differential between casing and tubing pressures.

3.29 Dynamic considerations

3.291 Bellows travel and protection

The bellows in a gas lift valve is the element which stretches or compresses to allow the ball to move on and off the seat. To protect the bellows from failure due to over-travel and external pressure, most gas lift valves are equipped to limit bellows travel. Examples of bellows protection methods are

(1) Support rings
(2) Hydraulic deformation
(3) Confined liquid
(4) Mechanical stops
(5) Isolation of bellows

Of course, the maximum travel of a gas lift valve is affected by the bellows protection system used, depending on the port size; the stem must travel a minimum distance to reach an equivalent fully-open position (see Fig. 3.229). Gas flow can be severly restricted if full bellows travel stops or is restricted prior to reaching the full open position. Dynamic testing shows that gas flow will continue to increase with additional travel for some port configurations, even though the theoretical equivalent orifice size has been exceeded.

3.29 Bellows load rate

The load rate is a measure of the force required to move a bellows a certain distance. When discussing gas lift valves, we define load rate as the pressure change required to affect a given stem travel in in./psi. Usually, the load rate of a valve should be linear over the operating range. Stacking of bellows convolutions reduces the load rate. Of course the load rate will be zero when the bellows is fully stacked. For valves with small initial dome volumes, compression of the bellows decreases load rate due to pressure-volume relationships. This problem amplifies when valves are charged with high dome pressures (greater than 1500 psi). Pre-

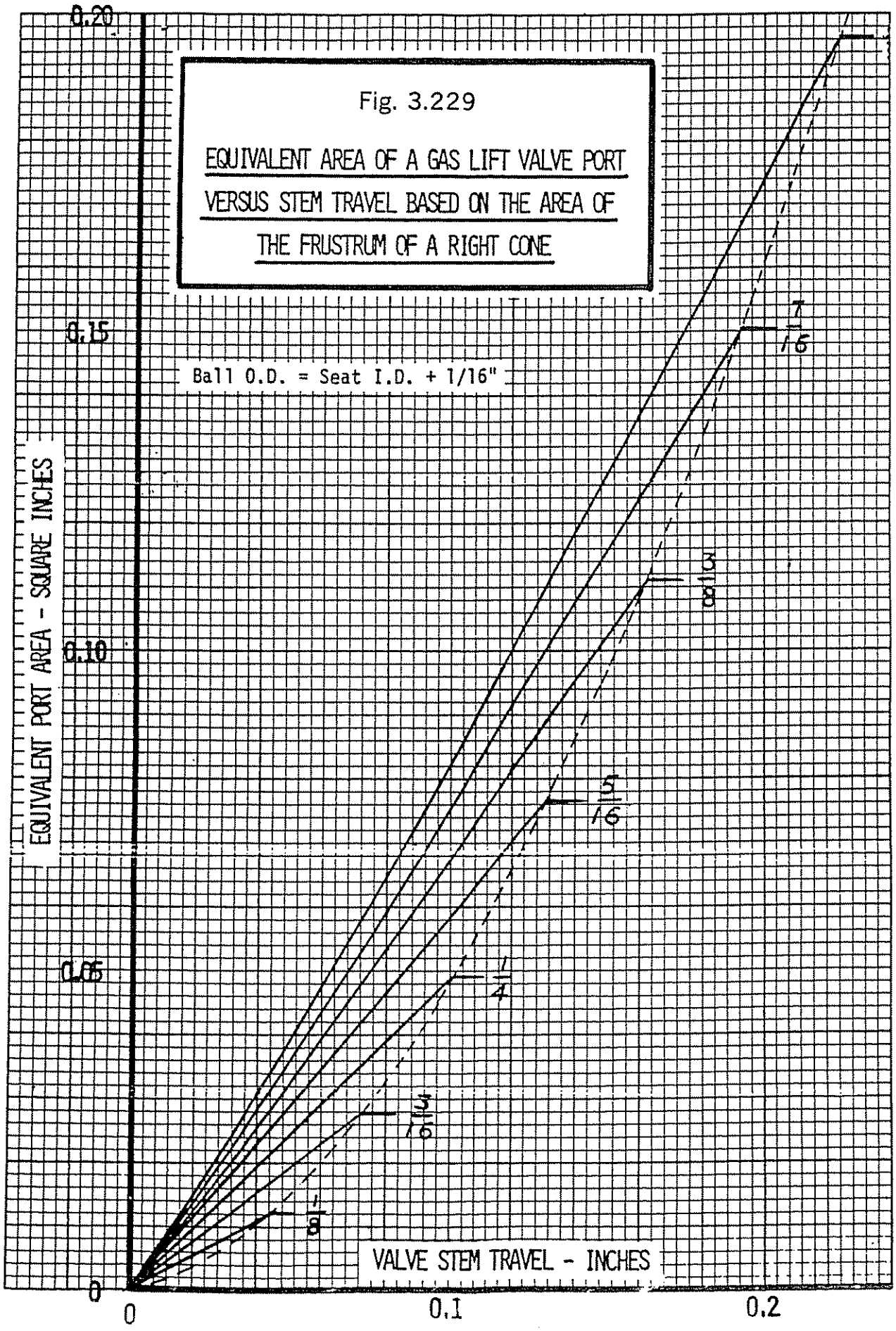


Fig. 3.229 (Courtesy H.W. Winkler)

mature reduction in load rate can significantly reduce gas passage through a gas lift valve.

3.293 Pressure valves

The single element pressure-charged intermittent valve was assumed to be affected only by casing pressure when in the open position. For this to be true, the pressure under the ball must be casing pressure. Under actual well conditions, when the ball approaches the seat upon a reduction in casing pressure, the pressure under the ball must drop approaching tubing pressure which will cause the valve to close at a higher casing pressure than predicted by the force balance equations. This does not happen in smaller ported valves when downstream choking is apparent. Figure 3.230 shows a single element pressure valve that closed on a reduction in tubing pressure with the casing pressure held constant.

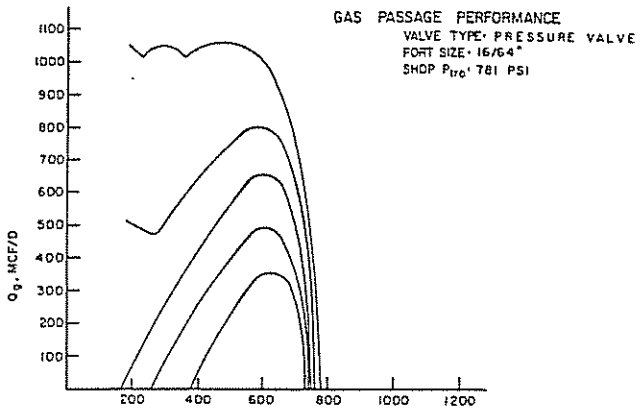


Fig. 3.230 Gas Passage Performance Curves (Casing Pressure Constant)

3.210 Class Problems

(1) Given:

Standard bellows charged valve 1½" O.D. × ⅜" port

Valve depth = 8000 ft
$\gamma_g = 0.7$
Surface opening pressure = 800 psi
Temp surface = 100°F.
Temp 8000' = 180°F.
Press in tubing = $P_{vo} - 300$

Find the test rack opening pressure if the valve is charged with nitrogen.

(a) 60°F.
(b) 80°F.
(Pressure operated valve)

(2) Work problem #1 for a 1" O.D. valve and $^5/_{16}$" port.

(3) Given:

1" O.D. pressure charged bellows valve with $^5/_{16}$" port

$\gamma_g = 0.65$
$P_{so} = 700$ psi
Temp top = 100°F.
Temp bottom = 160°F.
Depth = 7000 ft
Tubing load = 600 psi

Find: P_{TRO} for nitrogen charge

(a) 60°F.
(b) 80°F.
(Pressure operated valve)

(4) Given:

Depth = 7000 ft
$P_{so} = 900$ psi
Tubing load = 900 psi
Temp bottom = 160°F.
1½" O.D. valve with ¼" port
Pressure operated valve

Find:

(a) P_{TRO} for bellows charged valve
(b) P_{TRO} for spring charged valve
(bellows charge = 0)

(5) Given:

Depth = 6000 ft
$P_{so} = 700$ psi
Valve 1½" O.D. × ⅜" port
Temp surface = 80°F.
Temp 6000 ft = 160°F.
$\gamma_g = 0.65$
Tubing load = 600 psi
Pressure operated valve

Find: P_{TRO}

(a) Spring charged valve
(b) Bellows charged valve with nitrogen at 60°F.
(c) Combination bellows and spring charged valve (400 psi spring)

(6) Given:

$P_{TRO} = 850$ psig
1½" O.D. valve × $^7/_{16}$" port
Depth valve = 7000 ft
Temp bottom = 180°F.
Temp surface = 100°F.
$\gamma_g = 0.65$
Tubing Load = 650 psi
(Pressure operated valve)

Find:

Surface opening pressure and surface closing pressure for

(a) Bellows charged valve (Nitrogen = 60°F.)
(b) Spring charged valve (Zero bellows charge)
(c) Spring charged valve in combination with bellows charge (600 psi spring in test rack)

(7) For data of Problem 6, find the spread in the valve on the well for tubing pressures of 200, 400, and 600 psig for both a bellows charged and spring charged valve.

What is the tubing effect factor for this valve?

(8) Given:

Depth = 3000 ft
1½" × 5/16" valve
$\gamma_g = 0.65$
Temp top = 75°F.
Temp bottom = 120°F.
P_{so} = 350 psi
Tubing load = 50 psi
(chamber installation)
Bellows charged with
nitrogen at 60°F.

Find for pressure operated valve:
(a) P_{vo} at 3000 ft
(b) P_{vc} at 3000 ft.
(c) P_{tro}

(9) Given:

Intermittent well
$P_t = P_{vo} - 300$
7" casing
2" tubing
Depth = 8000 ft
P_{wh} = 50 psig
P_{so} = 800 psig
$\gamma_g = 0.65$
Bellows charged valve with
nitrogen at 60°F.
1½" O.D. valve

Find for pressure operated valve:

(a) P_{vo}
(b) Gas volume per cycle
(c) Correct spread to furnish gas
(d) P_{vc}
(e) Control port size

(10) Given:

Depth = 7000 ft
P_{so} = 650 psi
$\gamma_g = 0.70$
Tubing load = 500 psi
Temp top = 80°F.
Temp valve = 160°F.
P_{wh} = 50 psi
2½" Tubing × 7" casing
Bellows charged valve with
nitrogen at 80°F.

Find for pressure operated valve:

(a) Control port size for 1½" O.D. valve
(b) Control port size for 1" O.D. valve

(11) Given:

5½" casing × 2" tubing
Depth = 7000 ft
P_{so} = 700 psig
P_{wh} = 100 psig
Temp bottom = 160°F.
$\gamma_g = 0.65$
Tubing load = 400 psig
1" O.D. valve × ¼" port
(nitrogen charge)

Find P_{tro} for shop temp = 60°F. and 80°F. (pressure operated valve)

(12) Given:

Depth = 4000 ft
P_{so} = 500 psig
Temp bottom = 135°F.
Temp top = 100°F.
Separator press = 50 psi
Tubing load = $P_{vo} - 200$
Nitrogen charge at 60°F.

Find the control port size for choke control at the surface for both 1½" and 1" O.D. valves for the following combination of casing and tubing sizes:

(a) 7" csg by 2" tubing (pressure operated valve)
(b) 5½" csg by 2" tubing (pressure operated valve)
(c) 4½" csg by 2" tubing (pressure operated valve)
(d) Assume that this valve is to be used in a chamber installation and that the tubing pressure opposite the operating valve is 50 psi.

Find the control port sizes for a, b, and c above.

(13) Given:

Depth = 9000 ft
Temp bottom = 180°F.
Temp top = 100°F.
Tubing load = 550 psi
P_{so} = 850 psi
$\gamma_g = 0.65$
7" casing × 2½" tubing
P_{wh} = 100 psi
Nitrogen charged bellows at 60°F.

Find P_{TRO} for choke control at the surface for 1½" O.D. and 1" O.D. valves.
(Pressure operated valve)

(14) Given:

Fluid operated valve
(Bellows charged with nitrogen at 60°F.)
Tubing load to open = 500 psig
Port size = ¼"
Casing pressure = 700 psig
Temp at valve = 160°F.
Valve depth = 5800 ft
$\gamma_g = 0.8$

Find test rack opening pressure (tubing press to open when $P_{casing} = 0$). Also find operating casing pressure at valve when it opens.

(15) Given:

Fluid operated valve
Test rack tubing opening
pressure = 500 psig
Depth = 7000 ft
Temp at 7000 ft = 170°F.
Temp surface = 80°F.
$\gamma_g = 0.7$
Surface casing pressure = 900 psig

Find the tubing load to open the valve at 7000 ft for a 1½" × ¼" valve and a 1" × ¼" valve.
(a) Nitrogen charge at 60°F.
(b) Spring charged valve
(c) Combination spring and bellows charge (300 psi spring)

(16) Given:

Fluid operated valve
P_d = 500 psig
1½" × ¼" valve
Casing pressure = 750 psig

Find the tubing pressure (P_{TO}) to open the valve.

Find valve closing pressure.

(17) Given:

Fluid operated valve
Depth—4000 ft
Temp at 4000' = 130°F.
Temp surface = 90°F.
Surface casing pressure = 700 psig
1½" × ¼" valve
γ_g = 0.65
Tubing load to open valve
= 500 psi at 4000 ft = P_{TO}

Find:

Test rack tubing opening pressure

(a) Nitrogen charge = 80°F.
(b) Spring charge
(c) Combination spring and bellows charge (350 psi spring)

REFERENCE

1. Craft, B. C., W. R. Holden, and E. D. Graves, Jr., *Well Design Drilling and Production*, Prentice Hall, Inc., (1962).

3.3 TYPES OF GAS LIFT INSTALLATIONS

3.31 Introduction

In general, the type of installation is influenced principally by whether or not the well is to be placed on intermittent or continuous flow. Also, the selection of the type of gas lift valve depends on whether the well is to be placed on intermittent or continuous flow. Gas lift valves are designed such that many perform similarly to a variable orifice for continuous flow, depending upon tubing pressure. They may also come to a wide-open position for intermittent flow with a large opening to supply gas quickly to the tubing string for removing a liquid slug.

Well conditions will dictate the installation type to a large degree. The type of completion, such as open hole, is also important. In addition, conditions such as possible sand production and water and/or gas coning are vital points influencing installation design. In determining the initial type of installation, judgment should be based on future well performance, including BHP decline and PI decay. Multiple completions require a more complex installation. Offshore wells require a thorough analysis because of the high workover costs.

3.32 Open Installation

In this operation a tubing string is suspended into the well with no packer. Gas is injected down the casing-tubing annular space and fluids are removed out of the tubing (Fig. 3.31). This leaves communication between the casing and tubing, thereby restricting this type of installation to good wells that will exhibit a fluid seal. Normally this will mean wells that are suitable for continuous flow only. Although it would be possible to run an installation of this type for intermittent flow, it should be done only where a packer cannot be run for some reason. At any rate, an open installation should not be run where there is any possibility that gas will be blown around the bottom of the tubing,

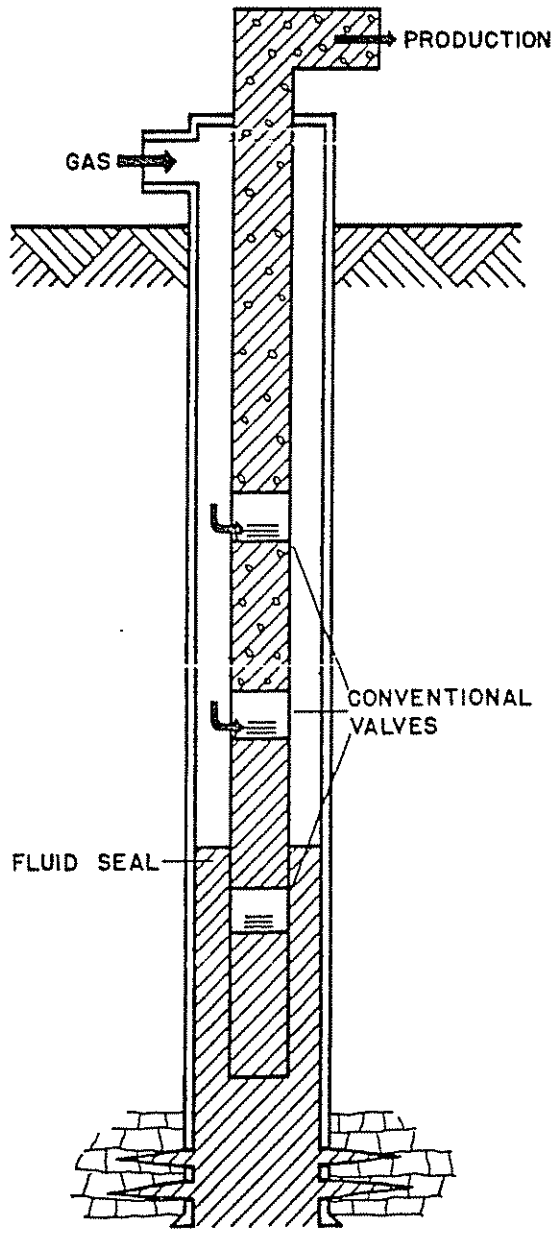


Fig. 3.31 Standard Valve Open Installation

unless the tubing can be positioned for optimum flow.

In this case, gas must be blown around the bottom of the tubing string. This offers the disadvantage of needing a high kick-off pressure for greater depths to start the well producing initially. Although this method can be used for continuous flow, it is extremely difficult to find the correct point of injection. Therefore, in most installations gas is injected above the bottom of the tubing through gas lift valves or by other means.

Another problem with open installations is that variable surface line pressure causes the fluid level to rise and fall in the casing annulus, thereby exposing all gas lift valves beneath the point of gas injection to severe fluid erosion. In pulling an installation of this type, generally all valves located beneath the point of gas injection will be cut out by fluid erosion.

Still another disadvantage of this type of installation is the well must be unloaded and restabilized every time it is shut down. Since no packer is in the well, the fluid will rise during shutdown time; this fluid must again be produced out the annulus. Also, a portion of this fluid will be U-tubed through the valves again, subjecting the valves to additional fluid erosion.

There is also the possibility that while producing the well some of the fluids will move up the annular space, through the lower gas lift valves, and into the tubing. This is because of lower friction, hence less resistance to flow in the annular space. This will tend to cause the lowermost valves to be cut out by fluid erosion.

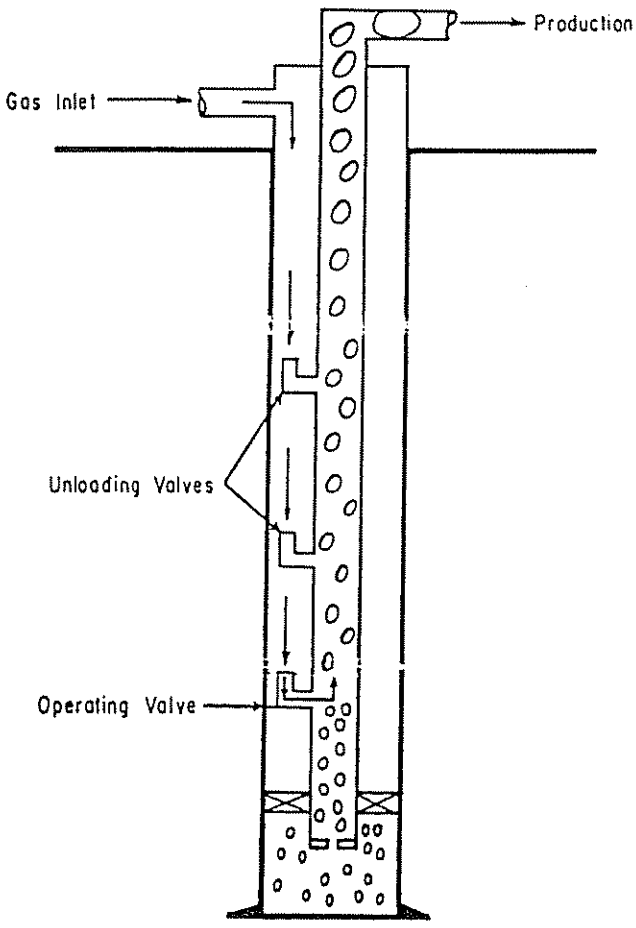


Fig. 3.32 Semi-Closed Installation

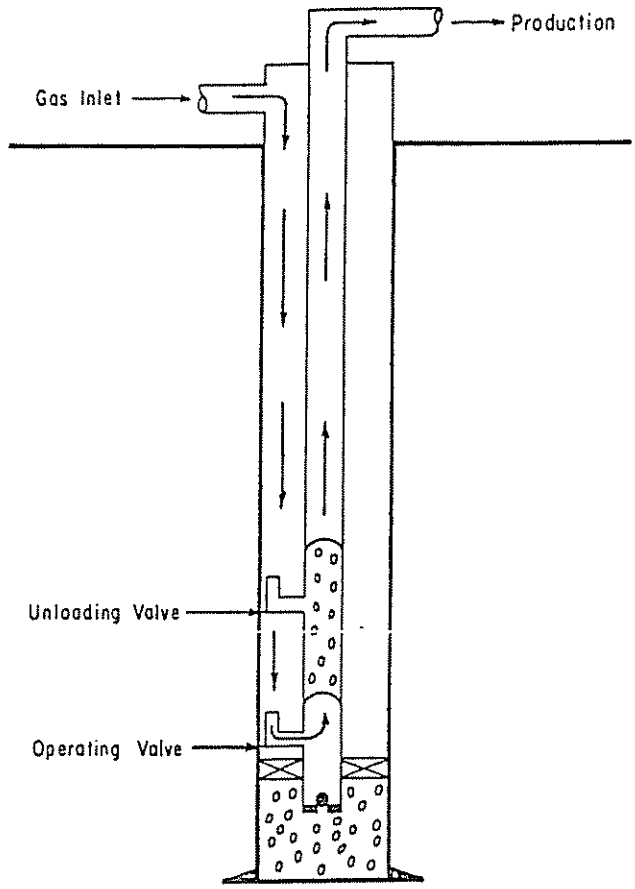


Fig. 3.33 Closed Installation for Intermittent Gas Lift

From the various disadvantages cited, it is evident that an open installation is not normally recommended. However, there are instances where running a packer is undesirable because of corrosion, bad casing, internal upset casing, etc. Where a packer cannot be run, the open installation must be run and will do a satisfactory job on most continuous flow wells. It will be very inefficient on an intermittent installation where gas is likely to blow around the bottom of the tubing string.

There is some controversy whether the casing pressure in the annular space places additional pressure on the sand face. As long as there is a fluid seal (gas cannot be blown around bottom) the same flowing BHP should be created with or without a packer in the well.

3.33 Semiclosed Installations (Fig. 3.32)

This installation is identical to the open installation, except that a packer is added to pack off between the tubing and casing. This type installation is suitable for both continuous and intermittent flow. It offers several advantages over an open installation. First, once the well has been unloaded there is no way for the fluid to come back into the casing-tubing annulus, since all the gas lift valves are run with a reverse check. Second, no fluid can leave the tubing and go into the casing space. Third, the packer prevents any fluid from coming around the bottom of the tubing and into the casing.

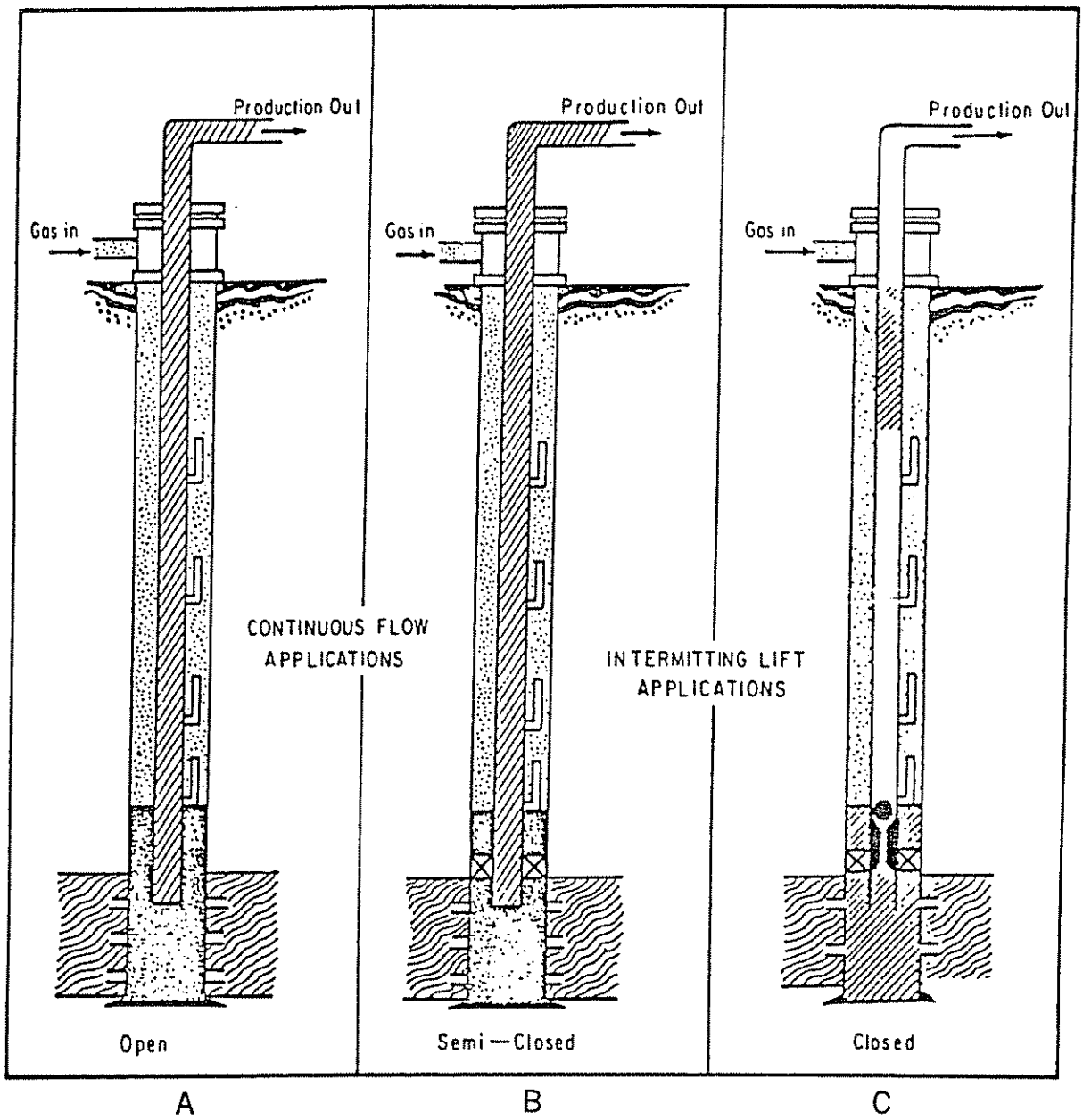


Fig. 3.34 Typical Gas Lift Installations

This type installation is also used for intermittent gas lift. Although the installation may be intermitting from the very bottom of the tubing and the top of the packer, the packer will still prevent casing gas pressure from acting on the formation. However, this installation makes no provision to keep the gas pressure (once it enters the tubing string) from acting on the formation through the tubing string, which may occur for intermittent lift.

3.34 Closed installations (Fig. 3.33)

This type of installation is similar to a semiclosed installation except that a standing valve is placed in the tubing string. Although the standing valve is normally placed at the bottom of the well, it may be placed directly below the bottom gas lift valve. This standing valve prevents the gas pressure (when injected into the tubing string) from acting on the formation.

A standing valve should be run on intermittent gas lift installations. Some provision should be made to place a standing valve in the well later if necessary. A standing valve will probably increase the daily production from a well of the intermittent type.

The open, semiclosed, and closed installations are shown in Figure 3.34 (A, B and C).

3.35 Chamber Installations

3.351 Introduction

Often a chamber installation increases the total oil production. A chamber is an ideal installation to run in

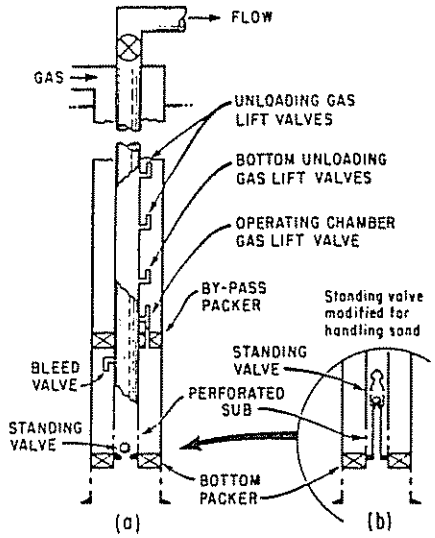


Fig. 3.35 Two-Packer Chamber Installations

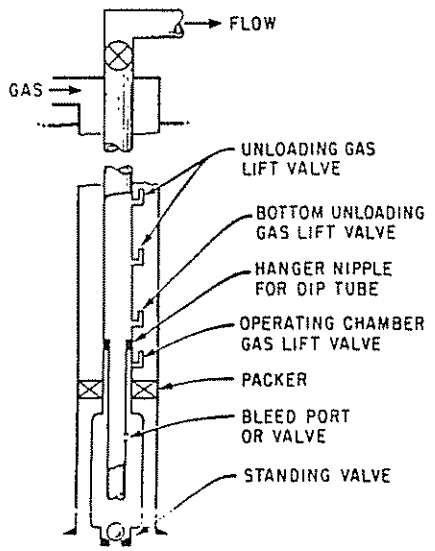


Fig. 3.36 Insert Chamber Installation

a low BHP, high PI well. This well will produce fairly high fluid volumes if a high drawdown is created on the sand face. A chamber allows the lowest flowing BHP possible to obtain by gas lift. The chamber uses the casing volume to store fluids. A "home-made" insert chamber can also be made that will hold more volume than the same length of tubing but not as much as the two-packer chamber. There are numerous instances whereby a chamber has doubled the production as compared to a standard intermittent installation. Various modifications to the standard two-packer chamber of Figure 3.35 are available.

3.352 Standard two-packer chamber (Fig. 3.35)

This type of chamber is installed to insure a large storage volume of liquids with a minimum amount of back pressure on the formation. The use of two packers may influence the decision in changing to an insert-type chamber.

The following sequence occurs in a typical chamber operation (assume that the cycle is starting just as a load of liquid is removed at the surface):

(1) The flowing BHP is increasing, thereby pushing well fluids up through the open standing valve, through the perforated nipple, and into the chamber space and the tubing string.

(2) The bleed ports are open, allowing solution gas to escape into the tubing from a point near the top of the chamber. This bleed port is important and should have ample port volume to allow the low-pressure gas to escape. Otherwise, this gas will be compressed in the upper portion of the chamber, restricting liquid entry.

(3) The chamber continues to fill with fluid until a load of liquid has accumulated.

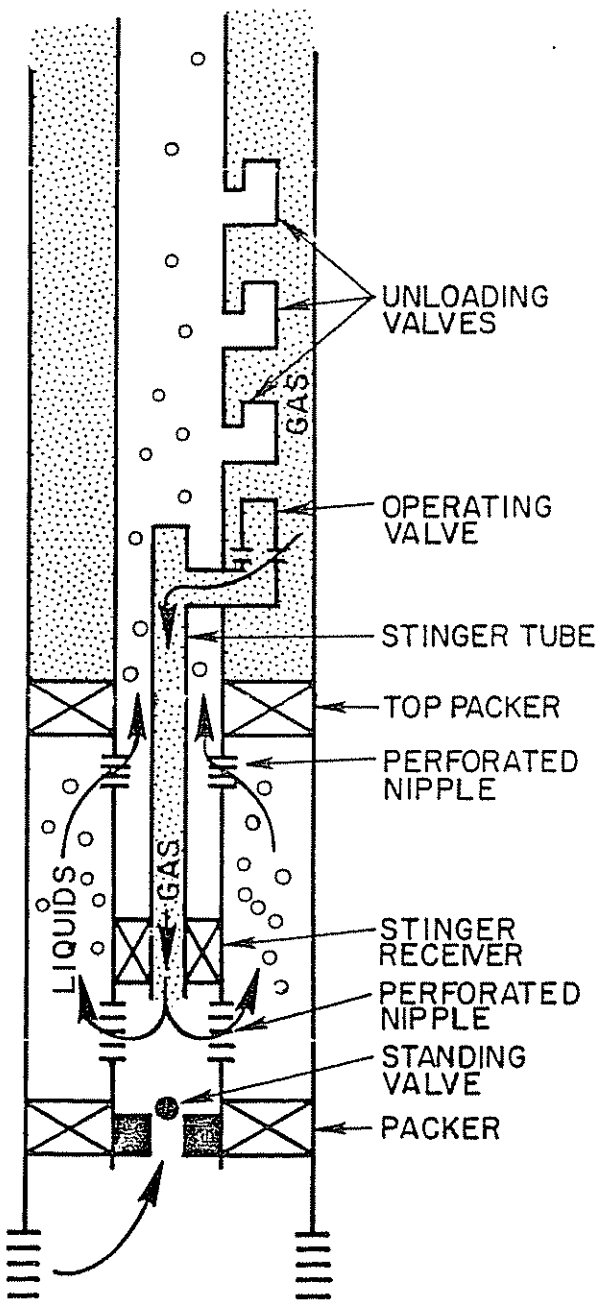


Fig. 3.37 Reverse Flow Chamber Installation

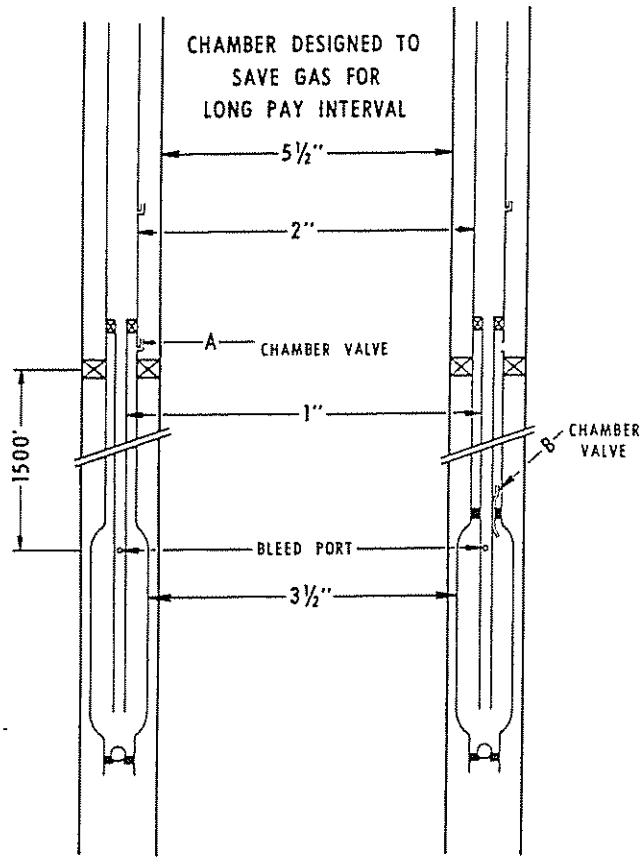


Fig. 3.38 Chamber Designed to Save Gas for Long Pay Interval

(4) Gas is injected into the well at the surface, opening the operating gas lift valve which allows the injection gas to enter on top of the accumulated liquid in the chamber. The resultant extra pressure closes the bleed ports and the standing valve.

(5) The accumulated liquid slug is U-tubed from the chamber into the tubing string and to the surface.

(6) Once the liquid slug is produced at the surface, the standing valve and bleed ports reopen, and fluids again begin to fill the chamber.

(7) The cycle is repeated.

3.353 Insert chamber (Fig. 3.36)

The insert chamber is normally used in lieu of the two-packer chamber where there is an open hole completion or long perforated interval. It assures that the greatest advantage can be taken of the existing bottom hole pressure. However, it has one disadvantage: it is made small to fit small diameter casings. It should be made from flush joint pipe to permit ease of running and pulling. Its sequence of operation is the same as for a two-packer chamber.

3.354 Reverse flow chamber installation

Figure 3.37 shows a new concept in chamber lift assuring a positive venting of all formation gas into the tubing string. In this case gas is injected down the small string as shown in Figure 3.37 and the fluids are U-tubed up the casing-tubing annular space. The fluids enter the tubing at the top of the chamber and are produced onto the surface through the tubing string. Once the chamber is empty, it refills in the normal manner but offers the advantage of permitting formation gas to vent into the tubing through the same opening where the fluids are produced. The standard chambers have a restricted venting port, depending upon the limited size of port, that can be placed in the differential valve or hole in the tubing. For wells with high gas-oil ratios, this method appears to be an excellent choice.

3.355 Special chamber to save gas for long pay interval

Figure 3.38 is a special chamber in a long pay interval (1500 ft or more) to save gas. Figure 3.38(A) shows the initial installation whereby the chamber valve is placed just above the top packer. The chamber itself is located some 1500 feet lower. For a standard cycle liquids fill in up to the top of the chamber at which time the gas lift valve opens. However, the gas lift valve is required to fill the 1500 ft of annular space between the 2" tubing and 1" dip tube with gas before the chamber fluids can be U-tubed out to the surface. For an average pressure of 500 psi, this represents some 130 scf per cycle for 100 psi change in casing pressure.

In Figure 3.38(B) the chamber valve has been moved to position B such that full gas pressure is maintained in the 1500-ft space at all times. The gas lift valve is now located at the very top of the chamber, and the gas volume required is reduced to the normal requirements for a chamber. For several wells in one field, this represents quite a savings in gas volumes. This type of

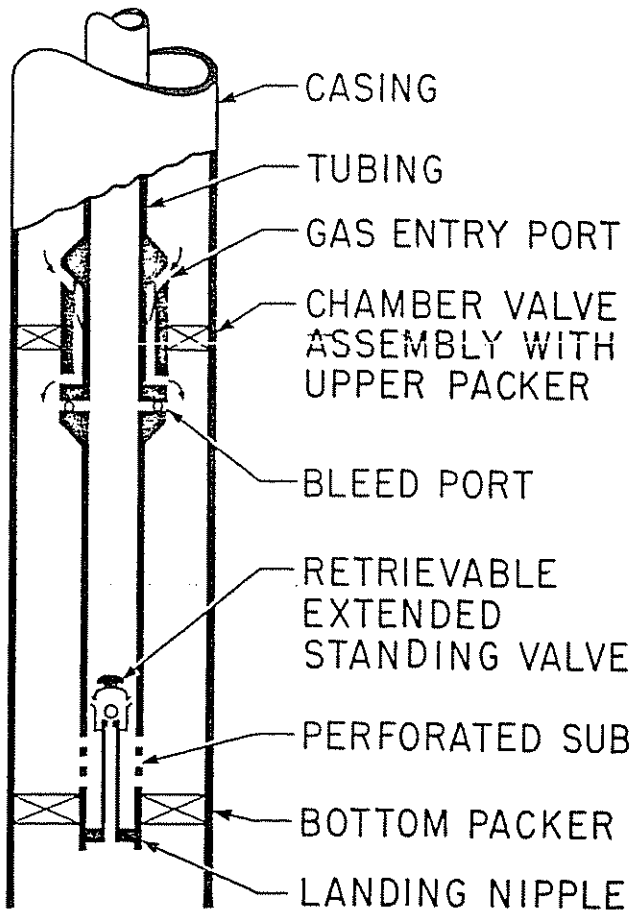


Fig. 3.39 Special Installation for Handling Sand

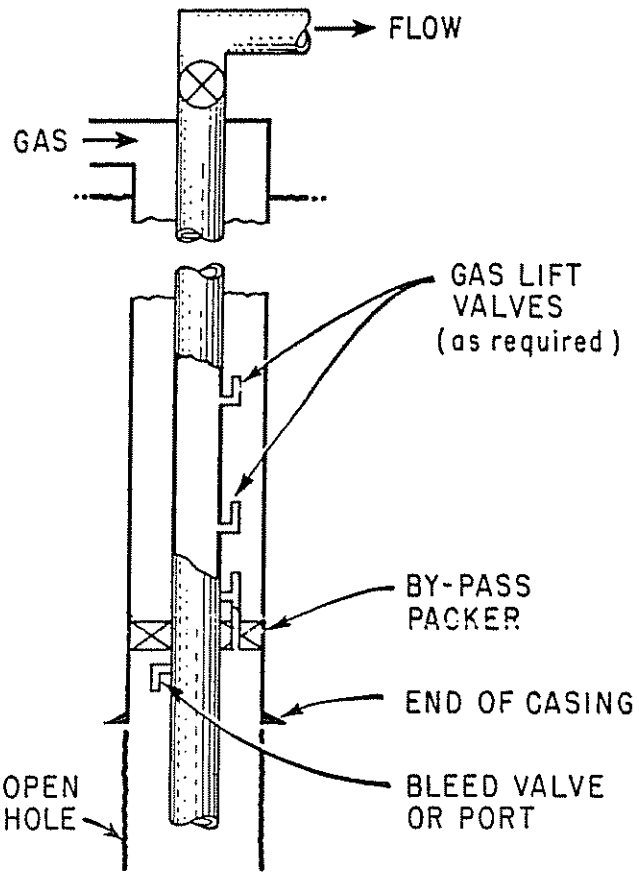


Fig. 3.310 Open Hole Chamber Installation

installation has been used successfully in Ecuador. The one disadvantage is the limitation on the size of valve, hence port size that can be placed at position "B." A very small port size here can also cause inefficient operations. However, by special arrangement the valve could be placed in an enlarged section of the tubing immediately on top of the chamber.

3.356 Special chamber installation for sand removal

A special chamber design (Fig. 3.39) is utilized for the production of sand-laden fluids. The standing valve is extended so that it is now located in the perforated tubing interval. When the chamber is emptied of fluids, the standing valve ball and seat are washed clean of sand. The perforations must be kept as close to the top of the packer element as possible. This may require a special packer design for extremely bad sand wells.

As fluids enter the chamber through the standing valve, the sand has a tendency to settle back. With the extended standing valve there is ample space for the sand to settle around the standing valve extension without plugging the standing valve. Sand will also settle back onto the top packer. However, once gas is injected, the washing action of the produced fluids through the perforated nipple will clean the sand from around the top of the packer. This installation has been successful where other installations have failed in the production of sand. In one case it operated very well where the well previously could not operate successfully on either standard intermittent lift or rod pump.

3.357 Open hole chamber installation

There have been a few instances where an operator prefers to run only one packer in a relatively small casing (taking away the disadvantage of an insert chamber). Open hole chambers have been run in some of these cases (Fig. 3.310). Its immediate disadvantage is subjecting the pay interval to the gas pressure and the subsequent pushing of fluids back into the formation. However, since some formations are relatively tight, it may be that this installation will exceed the production rate of a standard intermittent installation. It is not normally recommended.

3.358 Special chambers for bad casing and/or long perforated or long open hole interval

Figures 3.311 and 3.312 show two special chamber installations that can be run where bad casing exists or where the pay interval is very long in relation to total well depth. These installations have been successfully used in Talara, Peru, for low volume wells. Both installations of Figure 3.311 and Figure 3.312 do not permit gas injection pressure to be placed on the casing; in addition, the production gas can be vented to a very low pressure system in order to allow maximum fluid entry rate. Injection gas passes down the tubing-macaroni annular space in Figure 3.311 and brings the produc-

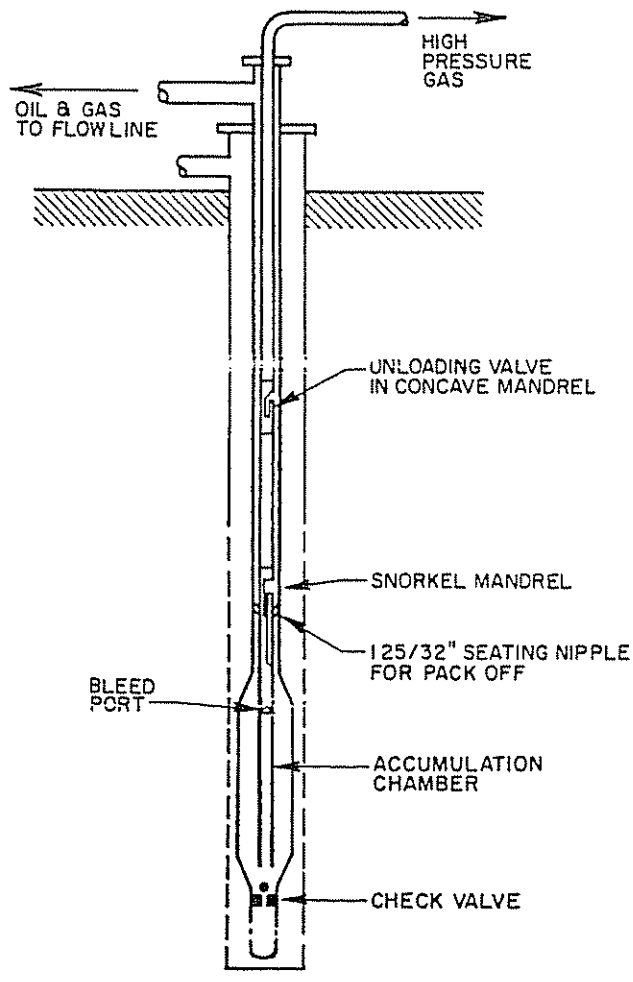


Fig. 3.311 Special Chamber for Macaroni Tubing Flow

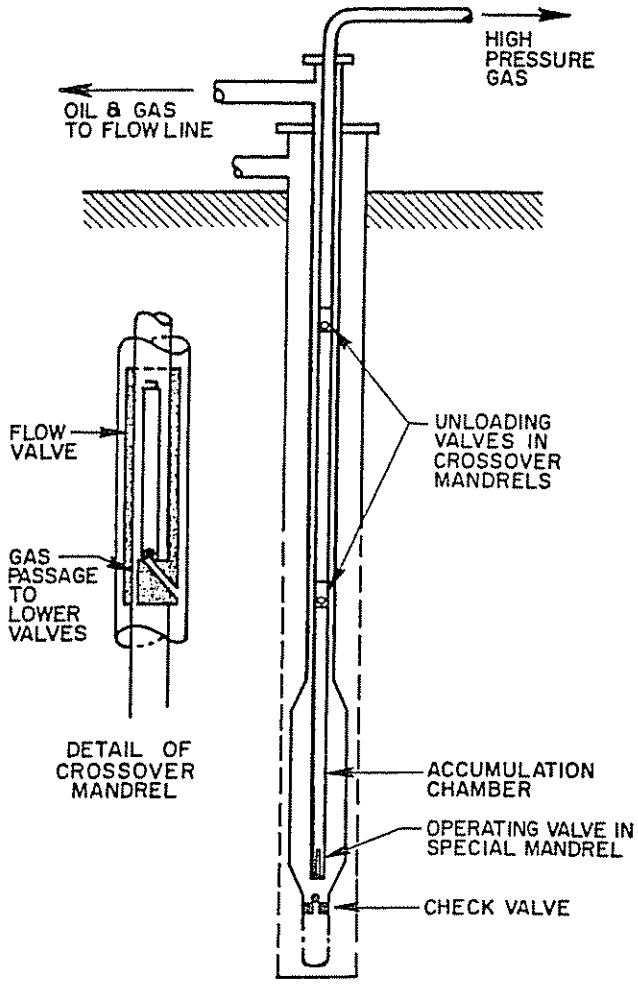


Fig. 3.312 Special Chamber for Macaroni Annulus Flow

tion up the macaroni string after U-tubing fluids from the chamber. In Figure 3.312 the gas enters the macaroni string and production comes up the tubing macaroni annular space.

3.359 Chamber above packer

Figure 3.313 shows a chamber installation with all the chamber located above the packer. The reason for running the chamber above may be because of sand or other expected difficulties. This type of installation functions well, and numerous similar installations have been installed. However, it does not run as deep as the insert chamber below the packer and it contains less volume per foot of length than the two-packer chamber.

3.3510 Automatic vent chamber system

DeMoss discussed the automatic vent chamber system which was developed to use the advantages of the conventional chamber and to eliminate the two basic disadvantages—no feed-in during lift periods, and the problem of dissipating formation gas.[2] The first problem was overcome by providing a method of venting the chamber of high pressure gas immediately after the transfer of liquids from the chamber into the tubing, allowing the chamber to start filling while the lift period is in progress. The second problem was overcome by allowing the formation gas to vent directly to the casing annulus instead of into the tubing.

Two types of automatic vent chamber (AVC) systems are available and have been used in the field. One system is gradient-operated by the well liquids in the chamber and the other system is pressure-operated with a time-cycle controller as the surface gas controls. The pressure-operated system is for wells with capacities up to 150 B/D and the gradient-operated system is for wells with higher capabilities.

Figure 3.315 shows a schematic of the gradient-operated AVC system, and Figure 3.314 shows a schematic of the pressure-operated AVC system.

The system consists of dual tubing strings, gas lift valves, and an automatic vent chamber. One tubing string is the production string and the other is high pressure lift gas. Large quantities of fluids are accumulated in the chamber with minimum back pressure on the formation. The chamber valve senses when the chamber is filled and automatically opens, communicating the high pressure gas from the lift gas line to the top of the chamber, expelling the fluids from the cham-

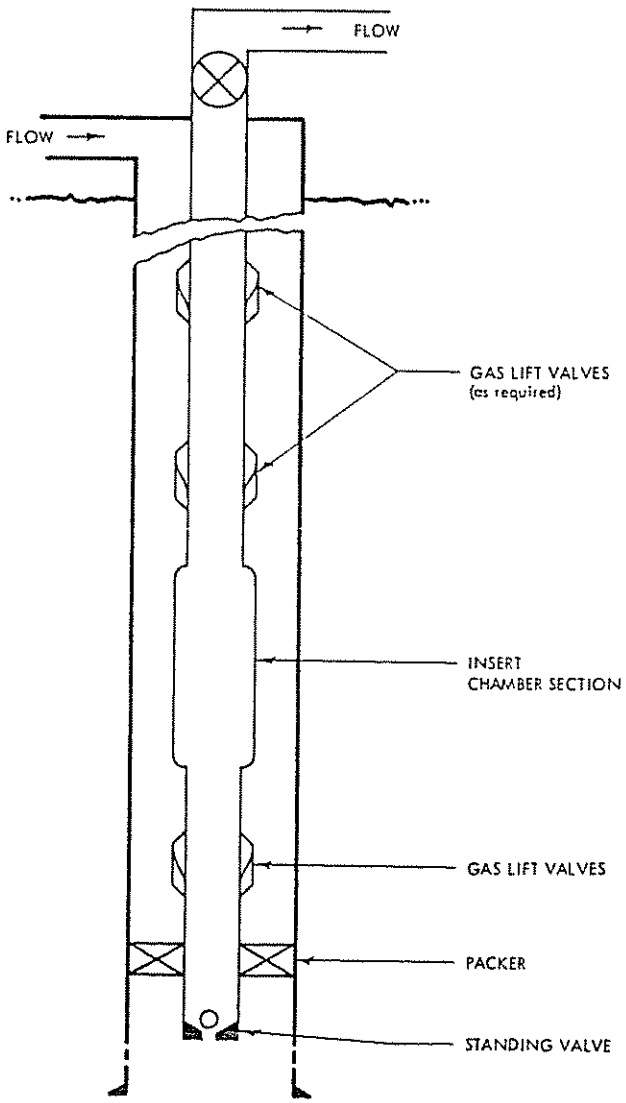


Fig. 3.313 Chamber Installation Above Packer

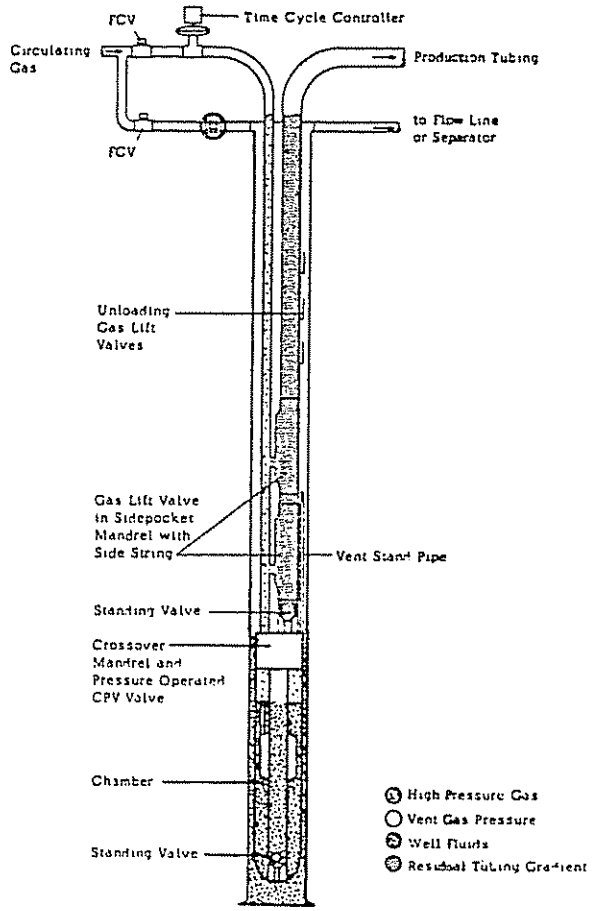


Fig. 3.314 Schematic of Pressure-Operated Automatic Vent Chamber

ber into the tubing. Once the chamber is empty of liquids, the chamber valve closes and vents the chamber to the low pressure annulus, allowing the chamber to refill while the liquids are being gas lifted to the surface.

Since no packers are used, the annulus is vented at the surface, resulting in minimum formation pressure and maximum production. Venting into the annulus also accommodates high gas-liquid ratio wells since the formation gas is free to pass up the large annular area.

Operation (Fig. 3.315)

Mechanical components of the AVC System are a special chamber pressure-vent (CPV) valve, quadruplex mandrel, accommodator, chamber, and gas lift system. Each of these elements plays an important roll in the operation of the overall system.

(1) *Chamber pressure-vent valve*—This valve is the heart of the system and is made up of a 3-way, 2-position main valve and a choice of two pilots for control of the cycle frequency. One pilot is controlled by the fluid level in the chamber. The alternate system is controlled from the surface, generally by a time cycle controller.

(2) *Quadruplex mandrel*—This mandrel receives the chamber pressure-vent valve and contains flow courses that connect the chamber to the production tubing, lift gas conductor, and vent pipe.

(3) *Accommodator*—The accommodator serves as a pressure relay. It senses the fluid weight in the dip tube and relays this signal to the CPV pilot for cycle control. The accommodator is not required for the surface cycle-control system.

(4) *Chamber*—Generally a bottle-type chamber is used, facilitating venting of the formation through the casing annulus to the surface. The chamber size is a function of the production desired, the available space, and the operating gas pressure. Chamber sizes may vary from as little as two barrels to more than twelve barrels.

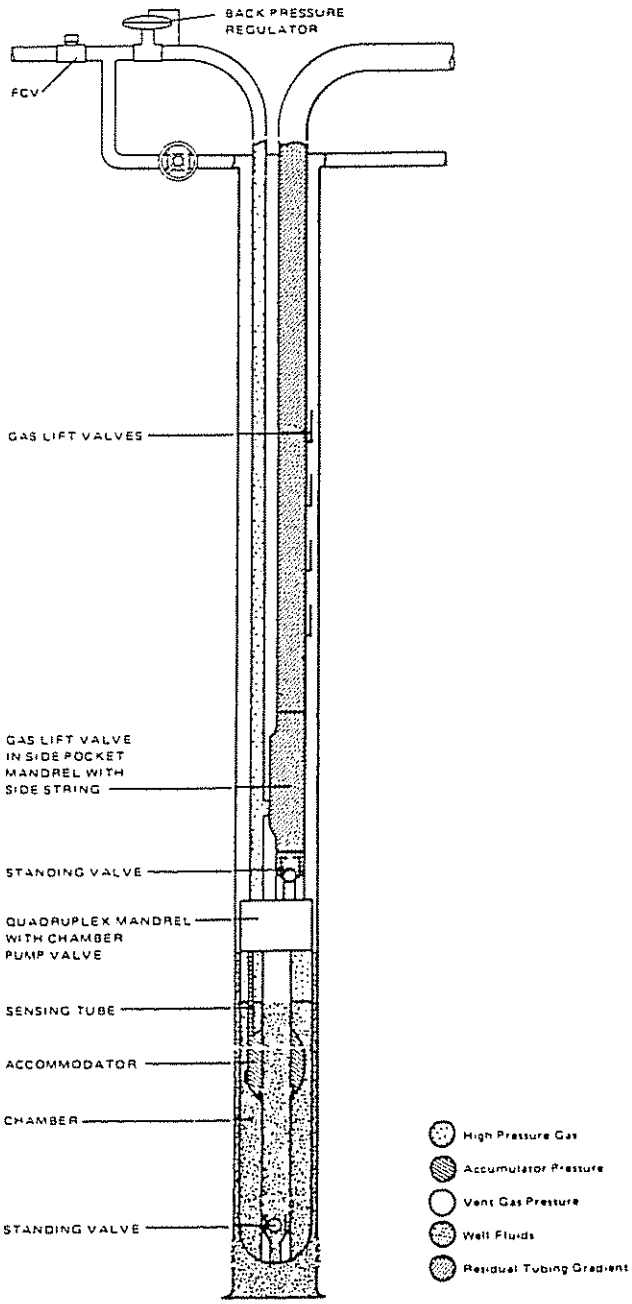


Fig. 3.315 Gradient-Operated Automatic Vent Chamber System

Sequence (Fig. 3.315)

Refer to Figures 3.315 and 3.316 for the following sequence of operation.

(1) The filled chamber signals the CPV valve to close the vent port and allow lift gas to enter the top of the chamber annulus.
(2) High-pressure lift gas forces the fluid from the chamber into the production tubing where it is retained by the standing valve.
(3) In this illustration the chamber is almost empty, thus completing the purge cycle. As the fluid leaves the chamber, the CPV closes and shifts to its power seat, closing off the high pressure gas inlet and opening the vent port to the annulus to relieve the high pressure gas from the chamber. With the fluid now in the tubing, it may be lifted to the surface using conventional gas lift means.
(4) In this view, the chamber is venting and refilling. The CPV valve will remain in the vent position until the chamber is refilled. The gas lift cycle is shown almost complete. The cycle frequency of the chamber can be independent of the lift frequency and is determined by the production capability of the well and the size of the chamber.

This gradient-operated system responds to increasing production capabilities of a well without adjusting the surface gas controls. The surface controls merely keep a defined pressure on the gas tubing. A production tubing pressure override stops gas input when the liquid slug arrives at the surface. As soon as the production tubing pressure declines, the circulating gas is again directed to the gas tubing (see Fig. 3.316).

This pressure record shows that one production tubing cycle is caused by one chamber cycle. The installation can be designed to give one tubing cycle with two or three chamber cycles. This means that a large liquid slug can be lifted in the production tubing, even when a small chamber is used. This system automatically increases the cycle frequency to match the well's production capability.

Compared to the gradient-operated AVC system, the pressure-operated AVC system must be manually adjusted to match the production capability of a well. Its sequence of operation is similar to the gradient-operated system. It uses a pressure-operated CPV valve and therefore does not need the accommodator and sensing tube. Figure 3.314 is a schematic of the pressure-operated system.

The surface gas controls are a time-cycle controller and choke. The timer is adjusted to cycle the chamber valve each time the chamber fills. The operating gas lift valve may be adjusted to function with one or more chamber cycles. Figure 3.317 shows a pressure record of this system. This record shows that two chamber cycles are pumped into the production tubing before the liquid is lifted to the surface.

In both of these systems, the chamber automatically vents after the CPV valve pumps the liquid up to the production tubing. This allows the chamber to be filling simultaneously with a lift period of the production tubing.

Several automatic installations have proven to be successful and are producing more than 1000 B/D in a few cases. They are particularly suited for low bottom hole pressure high productivity wells.

3.36 Macaroni Installations

Many ultra-slim hole completions are being used in the oil industry. Normally this means setting a string of 2⅜-3½ in. OD pipe for casing. Once this pipe has been set for casing, a smaller tubing (1-1½ in.) may be run for artificial lifting of the well. These smaller tubings are commonly referred to as "macaroni." The size of macaroni that can be run in an ultra-slim hole completion is limited to the maximum OD macaroni pipe size that can be run into the well and to the OD of the gas lift valve and mandrel. (In a paper by Thrash, the problems of gas lifting small diam completions are discussed.)[3]

Figure 3.318 shows a typical single-zone completion being lifted by running a string of macaroni tubing equipped with gas lift valves.

Figure 3.319 shows a concentric dual completion where macaroni tubing has been run to gas lift the lower zone only. The upper zone is still flowing. However, various crossover arrangements are available to switch either zone into the tubing string to be lifted. The strongest flowing zone can be put into the larger annular space.

Figure 3.320 shows a common installation where a string of macaroni tubing with gas lift valves has been run to lift one side of a parallel dual string. When the other side ceases to flow, it can be lifted in the same manner. The two macaroni strings allow a separate gas control to each string of valves, thereby eliminating the problems of lifting two strings from a common annulus gas supply. If the production capacity does not exceed the macaroni tubing sizes, this can be a more satisfactory method for completing parallel dual strings. Pulling one macaroni string can be performed without interfering with the other zone.

Figure 3.321 shows a concentric dual string where the lower zone can be lifted by running a string of macaroni pipe with gas lift valves.

Generally, it is more satisfactory to lift out through the tubing than to lift out through the annulus. The following reasons for this are given by Thrash:[3]

(1) For given production rates on continuous flow more gas must be injected into the annulus to set up a gradient whereby the necessary drawdown is achieved at the formation face.
(2) In intermittent type gas lift, the fluid head is consistently interrupted by the ever-changing annular cross-sectional area and shape as it ascends. Further, it is agitated by encountering coupling edges when coupled tubing is used.
(3) The ill effects of corrosion and erosion are not confined to the inner surface of the tubing. Instead, both the interior of the casing and the exterior of the tubing are subject to damage in annular flow.
(4) Paraffin deposition cannot be controlled or removed as effectively when lifting through the annulus.

What is the total fluid production possible for macaroni installations? These figures are given by Thrash.[3]

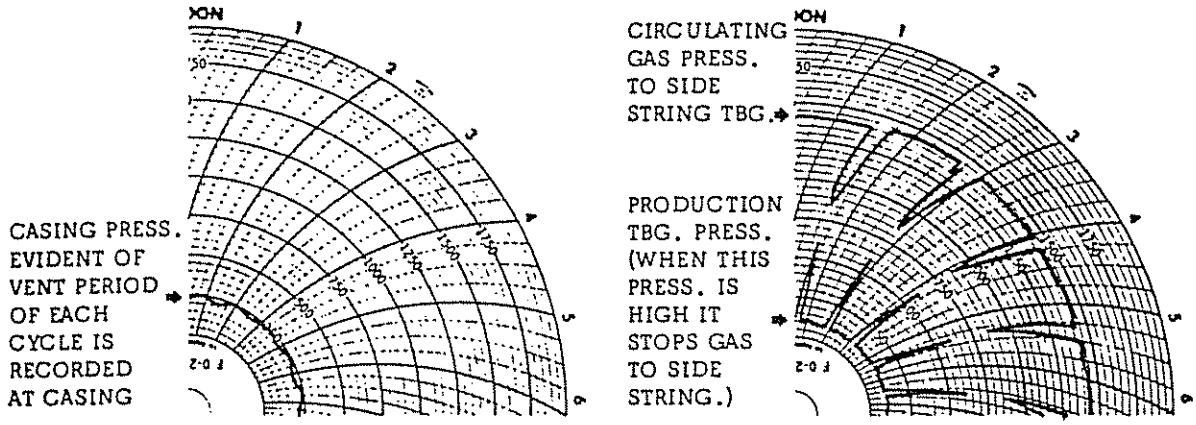


Fig. 3.316 Pressure Record of Gradient-Operated AVC System

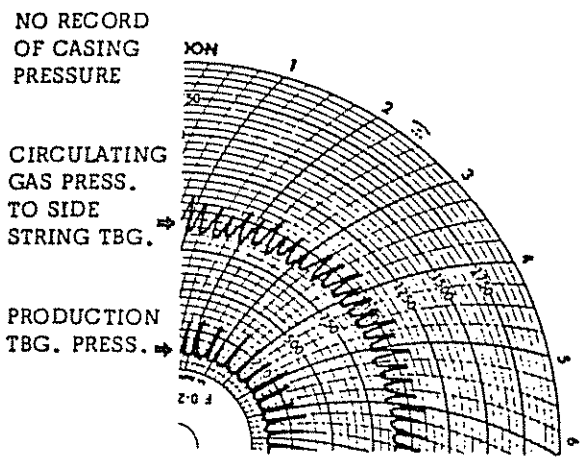


Fig. 3.317 Pressure Record of Pressure-Operated AVC System

Nominal tubing ID (in.)	Normal maximum gas lift gross production (B/D)
¾	150-200
1	250-300
1¼	450-500
1½	850-900

These volumes apply only to high fluid level continuous flow installations. Considerably smaller volumes should be lifted by intermittent lift.

3.37 Dual Installations

Dual installations are becoming increasingly necessary in modern-day lift problems. A dual installation is one in which two zones will be produced to the surface through two separate configurations. This is normally accomplished by two parallel strings of tubing (Fig. 3.322). Very few concentric dual installations are run except for the previously-mentioned macaroni installations where only one zone is to be lifted.

Various combinations of gas lift valves can be run on dual installations. Trouble-free dual installations are somewhat difficult to design. In many cases two zones

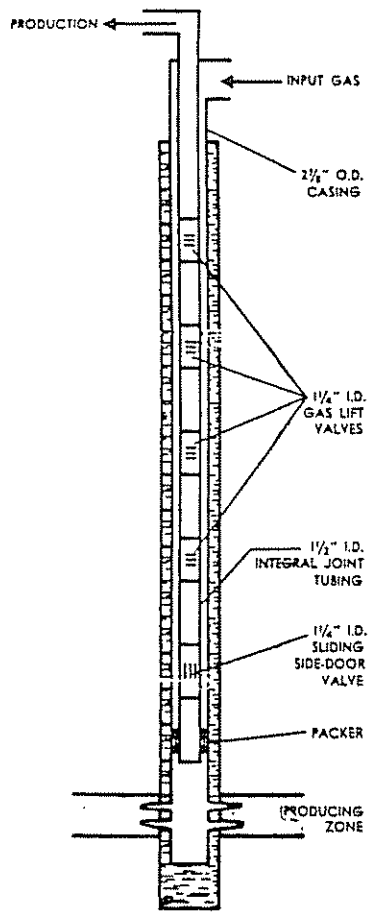


Fig. 3.318 Macaroni-Small Diameter Completion

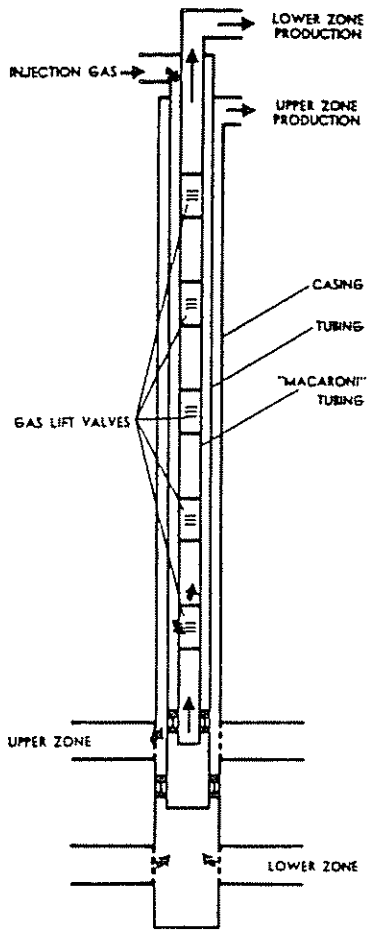


Fig. 3.319 Macaroni-Concentric Dual Completion

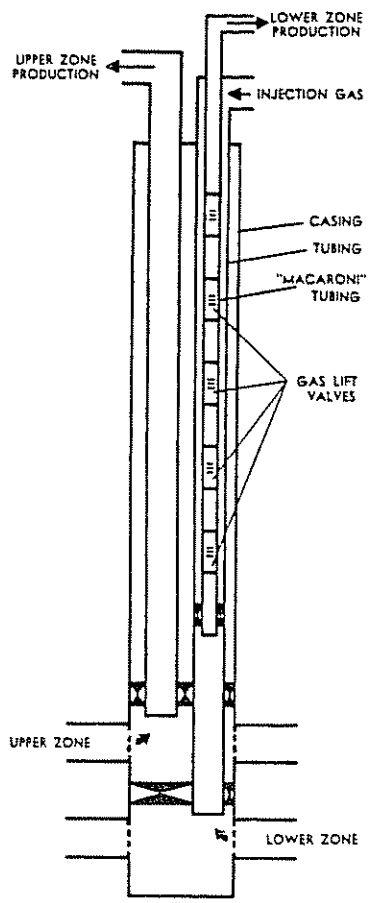


Fig. 3.320 Macaroni-Parallel String Completion

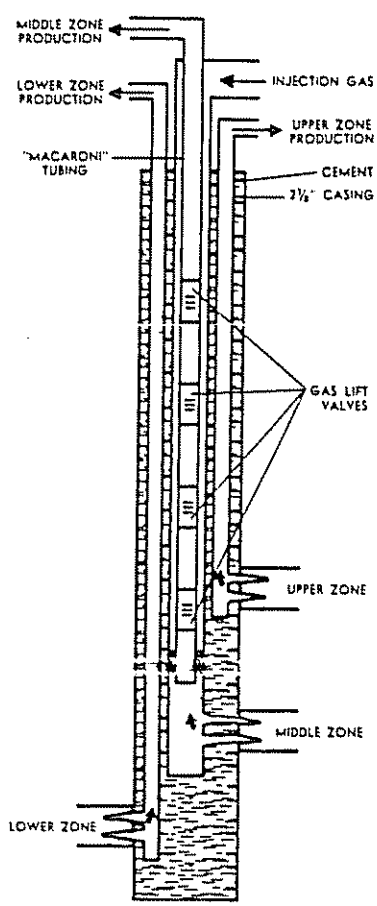


Fig. 3.321 Macaroni-Multiple String Small Diameter Completion

are wholly dissimilar. For example, one zone could be high volume continuous flow, whereas the other zone could be low volume intermittent lift. If the total depth is 10,000 feet, one zone could be lifting from 3,500 feet while the other zone would need to lift from bottom.

The ideal situation for a dual installation would be to have a string of gas lift valves on each tubing string that would operate independently of the other string. Generally, this means that one string of gas lift valves must operate independently of the injection gas pressure; of course, it is possible to use valves whereby both zones operate independently of the casing pressure (see section on design of dual installations).

3.38 Pack-off installations

The pack-off gas lift technique places a well on gas lift without disturbing the original tubing string initially completed for a flowing well and is discussed by Brown, Donaldson and Canalizo.[4]

This technique consists of installing a gas lift valve inside the tubing string between an upper and lower pack-off assembly. The two pack-offs are placed so they straddle a previously-perforated hole in the tubing. Gas from the annulus enters through the tubing, through the gas lift valve, and into the tubing string. Well fluids are produced up through the center of the pack-off assembly. Any number of these assemblies can be placed in a well similar to a regular gas lift installation.

Figure 3.323 shows a typical pack-off assembly. The lower collar stop is set with one~~-wire~~ ~~line trip~~, the tubing is perforated, and the entire upper assembly—including bottom pack-off, gas lift valve, upper pack-off, and upper tubing stop—is run and set with one-wire line trip.

The following installation procedure is a guide for installation of a typical pack-off assembly.

(1) Once the optimum location of the pack-off assemblies has been determined, the bottom collar stop is set in the nearest coupling to that point, of course setting the lowermost assembly first.
(2) A tubing perforator is then run to that point and rests on the bottom stop. This allows a hole to be mechanically punched through the tubing some 15 in. above the tubing stop. Normally, this hole is either a slot with dimensions of $^{3}/_{16}$ × ¾ in. (with cam perforator) or a circular hole of $^{5}/_{16}$ in. diam (with powder-charged perforator).
(3) If mud is in the annulus, it is circulated or gas lifted out through the perforated hole, although in some instances mud has been produced out through the gas lift valve itself. To prevent the mud from reaching the producing interval, a cir-

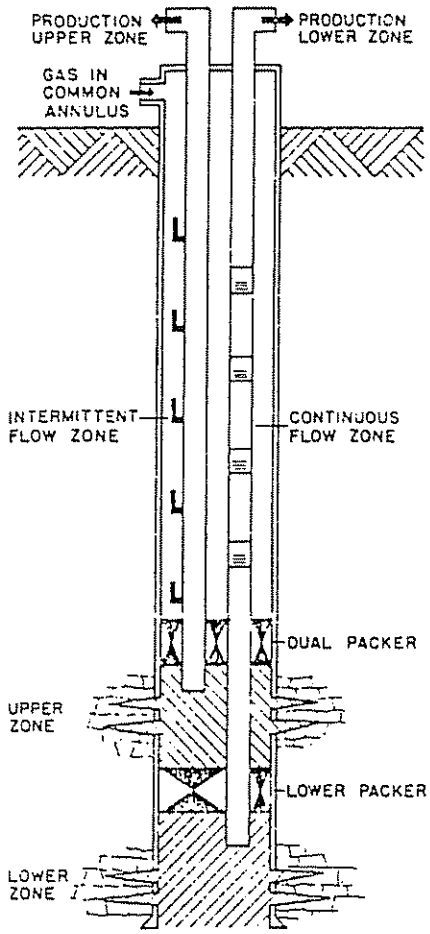


Fig. 3.322 Dual Installation With Parallel Tubing

culating plug may be placed on the lowermost stop. The tubing is then perforated and circulated. Some operators circulate a soap detergent to ensure good cleaning. The well should be circulated sufficiently so that a mud cake does not form in the annulus which might flake off and cause trouble later.

The well should be gas lifted for as long as 24 hours through the perforation prior to installing the valve to allow the normal flowing temp to expand the mud in the annulus, thereby producing it to the surface. This prevents the mud from plugging the gas lift valve.

(4) After the well has been sufficiently cleaned, the entire upper assembly, including lower pack-off, gas lift valve, upper pack-off and upper tubing stop, is run in and set on the lower collar stop as one unit (Fig. 3.323).
(5) Successive assemblies are run up the hole at predetermined spacing points.
(6) The well is then unloaded and placed on gas lift in the conventional manner. Caution should be taken to unload the well slowly, thereby minimizing damage to the valve from annular fluids.

Some additional applications for pack-off installations are to:

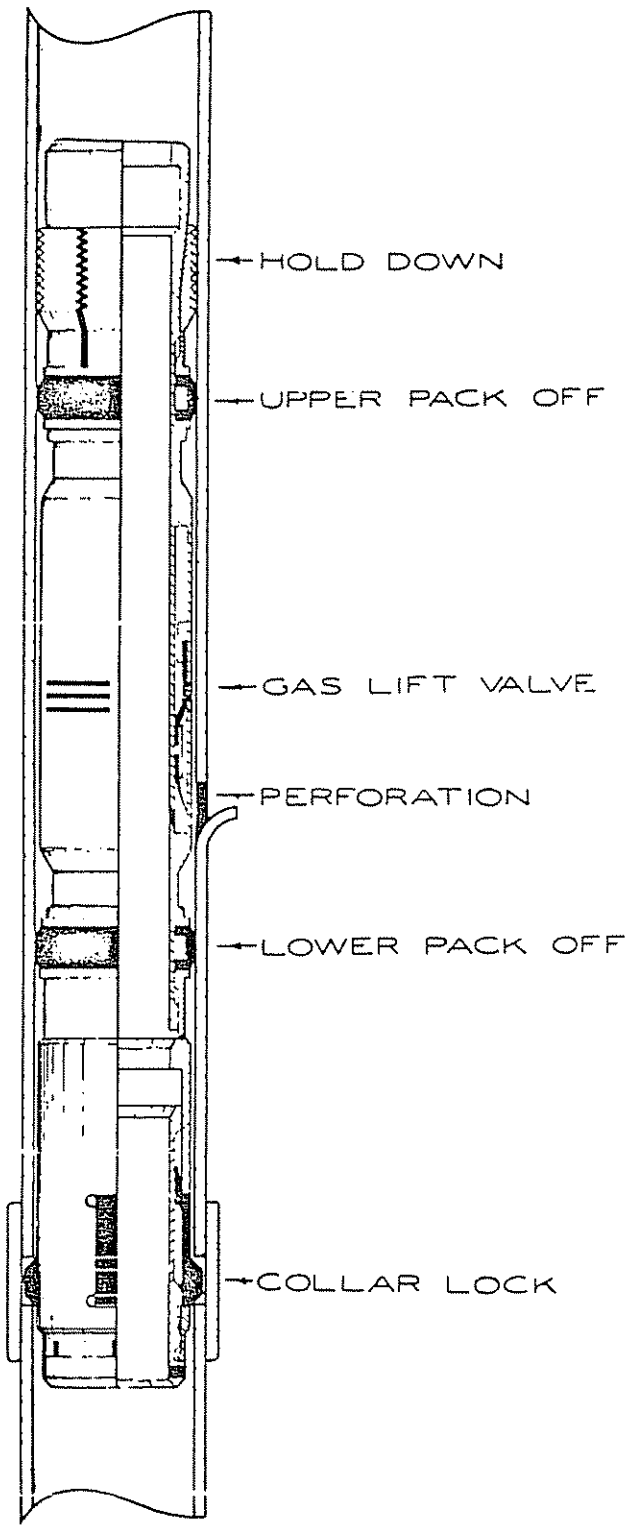


Fig. 3.323 Complete Pack-Off Gas Lift Assembly (after Brown, et al,[3] Courtesy Petroleum Engineer)

(1) unload water from a gas well
(2) pack-off a faulty gas lift valve (Fig. 3.324)
(3) repair a tubing leak (Fig. 3.325)
(4) repair blast pipe on a dual installation (Fig. 3.326)

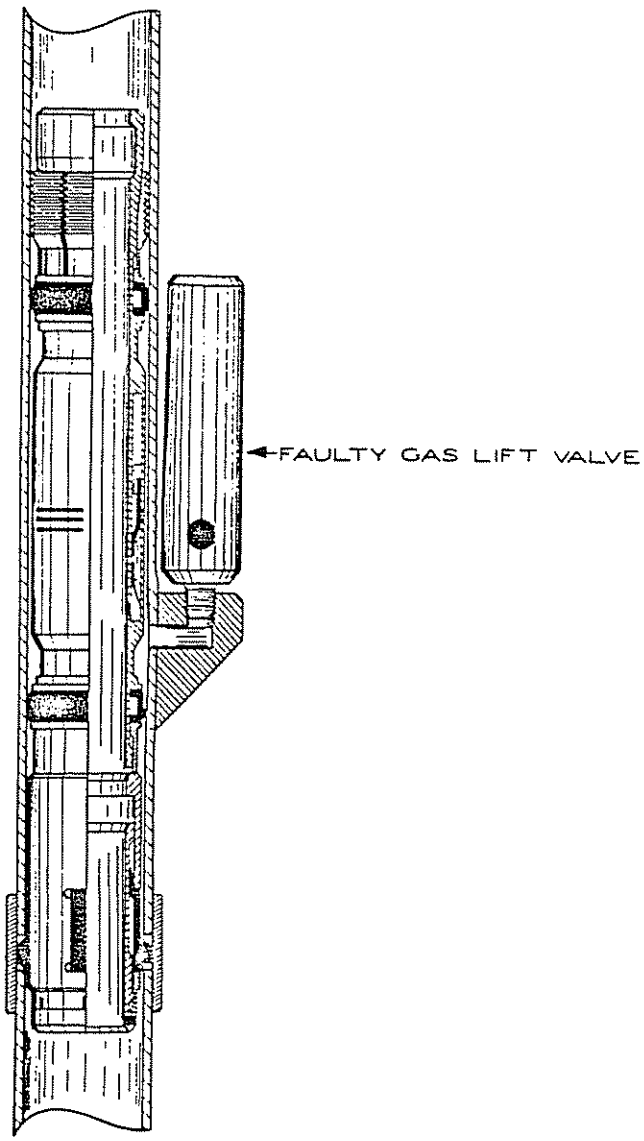


Fig. 3.324 *Packing-Off a Faulty Gas Lift Valve (after Brown, et al,[3] Courtesy Petroleum Engineer)*

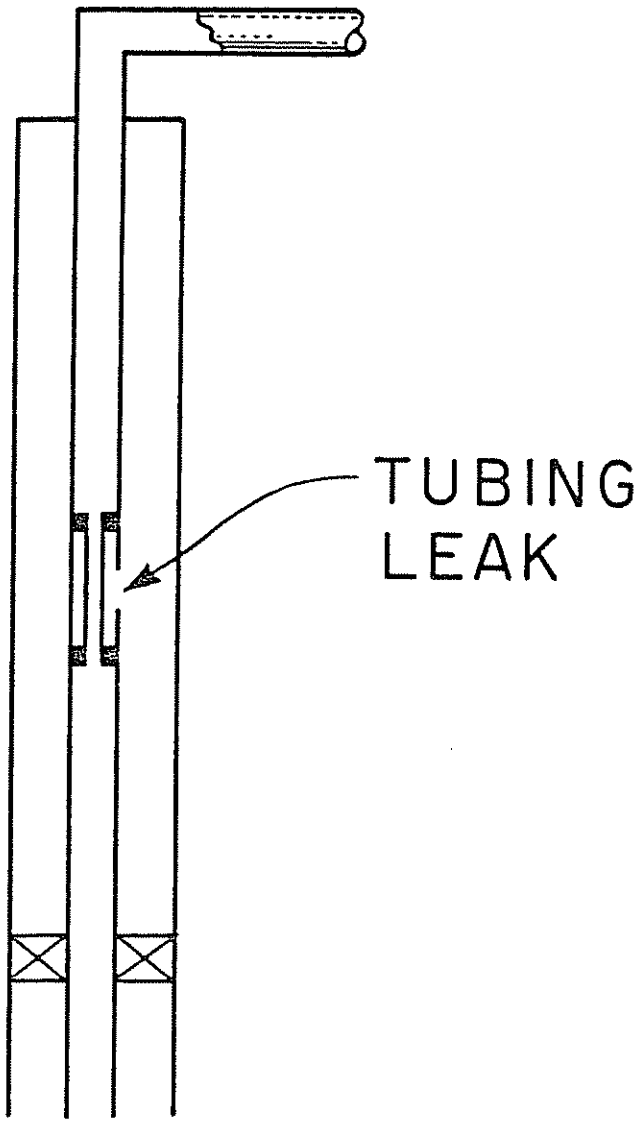


Fig. 3.325 *Repairing a Tubing Leak (after Brown, et al, [3] Courtesy Petroleum Engineer)*

(5) perforate an upper zone and place a well on production without pulling tubing
(6) extend depth of valves in original conventional installation
(7) extend the depth of gas lift valves on one zone of a dually completed well having two widely separated zones

3.39 Annular flow

An annular flow or casing flow installation injects gas down the tubing string and produces well fluids out the annulus (Figs. 3.327 and 3.328).

Annular flow is generally used when very large volumes of fluids must be moved, particularly when tubing capacity must be exceeded. It is best to stay with tubing flow if possible. There are some instances when it may be necessary to move anywhere from 5000-80,000 B/D of fluid. There are wells today that handle 50,000-60,000 B/D by using annular flow.

Many wells in the Middle East are on annular flow and may be producing up both the annulus and the tubing string simultaneously to obtain maximum flow rates. Wells offshore are generally restricted to tubing flow because of safety reasons and various regulations.

3.310 Installations to backwash injection wells

There are numerous wells used as disposal for salt water and as injection wells in water flood projects.

In either case, depending upon the amount of impurities in the water, the formation may become partially plugged so that the well no longer takes water at the original rate. A permanent gas lift installation offers an effective means to reverse the well flow—water is

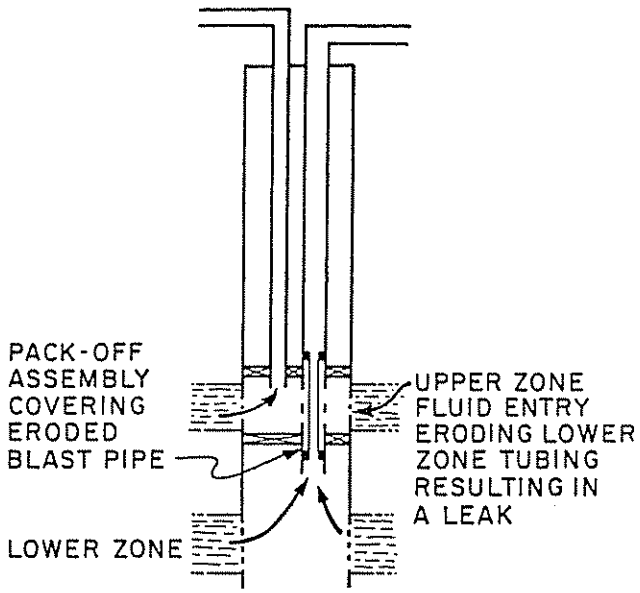


Fig. 3.326 Repairing Blast Pipe on a Dual Installation (after Brown, et al,[3] Courtesy Petroleum Engineer)

produced, not injected. This flow reversal removes any restrictions and cleans the casing perforations and the bottom of the well, allowing the formation to start taking water again.

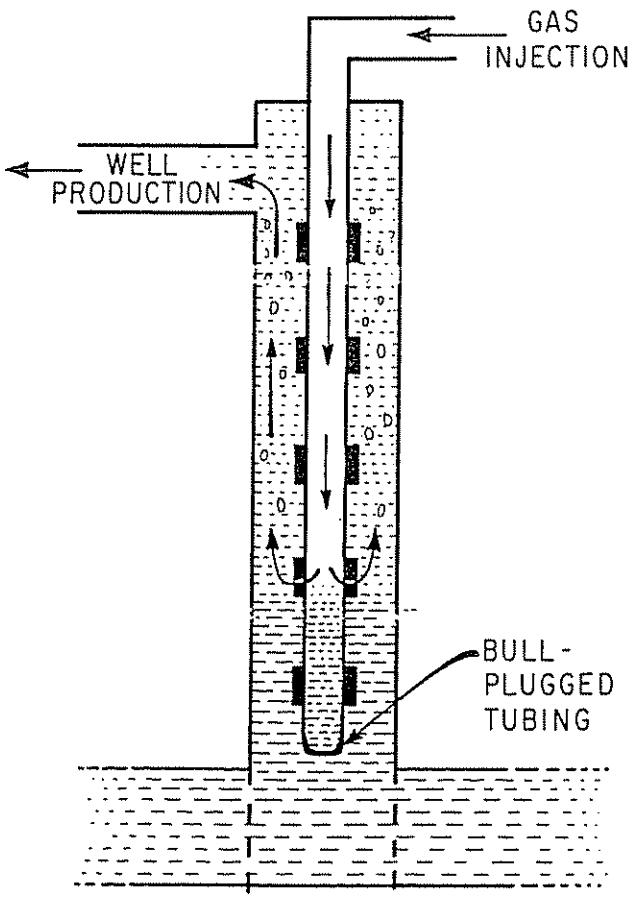


Fig. 3.327 Annular Flow Installation With Bullplugged Tubing

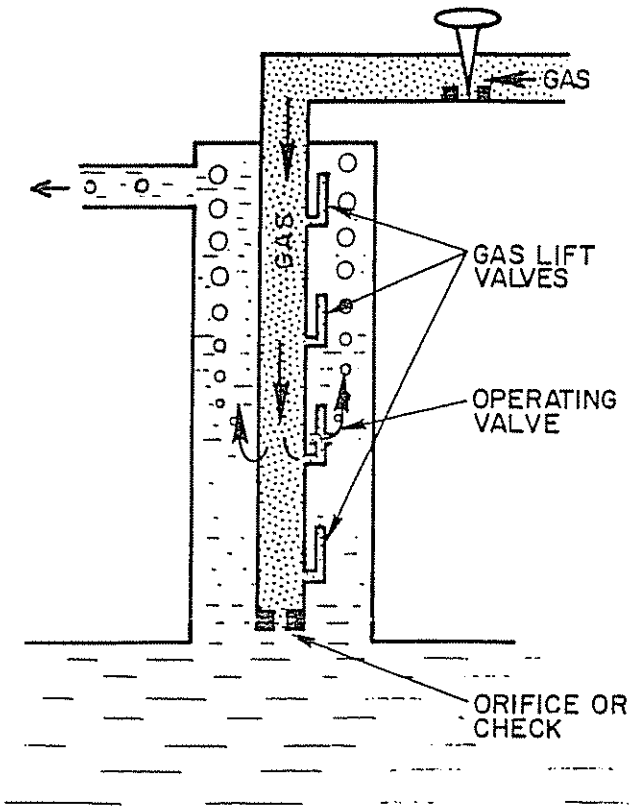


Fig. 3.328 Annular Flow Installation With Orifice on Bottom

An installation of this type must incorporate reverse checks so that water can be injected without cutting out the gas lift valves. Either tubing or annular flow may be utilized, depending upon the desired injection rate.

If several of these wells exist in one field and low-pressure gas is available, a portable compressor can be used to backwash each well in turn.

Although an injection well may take high volumes of water which produce high volumes of water in backwashing, volume is not the controlling factor in valve location. Gas lifting the well (backwashing) unplugs the formation so it can take the injection water at the desired rate. Therefore, the gas lift valves should be initially installed to a greater depth than for the normal gas lifting of the well. The deeper valves ensure a greater potential sand face pressure reduction than would be needed if the formation were not plugged. This allows the well to clean up much faster. Preferably, all valves should be of the continuous flow type; but if the available gas pressure and supply flow rate is limited, one or two intermittent valves are recommended for the bottom.

When backwashing begins, the fluid entry is retarded and the unloading progresses downward; as the fluid level is lowered and the sand face differential increases, the fluid entry rate will increase. The working fluid level will then move back up the hole and the increase in production will signal the success of the job. Wells are backwashed long enough to establish a constant output production rate

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2. DeMoss
3. Thrash, Purvis J., "What about Gas Lifting Tubingless Completions," *Petroleum Engineer,* (September 1960).
4. Brown, K. E., R. W. Donaldson, and C. R. Canalizo, "Pack-off Assembly Cuts Gas-Lift Installation Costs," *Petroleum Engineer,* (January 1960).

3.4 DESIGN OF GAS LIFT INSTALLATIONS

3.41 Introduction

Many factors must be considered in the design of a gas lift installation. One of the first is to determine whether the well should be placed on intermittent or continuous flow. Many installations must be designed without knowing whether the installation will do better on intermittent or continuous flow. Such borderline wells present a very difficult design problem. Gas lift valves are available that are suitable for both continuous and intermittent flow, although many valves can be used only for one or the other.

The purpose of gas lift valves is (1) to unload the fluids from the well so gas may be injected at the optimum point in the tubing string, and (2) to control the flow of injection gas under both unloading and operating conditions. As for all other types of artificial lift, gas lift creates the required flowing BHP so the well can produce the desired fluid rate.

The location of valves in the flow configuration is influenced by (a) available gas pressure for unloading, (b) fluid weight or gradient of the fluids in the well at time of unloading, (c) well inflow performance during the time of unloading, (d) surface back pressure at the well head against which the well fluids must be unloaded and produced, (e) fluid level in the casing and whether or not the well has been loaded with "kill" fluid, and (f) the BHP and well producing characteristics.

In most cases this information is available. However, in many instances the well's inflow performance is unknown. In this case a logical assumption based on the operator's knowledge of the well is made to arrive at a "design" production rate in B/D of liquid. The selection of such a production rate does not mean that the installation will actually produce the assumed or "design" quantity of fluid. It merely serves as a criterion to ensure that the valve installation will unload the well to the optimum point of operation while the well is feeding in fluids at the "design" rate.

Gas lift installations are flexible enough to adjust themselves to produce practically any rate. If a high design rate is used, it will probably mean that one or two extra valves have been used; but the installation should still perform satisfactorily for lower production rates. A logical "design" rate ensures that the number of valves used and their location in the tubing string is consistent with existing well conditions.

In a gas lift installation, all valve locations must be known so that dome pressure settings can be made in the shop that will give the desired downhole performance.

This chapter will cover design for both continuous and intermittent flow wells. In addition to single completions, design procedures for macaroni, dual, chamber, and pack-off installations will be covered.

3.42 Continuous flow design

3.421 Introduction

Figure 3.41(A) shows a typical continuous flow installation. There are six valves in the well. The top four valves are used as unloading valves to reach the fifth valve, which is the operating valve. One additional valve below the point of injection is added for safety and/or changing well conditions. Since one valve is below the operating valve, the casing kill fluid level seeks a point of balance between casing pressure and pressure in the tubing. Without this valve in the well, the liquid level in the casing would remain at the depth of valve 5 (operating valve). The four unloading valves remain inoperative until it is necessary to unload the well again for some reason, such as after "shut-in." If the reverse checks are functioning properly, the liquid level should not change in the casing after reaching stabilized conditions.

Fig. 3.41(B) is a continuous flow diagram illustrating the gradients and pressures involved in gas lifting a well. In this illustration depth is plotted vs. pressure and the pressure traverse of the fluids starts from the flowing BHP at the bottom of the well and moves all the way to the surface. Assuming an average flowing gradient beneath the point of gas injection and an average flowing gradient above the point of gas injection; start with the tubing pressure and calculate the flowing BHP as follows:

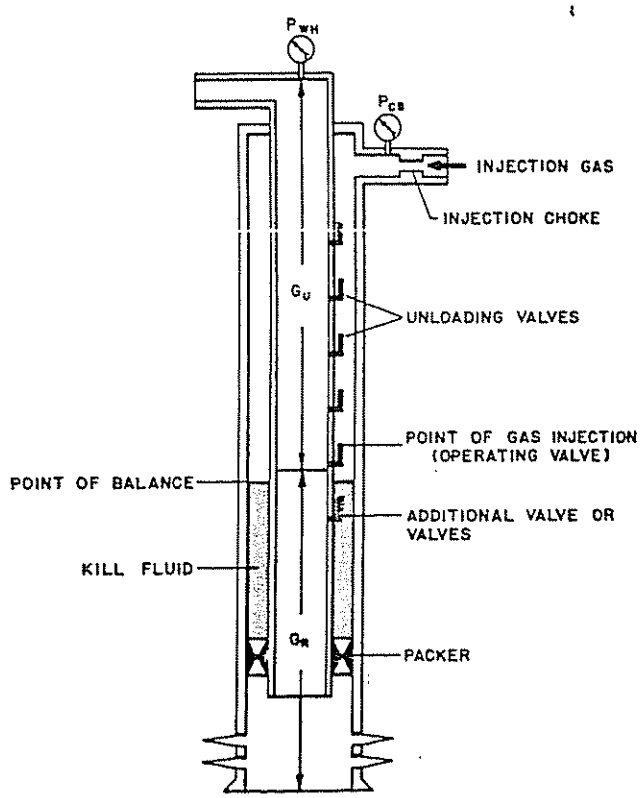


Fig. 3.41-A Continuous Flow Operation

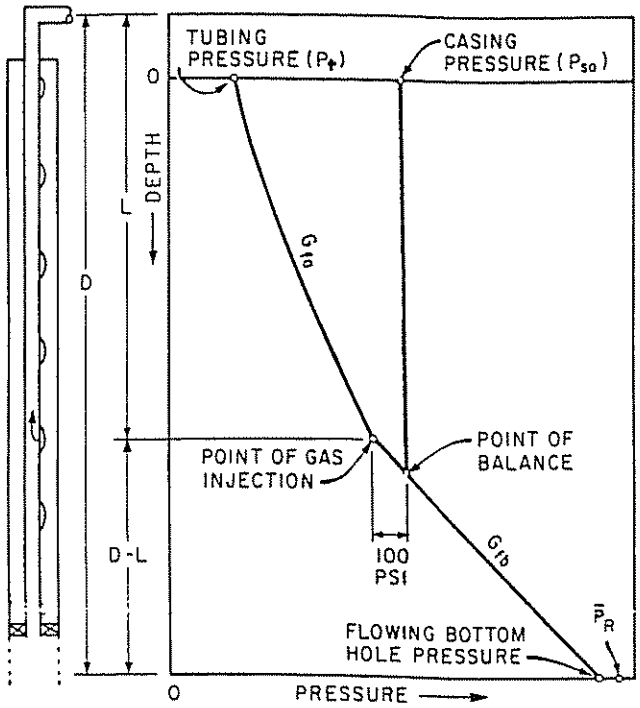


Fig. 3.41-B Continuous Flow Well Illustration

$$P_{wh} + G_{fa}L + G_{fb}(D - L) = P_{wf} \qquad (3.41)$$

where:

P_{wh} = wellhead pressure
G_{fa} = average flowing gradient above the point of injection
L = depth to point of gas injection
G_{fb} = average flowing gradient below the point of gas injection
D = total depth of well
P_{wf} = flowing BHP

This is the basis for continuous flow design. Continuous flow creates the necessary flowing BHP to allow the well to produce at a particular flow rate. Gas is injected at such a point that this pressure may be attained. Factors such as available pressure, available gas volume, flow configuration sizes, and flowing tubing pressure at the surface will influence production.

3.422 Factors to consider in the design of a continuous flow gas lift installation

3.4221 Requirements of continuous flow valves

Section 3.2 contains an explanation of continuous flow valves.

A continuous flow valve has operating characteristics different from an intermittent flow valve, particularly a throttling characteristic. Gas is regulated from the surface, and the valve opens to pass just this amount of gas. If more or less gas is needed, the valve should respond to a surface adjustment or to a change in tubing pressure.

For a bellows unbalanced valve to serve as a continuous flow valve, it must also have a throttling characteristic. The normal intermittent bellows valves or pilot bellows valves do not have this feature. Unless the differential pressure across these valves is extremely low, a valve of this type may intermit. Therefore, they are sometimes used in continuous flow wells by choking the valve to such a degree that it will not intermit. This is a ticklish design problem and the choke sizing becomes critical. If the choke size is too large, the valve will intermit; too small, and the installation will be robbed of gas or an upper valve will open, causing less efficient operations.

To make a standard intermittent valve throttle, choke the valve upstream from the seat, taper the stem, taper the seat, or achieve a variable orifice operation in some way. The tapered stem makes the valve sensitive to tubing pressure by producing a throttling effect which prevents full casing pressure from building up underneath the valve stem. The taper will then move in and out of the seat area as a function of tubing pressure, making a good continuous flow operation.

The port size makes a difference in the tubing sensitivity. If the port size is small (¼-in. diameter or less), a balanced condition is approached and the valve is less sensitive to tubing pressure. If the port size is large (⅜-in. diameter or greater), the valve is more sensitive to tubing pressure, but may have less tendency to throttle, even with a tapered stem. However, recent tests show the larger ported valves have better throttling characteristics. The larger ported valves have a greater safety factor in opening pressure in the operating condition because of the greater tubing pressure required to open the valve. This is particularly true of the upper unloading valves. Recent tests on most valves indicate they have a tendency to throttle, passing more gas for higher tubing pressures and less gas at lower tubing pressures. Both pressure operated and fluid operated valves are successfully used for continuous flow operations. (Refer to Section 2 for detailed discussion on valves.)

3.4222 Separator pressure and wellhead flowing pressure

Surface back pressure is an important design factor in continuous flow lift. Knowing whether or not the well must be unloaded into the system pressure or into a lower pressure is important in locating the first valve. For example, with 800 psi and a 0.50 psi/ft "kill" fluid, a liquid column of 1600 feet can be balanced. However, if the tubing pressure is 100 psi, only 1400 feet of liquid can be balanced. This is important in design, and the first valve should be as deep as practical to allow for closer spacing down the hole.

Horizontal flow correlations should be used to obtain the necessary pressure to produce certain fluid rates from the wellhead to the separator. Later examples will show this in detail.

3.4223 Location of the top valve

The location of the top valve is extremely important in a gas lift installation. In practically all installations the top valves are used for unloading purposes only. The location of valves in the operating region becomes even more important.

The top valve must be placed high enough so the well can be unloaded, yet deep enough for efficiency under operating conditions. (It may, in fact, turn out to be the operating valve). If a well is loaded with a 0.50 psi/ft "kill" fluid all the way to the surface and 800 psi is available to unload with, then the top valve may be placed at 800/0.50 = 1600 ft, neglecting gas column weight and assuming zero wellhead pressure. However, if valves are placed in a well where the fluid level starts at 3000 feet, it is safe to place this valve at 3000 feet or slightly lower. Placing this valve any lower than the fluid level is not recommended because the fluid level displaced by the tubing and packer will rise. The advantage of placing this valve at 3000 feet is twofold: closer spacing is possible, bracketing the point of gas injection, and fewer valves are needed.

3.4224 Injection gas pressure and volume

Two factors that control the production rate to a great extent are injection gas pressure and injection gas volume. As greater gas pressure is available, the point of injection can be progressively lowered, causing a lightening of the flowing gradient from a deeper and deeper point. Therefore, as the pressure is increased, the required injection gas volume can be reduced.

If an unlimited supply of gas is available, the minimum gradient can be attained in all installations for any gas pressure; therefore, maximum drawdown as well as maximum production can be reached.

For many continuous flow wells, higher production rates are impossible to obtain without increasing the surface operating pressure. A good approximation is to remember that a well may be placed on continuous flow to the following depth for most tubing sizes:

$$\text{Depth} = \frac{P_{so} - P_{wh}}{0.15} \qquad (3.42)$$

Example:

$$P_{so} = 900 \text{ psi}$$
$$P_{wh} = 100 \text{ psi}$$
$$\text{Depth} = \frac{900 - 100}{0.15} = \frac{800}{0.15} = 5333 \text{ ft}$$

For tubing sizes greater than 4" and for annular flow, this rule of thumb is not likely to hold true because an increase in flowing gradient is likely.

3.4225 Bottom hole temperature (BHT) and flowing temperature gradient

3.42251 Introduction

An important factor in the design of any gas lift installation is well temperature. Since practically all valves are subjected to increased operating pressures due to increased temperature, it is impossible to design a correct installation without this information unless, of course, a spring charged valve is used.

For the unloading process one set of temperature conditions exists, and for the installation in operation another set of temperature conditions exists. The same BHT (bottom hole temperature) exists in both cases. Figure 3.42 shows a correlation of surface temperatures recorded for a producing well with a temp of 176°F at 6000 ft. For a rate of 936 B/D and for a GLR of less than 300 scf/B, a temperature of 147°F was recorded at the surface. As noted, the flowing surface temp is a function of the liquid production rate and of the GLR (gas-liquid ratio).

There has been insufficient information published on the prediction of temperature traverses in tubing strings. A rigorous solution was given by Ramey and has been incorporated in some multiphase flow correlations.[1] A serious handicap to this solution is the requirement of correct heat transfer coefficients for casing, tubing, annular space with various fluids, and the surrounding formations. Most designers of gas lift installations have assumed straight line relationships between the bottom hole and surface flowing temperatures. This procedure has worked fairly well if the correct surface flowing temperature is known. However, a distinct curvature exists between top and bottom temperatures for higher flow rate wells, as noted in Figure 3.43. The assumption of a linear relationship can be off as much as 10-30°F in some cases. Also, this curvature makes it somewhat difficult to predict the exact traverse for temperature. Kirkpatrick offered one of the first practical solutions to the problem.[2] For wells that

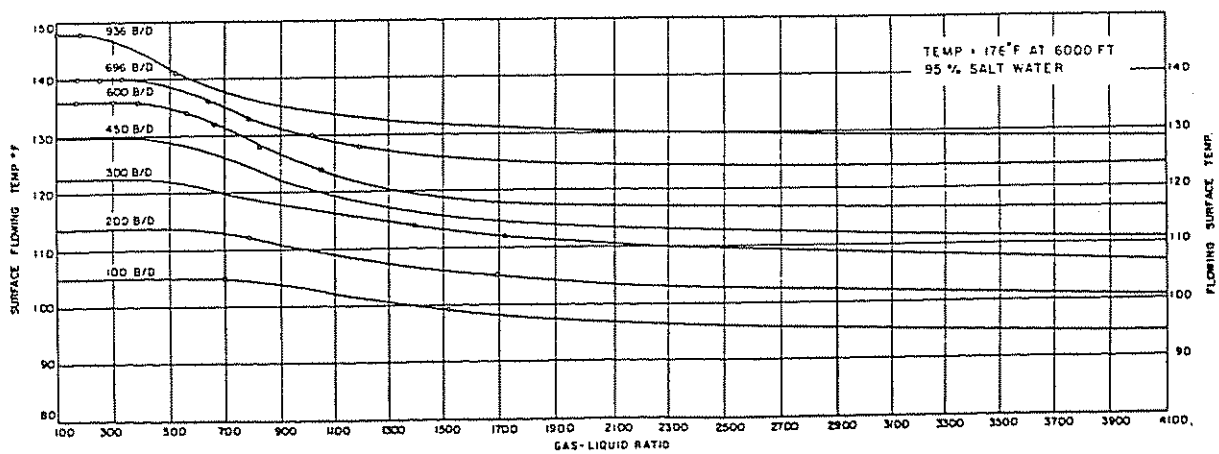


Fig. 3.42 Correlation for Surface Flowing Temperature (after Fancher and Brown,[12] Courtesy SPE)

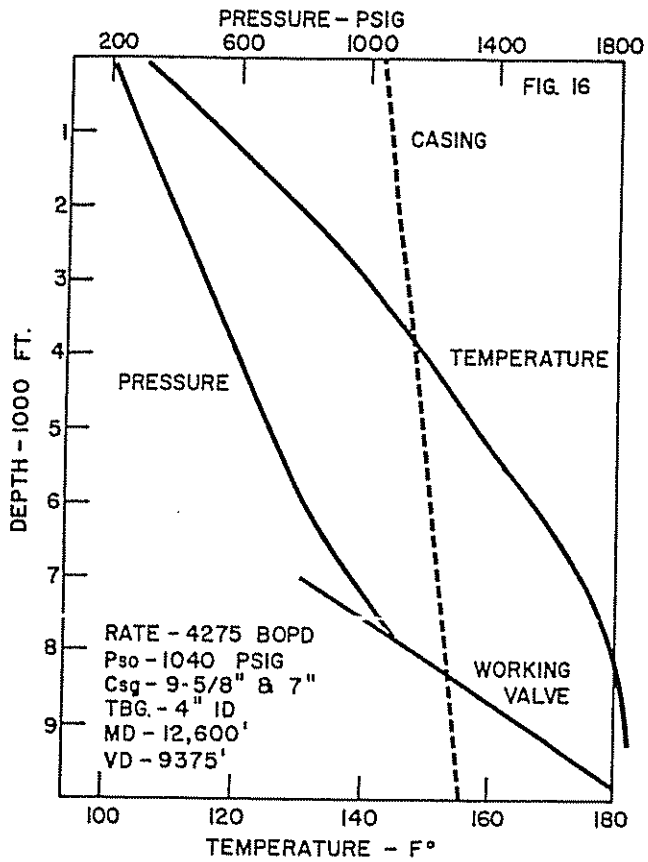


Fig. 3.43 Flowing Pressure & Temperature Survey

fall in the range of his correlation, the results are reasonable.

A thesis published at the University of Tulsa on this subject is presented in the following discussions.

3.42252 Kirkpatrick's solution[2]

Kirkpatrick prepared a correlation for predicting temperature depending upon the geothermal gradient and total liquid flow rate (refer to Fig. 3.44). His correlation is valid only for tubing sizes of 2", 2½", and 3" diameters. He also shows a limiting rate of 2000 B/D. By knowing the bottom hole temperature, tubing size, and geothermal gradient, we can obtain a flowing temperature gradient from which we can find the surface flowing temperature. His solution assumes a straight line between surface and bottom hole temperature. An example is given in Figure 3.44.

3.42253 Shiu's correlation

Shiu prepared a study from 370 well cases collected from three different geographic areas.[3] The first group contains 219 directional well cases which were collected from the Gulf of Mexico. The second group contains 41 vertical well cases which were collected from Cook Inlet, Alaska. A group of 110 vertical well cases were collected from Lake Maracaibo, Venezuela. The information in the first group was taken from the well data bank at the University of Tulsa, while those in the second and third groups were originally recorded in the temperature survey charts for each well case. The variable ranges of the data are summarized in Table 3.41.

TABLE 3.41

Variable	Average value	Minimum value	Maximum value
Tubing flowing pressure (Psig)	210.2	50.0	444.0
Tubing diameter (inches)	3.4	2.0	4.0
Liquid density (lb_m/ft^3)	55.4	53.0	62.4
Oil gravity (°API)	28.3	15.0	35.4
Gas gravity (air = 1)	0.76	0.70	0.82
Geothermal gradient (°F./100 ft)	1.10	0.80	1.11
Surface flowing temp. (°F.)	120.9	70.0	164.0
Bottom hole temp. (°F.)	183.9	130.0	230.0

The basis of the study was the use of the general energy equation which was applied by Ramey in his approach to the wellbore heat transmission problem.[1] The general expression of the energy conservation law states that the total energy change in the system is the sum of the change in kinetic energy, potential energy, enthalpy, and heat transfer due to flow work. Figure 3.45 shows a schematic diagram of the wellbore heat problem in a flowing well. Shiu started with the following equation:

$$dH + \frac{gdz}{g_c J} + \frac{udu}{g_c J} = dQ - \frac{dWf}{J} \qquad (3.43)$$

He arrived at his final equation in the following form:

$$T_{surf} = T_{BH} - g_T z + g_T A - g_T A C^{(-z/A)} \qquad (3.44)$$

The temperature T is a nonlinear function of Z and A if the bottom-hole temperature and geothermal gradient are constant values. Shiu used a modified stepwise multiple linear regression technique which fit an equation of the following form:

$$Y = C_o + C_1X_1 + C_2X_2 + C_3X_3 + \cdots C_nX_n \qquad (3.45)$$

He arrived at an equation for determining A which is:

$$A = e^{-5.065}\, W_l^{0.4882}\, d^{-0.3476}\, V_o^{0.2519}\, V_g^{7.724}\, P_L^{2.915}\, P_{lf}^{0.2219} \qquad (3.46)$$

The plot of the actual relaxation distance vs. the relaxation distance calculated by Equation 3.46 for all the test well cases is shown in Figure 3.46. The following summary tabulation shows comparisons to Kirkpatrick's method.

SUMMARY OF THE COMPARISON OF RESULTS

Statistical Parameter	Shiu's Method	Kirkpatrick's Method
average percent error	−0.733	−1.826
standard deviation	8.259	11.314

Chart To Be Used Directly For 2 1/2" Tubing.
For 2" Tubing Multiply The Actual Flow Rate By 2.
For 3" Tubing Divide The Actual Flow Rate By 1.5.

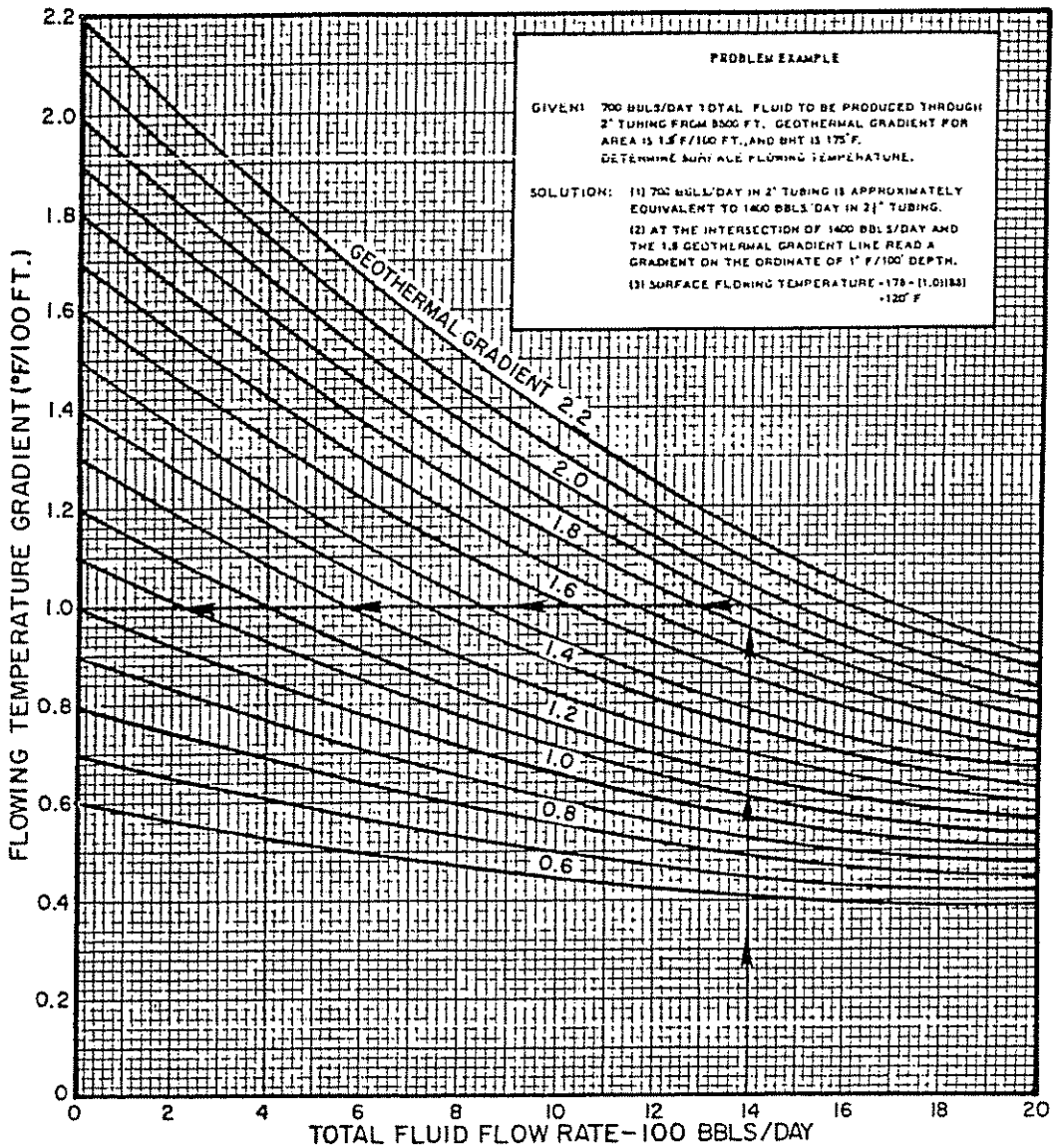


Fig. 3.44 Flowing-Temperature Gradient for Different Flow Rates, Geothermal Gradients, and Tubing Sizes

Shiu's method extends beyond the range of the work of Kirkpatrick in that it is good for other tubing sizes and all rates. Also, the temperature profiles predicted by Kirkpatrick's flowing temperature gradient chart are straight lines, while those predicted by Shiu are curves similar to the temperature profiles generated by the field data. Several comparisons were made by Shiu and typical plots are noted in Figures 3.47(A) and 3.47(B). This represents one of the most accurate and practical means of determining temperature traverses in a well.

EXAMPLE PROBLEM BY SHIU'S METHOD

Shiu gave the following example to illustrate procedures for calculating a surface flowing temperature using the Kirkpatrick method and his own.

Given:

$q_o = 2218.85$ STB/day
$q_w = 11.15$ STB/day
$q_g = 1761.17$ Mscf/day
$d = 3.0$ inches

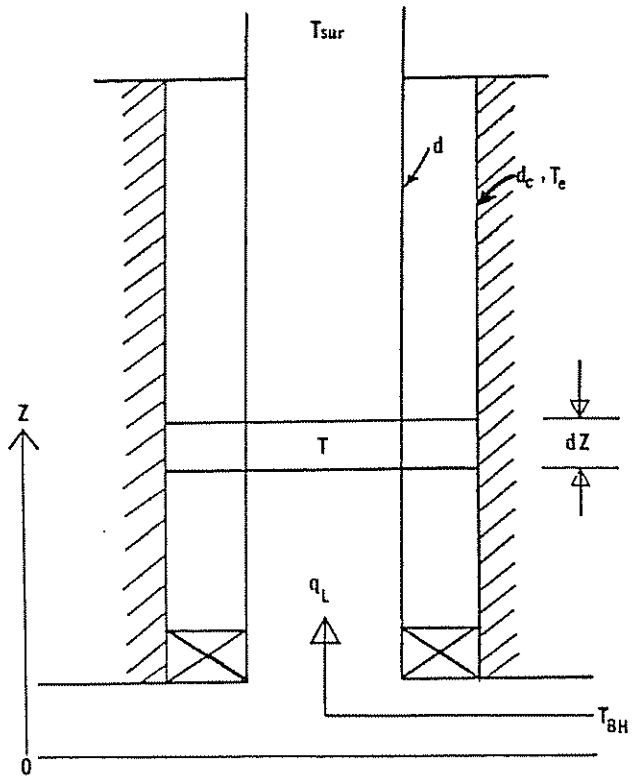


Fig. 3.45 Schematic Diagram of Wellbore Heat Problem in a Flowing Well (after Shiu)

$\gamma_o = 24.0$ °API
$\gamma_g = 0.7$
$\gamma_w = 1.10$
$g_T = 1.0625$ °F/100 ft
$Z = 5985$ ft
$T_{sur} = 134$ °F.
$T_{BH} = 173$ °F.
$P_{tf} = 150$ psig (flowing well-head pressure)

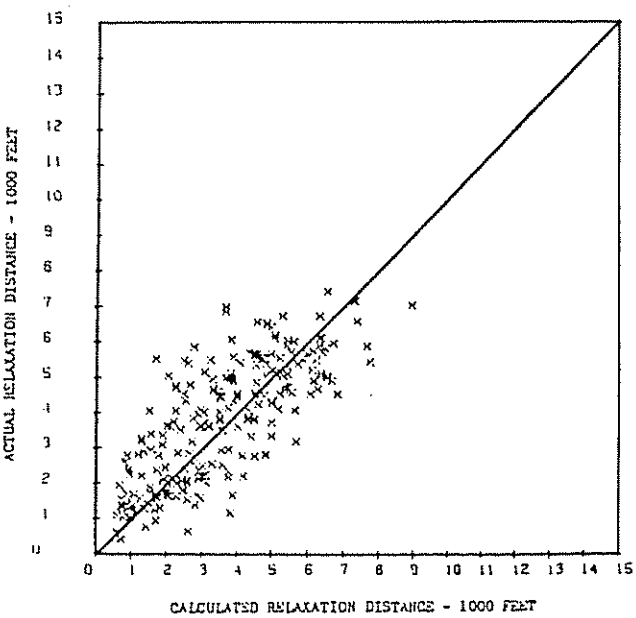


Fig. 3.46 Plot of Calculated Relaxation Distance vs. Actual Relaxation Distance (after Shiu)

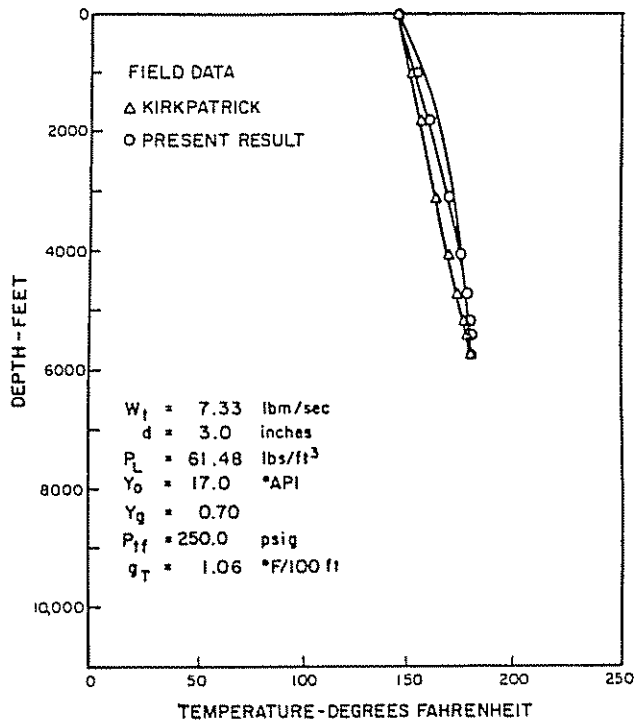


Fig. 3.47-A Temperature Profiles for Median Oil API Gravity (after Shiu)

Find the surface flowing temperature by using Kirkpatrick's temperature gradient chart and applying the method developed by Shiu.

Solution I. Applying Kirkpatrick's temperature gradient chart:

(1) Calculate total liquid flow rate, $q_L = q_o + q_w = 2218.85 + 11.15 = 2230.0$ STB/d

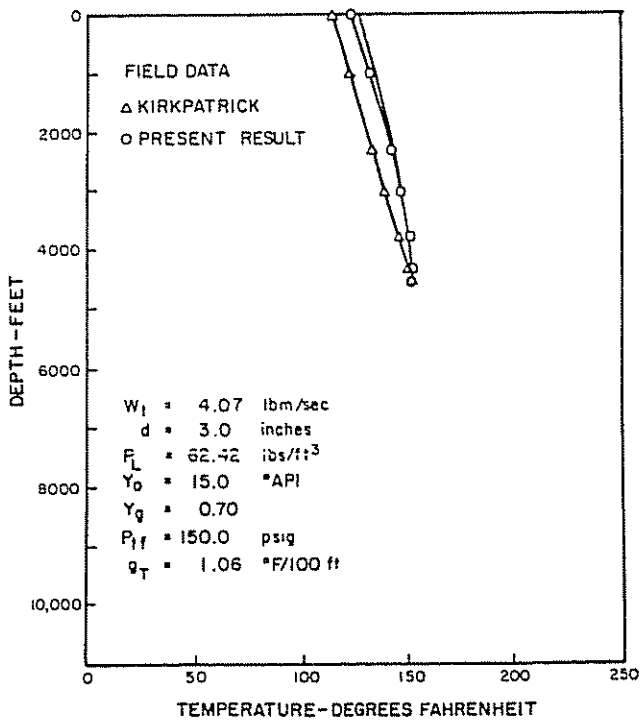


Fig. 3.47-B Temperature Profiles for High Liquid Density (after Shiu)

(2) From the given data, $g_T = 1.0625$ °F./100 ft, and by checking Fig. 3.44 the temperature gradient is found to be 0.58 °F./100 ft

(3) Consequently, the surface flowing temperature can be computed as

$T_{sur} = T_{BH} - 0.58$ (well depth)/100.

$T_{sur} = 173.0 - 0.58\,(5985)/100. = 138.29$ °F.

Solution II. Applying Shiu's Method

(1) Calculate W_t, ρ_L

Since $W_t = W_w + W_o + W_g$, then:

$$W_w = (62.4) \cdot (\gamma_w) \cdot (q_w) \cdot (5.615) \cdot \frac{1}{(86400)}$$

$$= (6214) \cdot (1.10) \cdot (11.15) \cdot (5.616) \cdot \frac{1}{(86400)}$$

$$= 0.050 \text{ lb}_m/\text{sec}$$

$$W_o = (62.4) \cdot (\gamma_o) \cdot (q_o) \cdot (5.615) \cdot \frac{1}{(86400)}$$

$$= \left[(62.4) \left(\frac{141.5}{(13/15) + 24.0)} \right) \right] \cdot (2218.5) \cdot (5.615) \left(\frac{1}{86400} \right)$$

$$= 8.188 \text{ lb}_m/\text{sec}$$

$$W_g = (0.0764) \cdot (q_g) \cdot (\gamma_g) \cdot (1000.) \cdot \left(\frac{1}{86400} \right)$$

$$= (0.0764) \cdot (1761.7) \cdot (0.7) \cdot (1000.) \cdot \left(\frac{1}{86400} \right)$$

$$= 1.091 \text{ lb}_m/\text{sec}$$

so

$$\rho_L = [(\gamma_W)(q_w) + (\gamma_o)(q_o)] \cdot (62.4) \cdot \frac{1}{(q_o + q_w)}$$

$$= [(1.10)(11.15) + \frac{(141.5)}{(131.5 + 24.0)} \cdot (2218.85)] \left(\frac{62.4}{2230} \right)$$

$$= 56.781 \text{ lb}_m/\text{ft}^3$$

(2) Since $P_{tf} = 1.50$ psig, $d = 3.0$ inches, $\gamma_o = 24.0$, and $\gamma_g = 0.7$, and from Eq. 3.6

$$A = e^{-5.065}(W_t)^{0.4882}(d)^{-0.3476}(\gamma_o)^{0.2519} (\gamma_g)^{4.724}(\rho_L)^{2.915}(P_{tf})^{0.2219}$$

$$= e^{-5.065}(9.329)^{0.4882}(3.0)^{-0.3476}(24.0)^{0.2519} (0.7)^{4.724}(56.781)^{2.915}(150.0)^{0.2219} = 2085.7 \text{ ft}$$

Therefore, by using Equation 3.44 the surface flowing temperature can be computed as:

$$T_{sur} = T_{BH} - g_T Z + g_T A\,(1 - e^{\frac{-z}{A}})$$

$$= 173.0 - (0.010625) \cdot (5985) + (0.010625) \cdot (2085.7) \cdot (1 - e^{\frac{-5985.}{2085.7}})$$

$$T_{sur} = 130.3 \text{ °F.}$$

The temperature can be calculated at various depths and the temperature traverse can be constructed.

3.4226 Unloading gradients and spacing of gas lift valves

The location of the optimum point of gas injection offers no serious problem. However, some controversy exists as to the best method for spacing gas lift valves down to this point. This may be done graphically or analytically. One question that generally arises is, "What unloading gradient should be used to space valves down to the point of injection?" In the initial unloading process, the annulus as well as the tubing may be standing full of fluid to the surface. After the operating valve is reached, a particular flowing gradient will be established between this point and the flowing surface tubing pressure. This gradient is satisfactory for spacing valves. A straight line may be drawn between the point of gas injection and the surface flowing tubing pressure and valves spaced using this average gradient line.

It is also possible to attain a so-called minimum gradient in the unloading process. This means that ample gas supply must always be available to obtain the minimum gradient from any unloading valve. If this is the case, valve spacing can be based on this line. This method is not recommended since the valves may get too far apart and reach a stymied condition. The additional initial cost of one or two gas lift valves is less than pulling the well for respacing valves. Spacing is discussed in detail in Section 3.29.

3.4227 Flow configuration sizes and production rates

The selection of the correct flow configuration size is important from both an economic and a rate standpoint. Two cases would be: (1) wells in South Texas that use 1", 1¼", or 1½" tubing for rates between 10-200 B/D; (2) wells in the Middle East that flow up the annulus of 7", 9⅝", 10¾", and 13⅜" casing to produce between 5,000-80,000 B/D. Various tubing sizes are used depending on gas volumes needed. The most common flow configurations in the U.S. are 2⅜" O.D. (1.995" I.D.) and 2⅞" O.D. (2.441" I.D.) tubing.

As an approximation, Table 3.42 serves as a guide in

TABLE 3.42
APPROXIMATE MAXIMUM AND MINIMUM FLOW RATES FOR GOOD CONTINUOUS FLOW OPERATIONS

Tubing size (diam., in.)	Maximum production rate (B/D)	Minimum production rate (B/D)
1	350	25-50
1¼	600	50-75
1½	1,000	75-125
2	2,500	200-250
2½	3,000	350-500
3	4,000	500-750
4	10,000	1000-1500
5½	30,000	3000-5000
7	50,000	5000-7000

Annular configuration	Maximum production rate (B/D)	Minimum production rate (B/D)
5½ × 2⅜	8,000	3,000
7 × 2⅜	15,000	5,000
7 × 2⅞	12,000	4,000
7 × 3½	10,000	3,500
9⅝ × 3½	15,000	5,000
9⅝ × 4	20,000	6,000
10¾ × 4	40,000	8,000
13⅜ × 4	60,000	10,000

determining the maximum rates possible under good continuous flow conditions and the minimum rate at which continuous flow operations should be attempted.

3.4228 Valve settings

Because valves have different operating characteristics, their shop settings are different. These shop calibrations vary with many factors, including whether the valve is pressure operated or fluid operated, pressure charged or spring charged, or a combination thereof. The temperature, gas column pressure, and tubing pressure may affect the setting of the valves. Valves may take a drop in pressure of 15-25 psi per valve down the hole, or they may all be set for the same surface opening or closing pressure. This drop is the difference in surface operating pressure from one valve to the next. It is sometimes necessary to insure no valve interference.

Since an optimum design usually results when the point of injection is reached with maximum pressure, the increase in valve opening pressures due to increased flowing temp may be utilized to reach the point of injection with maximum pressure.

A properly designed bellows valve installation may be set with all valves at the same surface opening or closing pressure. These valves are tubing-sensitive, the lower valve opening first because of the higher tubing back pressure acting against it.

Fluid operated valves are set to open principally on tubing pressure, yet can be influenced by gas column pressure and temperature.

3.4229 Approximations to be used in continuous flow installations

Although detailed design procedures should be used where possible, the following approximations will give a quick idea of various factors in continuous flow design.

(1) *Pressure needed:* 100 psi per 1000 ft of depth up to 1500 psi with a minimum of 300 psi. Pressures as high as 1800-2200 psi have been used with valves, but for good equipment life pressures above 1800 psi are not recommended. For orifices only, of course, any pressure can be used.

(2) *Gas volumes needed:* 150-250 scf/B per 1000 ft of lift. Present-day flowing gradient curves and computer calculations permit accurate checks on gas volumes needed.

(3) *Tubing allowable continuous flow depth:* (2⅜" and 2⅞" O.D. tubing only)

$$\frac{P_{so} - P_{wh}}{0.15} = \text{depth} \qquad (3.47)$$

(4) *Annulus allowable continuous flow depth:*

$$\frac{P_{so} - P_{wh}}{0.20} = \text{depth} \qquad (3.48)$$

P_{so} = operating lift pressure at surface

P_{wh} = flowing wellhead pressure

(5) *Maximum and minimum rates:* (see Sec. 3.227)

3.42210 Types of installations

Refer to Section 3.3 for the various types of installations. Continuous flow is generally run in an open or semiclosed installation. A packer should be run on every continuous flow installation if possible. Many wells require high production rates and will utilize casing flow.

3.42211 Use of multiphase flow correlations

Where possible, the most accurate multiphase flow correlation for a particular area should be used, incorporating field data and laboratory fluid property correlations. For the field design engineer, this may not be available and prepared flowing gradient curves can be used, such as found in Volume III of this book series. Most major companies have prepared working curves. These must be used with care since they have been prepared from one correlation, such as Hagedorn[4] or Duns and Ros,[5] and may be in error for certain flow conditions.

3.423 Design procedure for a continuous flow installation

To design a continuous flow gas lift installation, the following information should be obtained:

(1) depth of well
(2) size of tubing and casing (unless being specified)
(3) producing conditions of importance, such as sand, paraffin, etc.
(4) size and length of surface flowline
(5) separator back pressure
(6) expected surface flowing tubing pressure
(7) desired producing rate
(8) percent water cut
(9) sp gr of injection gas
(10) injection gas pressure and volume available at the well site (unless being specified)
(11) inflow performance characteristics of the well
(12) bottom hole temperature and geothermal gradient
(13) surface flowing temp
(14) API gravity of oil
(15) specific gravity of water
(16) specific gravity and amount of solution gas at differing pressures
(17) static BHP
(18) formation volume factors for the oil at differing pressures
(19) viscosity of the crude, surface tension, etc.

Many of these items are used in the design of a continuous flow installation. The greater the available information, the better the installation design.

In many cases, there is a lack of sufficient information to permit an optimum design. Procedures will be given for handling this type of well.

3.4231 Determining the point of gas injection

If sufficient well data is available, the first step in continuous flow design is to determine the point of gas injection. This procedure is the same for all types of gas

lift valves, but it may vary depending upon whether or not a drop in pressure between valves is taken. Also, it is necessary to understand this procedure in order to determine the rate possible by gas lift.

In this procedure, maximum benefit is derived from the available injection pressure. This means the deepest point of injection will be obtained, depending upon the injection gas pressure.

The following procedure serves as a guide for designing an installation on this basis. Plot all information on a sheet of rectangular coordinate paper.

(1) Plot depth on the ordinate (vertical) with zero depth at the top and maximum depth at the bottom. If flowing gradient curves are to be used, these scales must be the same.
(2) Plot pressure on the abscissa (horizontal) with pressure increasing from 0 at the origin to a maximum.
(3) Plot the static BHP at the correct depth (bottom of well).
(4) Determine the necessary "drawdown" in pressure to produce the desired flow rate. If this is to be the maximum rate possible, refer to Section 3.4232 for determining this value. Refer to Volume I, Chapter 1, for a discussion on inflow performance. For a constant PI (flow above bubble point pressure) find the drawdown in pressure for any rate.
(5) Subtract the drawdown from the static BHP to obtain the flowing BHP. Note this pressure at depth.
(6) From the point of static BHP, extend the static gradient line up the hole until it intersects the ordinate (depth). This will be the static liquid level in the well for $P_{wh} = 0$ and, in the event the well is not loaded, can be used as the point of location for the first gas lift valve.
(7) From the point of flowing BHP, plot the flowing pressure traverse below the point of gas injection. This may be calculated in detail or may be traced directly from flowing gradient curves. Another procedure used when detailed information is unavailable is to assume the flowing gradient beneath the point of injection can be approximated by the static gradient of the mixture of oil and water. For high flow rates of water and low gas-liquid ratios, this line may be nearly equal to the flowing gradient line.
(8) Select the surface operating pressure—pressure that can be maintained at the well-site to operate the gas lift well. Allow for control of the injection gas and safety in variable line pressures. The full line pressure less some safety factor (kick-off pressure) may be used to unload the well initially, but the operating pressure is used in finding the point of gas injection.
(9) Mark the kick-off pressure and surface operating pressure at zero depth and extend downward until intersecting the flowing gradient line of Step 7, noting the increase in pressure (gas column weight). Normally, a static gas column can be assumed, but in some cases the drop in pressure due to friction must be accounted for. This is particularly true for annular flow when gas is being injected down a tubing string. For example, if gas is being injected downward in 2⅜" O.D. tubing (1.995" I.D.), the friction approximately off-sets the increased column pressure at a rate of 2.75 million scf/day for $P_{so} = 1000$ and for $\gamma_g = 0.60$. Even for gas injected down the annulus, the friction should be checked for extremely high gas flow rates. Refer to Volume III of this series for curves showing this pressure loss.
(10) Mark the point where the operating casing pressure intersects the flowing gradient line as the point of balance between tubing and casing pressure. For an open installation (no packer) this would be the annular working fluid level.
(11) Subtract 100 psi from the pressure in the casing at this point (Fig. 3.41(B)). Return up the hole a distance equivalent to this 100 psi on the flowing BHP line and note this point. This will be the point of gas injection. This is necessary to ensure gas entry and to ensure that the well can be unloaded to this point of gas injection.

The selection of this differential (100 psi for this example) is extremely important and is controlled by the valve spacing. (See Section 3.42922)

(12) Approximate the flowing well-head pressure (P_{wh}) and mark this value at zero depth. A detailed procedure will be given later for incorporating this into the design procedure.
(13) Connect the flowing well-head pressure to the point of gas injection by means of computer calculations or by tracing the appropriate flowing gradient curve in Volume III of this series. This curve will in turn give the total GLR required to produce the well.

By subtracting solution gas from this total gas volume, the required injection GLR is obtained, from which the required gas flow rate can be determined.

The determination of this point of gas injection is correct for a particular well at a particular time in its life. Depending upon the type of drive mechanism by which this well produces, its characteristics may change considerably in one or two years.

For example, it may be experiencing a fast decline in pressure and/or a changing PI. Therefore, a close look at the expected well performance should be made at this time. The point of gas injection may lower a great deal within the following year or two. If so, the location of the point of injection as anticipated for one and two years ahead should be determined. This will prevent pulling the well in the near future to respace valves.

The following example shows the method for finding the point of gas injection. (Refer to Fig. 3.48).

EXAMPLE PROBLEM #9

Given information:

Depth = 8000 ft
Desired rate = 1000 B/D (100% oil)
Tubing size 2⅜" O.D. (1.995" I.D.)
Flowing well-head pressure = 100 psig (P_{wh})
Static bottom hold pressure = 2650 psig ($\overline{P}_R$)
Productivity index = 2 (assume constant)
Solution gas-oil ratio = 200 scf/B (R_s)
Specific gravity of injection gas = 0.70 (γ_g)

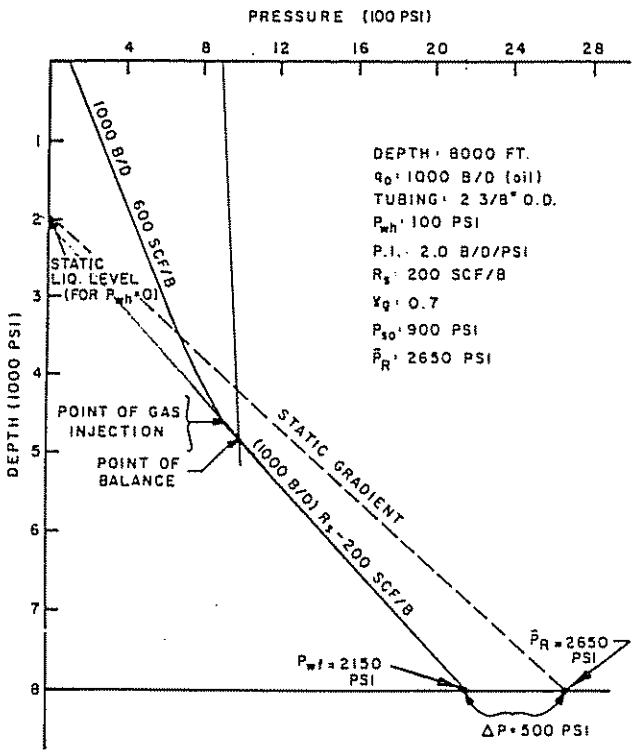


Fig. 3.48 Example Problem to Find Point of Gas Injection

Surface operating pressure = 900 psig (P_{so})
°API = 40
Bottom hole temperature = 200°F.
Flowing surface temperature = 120°F.

Determine the point of gas injection:

(1) Plot depth scale on vertical (0-8000 ft)
(2) Plot pressure scale on horizontal (0-2800)
(3) Plot $\bar{P}_R$ (2650 psig at 8000 ft)
(4) From the PI (productivity index) calculate the pressure drawdown necessary to give the desired production rate:

$$\Delta p = \frac{q_L}{J} = \frac{1000}{2} = 500 \text{ psi} \qquad (3.49)$$

where:

q_L = total liquid rate
J = productivity index

(5) Subtract the drawdown from the static BHP to obtain the flowing BHP:

$P_{wf} = \bar{P}_R - \Delta P = 2650 - 500 = 2150$ psi

Plot 2150 at 8000 ft

(6) From the point of static BHP, extend the static gradient line up the hole until it intersects the ordinate.

This gives the static fluid level for zero wellhead pressure. It is of no value in determining point of gas injection but may influence the location of the top valve. For a 40° API oil this represents a gradient of 0.357 psi/ft and intersects the ordinate at 1978 ft.

(7) From the point of flowing bottom hole pressure (2150 psi) extend the flowing gradient for R_s = 200 scf/B upward until approximately reaching a pressure equivalent to P_{so} (900 psi). This will be sufficient to obtain the point of gas injection. This may be calculated or traced from Figure 3B.1 (Appendix). (Similar curves may be found in Volume 3 of this series.)

(8) Mark the surface operating pressure (P_{so} = 900 psi) at the surface.

(9) Extend a line representing gas column pressure increase downwards from P_{so} = 900 until intersecting the flowing gradient line of Step 7. For 900 psi this will be 21.2 psi/1000 ft (note Fig. 3A.1—Appendix).

(10) Mark the point where the operating casing pressure intersects the flowing gradient line as the point of balance between tubing and casing pressures. This is at 4800 ft.

(11) Subtract 100 psi from the pressure at this point. Return up the hole a distance equivalent to this 100 psi on the flowing BHP line and note this point. This is the point of gas injection and is 4625 ft.

(12) Plot the surface flowing tubing pressure at zero depth (100 psi).

(13) The next step is to find the correct gas liquid ratio that will connect the flowing wellhead pressure of 100 psi to the point of gas injection. This may be calculated or traced from a figure similar to Figure 3B-1 (Appendix B) and is found to be 600 scf/B, which is the total gas-liquid ratio.

(14) Find the total injection gas necessary.

(Total G/L − R_s) q_L = Gas Vol. =
(600 − 200)(1000) = 400,000 scf/D

(15) Find the correct orifice size (gas lift valve port size) to pass the correct volume of gas.

(a) Find the temperature at the operating injection depth. For this case, assuming a linear temperature gradient the temperature at 4625 ft is $120°\text{F.} + \frac{200 - 120}{8000}(4625) = 176°\text{F.}$

(b) Correct the gas volume of 400,000 scf/D for temperature and specific gravity by the following equation.

Correction factor = $.0544\sqrt{\gamma_g T}$ (3.410)
where T = °R
Corr. factor = $(0.0544)\sqrt{(0.7)(176 + 460)} = 1.148$
Corr. volume = (1.148)(400,000) = 459,131 scf/D

(c) Refer to Figure 3C.1(a) in Appendix 3C to find the choke size. The upstream and downstream pressure read from Fig. 3.8 is 1000 psi and 900 psi, respectively. The correct valve port size is then between an 11/64 and 12/64 size. Select the 12/64" size.

CLASS PROBLEMS
(TO FIND POINT OF GAS INJECTION)

(1) Given:

8000 ft	2⅜" O.D. Tubing
q_o = 1000 B/D (all oil)	γ_g = 0.7
P_{wh} = 100 psi	P_{so} = 900 psi
R_s = 200 scf/B	$\overline{P}_R$ = 2650 psi
PI = 2	
Temp Bottom = 170°F.	
Temp Top = 100°F.	

Find: (a) Point of gas injection
(b) Gas volume needed
(c) Valve port size
(d) Repeat a, b, and c for P_{so} = 1200 psi

(2) Given:

q = 600 B/D (50% water)	2" Tubing
$\overline{P}_R$ = 1950 psi	G/O = 200 scf/B
PI = 1.5	Depth = 6000 ft
γ_g = 0.65	P_{so} = 1000 psi

Find:

(a) Point of gas injection
(b) Gas volume needed
(c) Valve port size

(3) Given:

9000 ft	G/O = 200 scf/B
$\overline{P}_R$ = 2900 psi	γ_g = 0.7
PI = 5	q_o = 4000 B/D
4" Tubing	Temp Bottom = 180°F.
P_{wh} = 100 psi	Temp Top = 120°F.

Find:

(a) Point of gas injection
(b) Gas volume
(c) Port size

(4) Given:

3" Tubing	50% oil
P_{wh} = 100 psi	q_L = 2000 B/D
R_s = 400 scf/B	γ_g = 0.8
Depth = 10,000 ft	P_{so} = 1100 psi
$\overline{P}_R$ = 2700 psi	
PI = 5	
Temp Top = 120°F.	
Temp Bottom = 220°F.	

Find:

(a) Point of gas injection
(b) Gas volume

(5) Given:

3" Tubing	All oil
q_o = 3000 B/D	γ_g = 0.65
$\overline{P}_R$ = 2800 psi	PI = 7.5
GOR = 300 scf/B	9000 ft

Find:
(a) Point of gas injection
(b) Gas volume
(c) Orifice size

(6) Given:

4" Tubing	q_o = 4000 B/D
8000 ft	50% oil - 50% water
P_{so} = 1200 psi	γ_g = 0.8
$\overline{P}_R$ = 2800 psi	PI = 4
R_s = 400 scf/B	
Temp Bottom = 170°F.	
Temp Top = 100°F.	

Find:
(a) Point of gas injection
(b) Gas volume
(c) Orifice size

(7) Given:

Depth = 9000 ft
2½" Tubing
G/O = 400 scf/B
90% water
P_{wh} = 80 psi
$\overline{P}_R$ = 3050 psi
PI = 5
q_L = 1000 B/D
P_{so} = 1200 psi
γ_g = 0.65

Find:
(a) Point of gas injection
(b) Gas volume needed
(c) Port size needed

(8) Given:

P_{wh} = 100 psi
PI = 1.0
R_s = 200 scf/B
γ_g = 0.65
8000 ft
$\overline{P}_R$ = 2800 psi
q_L = 500 B/D (95% water)
P_{so} = 900 psi
2⅜" O.D. Tubing

Find:

(a) Point of gas injection
(b) Gas volume needed
(c) Port size needed

3.4232 Determining flow rates possible by gas lift, by Pedro Regnault

3.42321 Introduction

As in a flowing well, the variables which affect production rates in continuous flow gas lift can also be classified into two groups: those which can be controlled and those over which there is little or no control. The first group includes flowline size and length, tubing size and length, surface restrictions, and possible separator pressure. The other group includes the fluid properties, the reservoir static pressure, and the inflow ability or productivity index.

The maximum flow rate possible from a continuous flow gas lift well can be determined using the same procedure as in flowing wells, except that two separate flowing pressure traverses are encountered when gas is being injected at any point higher than the total depth of the well. The continuous flow gas lift problem requires a longer, more tedious solution because an additional variable, the gas-liquid ratio, must be considered.

For a constant liquid flow rate, the gas-liquid ratio

will remain constant up to the point of gas injection from the bottom of the well (solution gas only). From the point of gas injection to the surface, the gas-liquid ratio increases due to the gas injected. This total gas-liquid ratio is the sum of the solution gas-liquid ratio plus the injection gas-liquid ratio.

To determine the flow rate possible by continuous flow gas lift, two diagrams are constructed:

(1) pressure-depth diagram which determines several variables, such as the injection gas point, flowing bottom-hole pressure (P_{wf}), and surface flowing pressure (P_{wh}); and

(2) pressure-flow rate diagram which includes the inflow performance curve and the multiphase flow performance curve.

Vertical flow correlations determine the possible flow rate with a constant wellhead pressure. A combination of the vertical flow correlations and horizontal correlations are used in the variable wellhead pressure approach.

Solutions are offered for (1) constant wellhead pressure and (2) variable wellhead pressure.

For a constant wellhead pressure, the pressure-flow rate diagram procedure and the equilibrium curve procedure are discussed; for a variable wellhead pressure, the wellhead pressure-flow rate diagram procedure and the flowing bottom-hole pressure-flow rate diagram procedure are discussed.

3.42322 Solution for a constant wellhead pressure

A constant surface wellhead pressure may or may not exist. Assuming no wellhead restrictions, a constant wellhead pressure for, say, offshore wells on platforms with very short flow lines can be assumed. To predict the maximum flow rate possible by continuous flow gas lift with a constant wellhead pressure, two methods can be used:

(1) the pressure-flow rate diagram procedure

(2) the equilibrium curve procedure

In both methods, the flowing bottom-hole pressure is the controlling parameter in determining the production rate. One solution determines the flowing bottom-hole pressure by using a depth-pressure diagram in conjunction with a pressure-flow rate diagram to determine the maximum production. For the equilibrium curve solution, the flowing bottom-hole pressure is determined from the depth-pressure diagram by interpolation, and the maximum flow rate is determined from the inflow ability of the well. That is:

$$q = (\overline{P}_r - P_{wf})\,PI \text{ (for a constant PI)} \qquad (3.411)$$

where:

q = production flow rate, B/D
P_{wf} = flowing bottom-hole pressure, psi
$\overline{P}_r$ = static bottom-hole pressure, psi
PI = productivity index, B/D per psi drawdown

Two possibilities exist for both solutions: unlimited gas available and limited gas available. Unlimited gas supply permits obtaining the minimum flowing gradient in the tubing string. The limited gas supply will probably mean less production. This is not necessarily true in all cases because the limited gas supply may be sufficient to obtain the maximum rate possible.

To predict flow rates possible by continuous flow gas lift for wells with a constant wellhead pressure, it is only necessary to use the vertical multiphase flow correlations. Fluid properties, the static bottom-hole pressure, diameter of tubing, available gas pressure for injection, and the productivity index are known. With this information and assuming no well restrictions, the rate is determined by the following procedures.

3.423221 Pressure-flow rate diagram procedure

From a constant wellhead pressure, the flowing gradient curve for an assumed rate above the point of injection is calculated or traced for a total gas-liquid ratio which is close to the available or the minimum gradient until it reaches the injection point line constructed from gas injection pressure available. From this point to the bottom (total depth), the flowing gradient curve below the injection point is traced or calculated considering the solution gas only (formation gas-liquid ratio), and the flowing bottom-hole pressure is determined. This procedure is repeated for different flow rates.

With the values of flowing bottom-hole pressure and considering the inflow performance of the well in a pressure-flow rate diagram, the maximum possible flow rate is determined. The maximum injection depth is also determined graphically from the depth-flow rate diagram.

For the two possibilities that exist (unlimited gas available and limited gas available), the solution procedure is the same except that if limited gas is available, the gas-liquid ratio above the point of gas injection must be determined for each flow rate depending on the total injection gas volume available.

Step-wise procedure:

(1) Using a sheet of rectangular coordinate paper, prepare a graph of well depth vs. pressure.

(2) Plot depth on the ordinate (vertical) with zero depth at the top and maximum depth at the bottom.

(3) Plot pressure on the abscissa (horizontal) with pressure increasing from zero at the origin to a maximum. The graph should have the same scale as the flowing gradient curves if they are to be used.

(4) Plot the gas surface operating pressure (P_{so}) at the surface.

(5) Extend the surface operating pressure down the hole, taking into account the increase in gas column pressure from the gas specific gravity and appropriate correlation. For low gas injection rates use the static gas column. For high injection rates and annular flow, consider the frictional loss of the gas.

(6) Subtract 100 psi from the gas surface operating pressure (P_{so}) and plot this value at the surface. From this point, extend a line parallel to the gas surface operating pressure line to the total depth.

(7) The line obtained in Step 6 represents the position of the various injection points depending upon the production rates.

(8) Plot the surface wellhead pressure at zero depth.

(9) Assume various flow rates.
(10) If flowing gradient curves are to be used, select the proper vertical flowing gradient curve for each flow rate assumed. The following steps may also be performed by computer calculations.
(11) Overlay and align the depth-pressure graph on the chosen flowing gradient curve and slide downward along the baseline (depth in feet) until the flowing surface wellhead pressure intercepts the highest gas-liquid ratio (the minimum fluid gradient possible) for unlimited gas available. Trace the flowing gradient above the point of injection downwards until intersecting the injection pressure line obtained in Step 7. This represents the point of gas injection. This step may also be performed by computer calculations using an appropriate multiphase flow correlation.

For limited gas, the total gas-liquid ratio must be obtained as follows:

$$GLR_i = \frac{q_g}{q_L} \tag{3.412}$$

where:

GLR_i = injection gas-liquid ratio in Scf/B
q_g = injection gas volume available in SCF/D
q_L = flow rate assumed in bpd

then:

$$TGLR = GLR_i + R_s \tag{3.413}$$

where:

TGLR = total gas-liquid ratio in Scf/B
R_s = formation gas-liquid ratio in Scf/B

The procedure is the same as for unlimited gas except that the curve for the respective total gas-liquid ratio available is used instead of the minimum gradient curve.

(12) From the point of gas injection obtained in Step 11, slide the depth-pressure graph up or down until the desired solution gas-oil ratio is reached (formation gas-liquid ratio) and from this point trace the flowing gradient traverse below the point of gas injection until it reaches the depth of the well (center of perforation). This step may be computer calculated.
(13) Note the flowing pressure at total depth. Repeat this procedure for each flow rate assumed.
(14) A pressure-flow rate diagram is prepared which includes the inflow performance curve and the gas lift flow rate vs. flowing pressure data (multiphase flow curve).
(15) The flow rate possible (also the maximum rate possible from the well since we have no restrictions) and the corresponding flowing pressure are read from the intersection of the two curves obtained in step 14.

3.423222 Equilibrium curve procedure

For the equilibrium curve procedure, the maximum flow rate possible from a well by continuous gas lift is predicted by interpolation, using the depth-pressure diagram, where the interrelation between the four variables (production rate, total gas-liquid ratio, injection depth, and pressure inside the tubing at injection depth) are represented by an equilibrium curve.

The equilibrium curve is defined as the locus of intersection of points of the flowing pressure gradient curves for various production rates from the reservoir depth upwards with the total gas-liquid ratio flowing pressure gradients for corresponding production rates as extended downwards from the wellhead pressure at zero depth.

From the flowing bottom-hole pressures, as determined from inflow ability for different flow rates, the flowing gradients (for formation gas-liquid ratio only) curves are traced upwards. The gas lift gradients are traced downward, beginning from the given constant wellhead pressure, selecting a total gas-liquid ratio which is close to the available or required amount. Intersection of the curves of equal rates of the two sets gives the equilibrium points. The curve through these points gives the equilibrium curve.

The maximum obtainable injection point is the interception of the equilibrium curve with the gas injection (gas injection line) gradient. Find the flowing bottom-hole pressure by interpolation and predict the maximum possible flow rate from the inflow ability of the well.

The significance of the equilibrium curve is that it gives the equilibrium conditions at any depth after the well has started to produce. These conditions determine whether or not a deeper injection point can be reached. It also shows the production rates possible with higher or lower injection pressures, different amounts of gas-lift gas, and future conditions or well characteristics, e.g. lower $\overline{P}_r$, PI, formation GLR.

As in the other method, two possibilities can exist: unlimited gas available and limited gas available.

Step-wise procedure

Although the procedure is given for using "gradient curves," the use of computer calculations would follow in the same manner.

(1) Using a sheet of rectangular coordinate paper, prepare a graph of well depth vs. pressure. The graph should have the same scale as the flowing gradient curves to be used.
(2) Plot depth on the ordinate (vertical) with zero depth at top.
(3) Plot pressure on the abscissa (horizontal) with pressure increasing from zero at the origin.
(4) Assume various production rates.
(5) For each flow rate, determine the required flowing bottom-hole pressure from the inflow ability of the well. that is:

$$P_{wf} = (\overline{P}_R - q_L/PI) \quad \text{(For constant PI)}$$

For solution gas drive wells, an inflow curve will be needed.

(6) Independent of the gas injection pressure available (P_{so}), determine the point of gas injection for each flow rate as follows:
(a) Mark the flowing bottom-hole pressure obtained in Step 5 on the depth-pressure diagram at the total depth and mark the wellhead pressure at the surface.
(b) Overlay and align the depth pressure graph on the flowing gradient curve chosen (flow

rate); slide up or down along the baseline (depth in feet) until it reaches the formation gas-liquid ratio. From this point, trace the corresponding flowing gradient upward (flowing gradient below gas injection point).

(c) Overlay and align the depth-pressure graph on the flowing gradient curve chosen (flow rate), slide down along the baseline (depth in feet) until the flowing surface wellhead pressure intercepts the gas-liquid ratio that gives the minimum value of the flowing gradient (unlimited volume of gas available) or the total gas-liquid ratio (limited volume of gas available). Trace downward until intersecting the flowing gradient curve obtained in Step 6b. The intersection of the curves of equal production rates gives the equilibrium points. Repeat for each assumed rate.

(7) Draw a curve through these intersecting points. This is known as the equilibrium curve.

(8) Mark the surface gas operating pressure at the surface. Extend the surface operating pressure downwards, taking into account the gas column pressure determined from the gas specific gravity and appropriate correlation.

(9) Subtract 100 psi from the gas surface operating pressure (P_{so}) and plot this value at the surface. Draw a line parallel to the gas surface operating pressure until it intercepts the equilibrium curve. This line is the point of gas injection line.

(10) Determine the actual point of gas injection by noting the intersection of the point of gas injection line and the equilibrium curve.

(11) Determine the flowing bottom-hole pressure by interpolation, after extending a curve from the point of gas injection to the bottom of the well.

(12) Determine flow rate possible (also the maximum rate from the well since there are no restrictions) from the inflow performance of the well. For a constant PI, this rate is determined:

$$q_L = (\bar{P}_R - P_{wf})(PI)$$

where $\bar{P}_{wf}$ is determined from Step 11.

3.423223 Example problems

To determine the maximum production rate from a well by continuous flow gas lift with a constant wellhead pressure, a correlation in the form of flowing pressure gradient curves is used. The Hagedorn and Brown[4] vertical multiphase flow correlation and the Duns and Ros[5] vertical multiphase flow correlation were used for comparison purposes.

The fluid physical properties used in this study were assumed to be the same as used in the correlations to predict the flowing pressure gradient curves.

The problem is worked using the two solution procedures described previously, i.e. pressure flow rate diagram and equilibrium curve method. The two possibilities of unlimited gas available and limited gas available are included. The following well information was used in the example problems worked by Marin:[6]

Depth of well = 8,000 ft
Tubing size = 2.441 in. I.D. (2⅞" O.D.)
Producing gas-oil ratio = 200 Scf/B
PI = 3 bpd/psi (assume linear)
P_{so} = 900 psi
$\bar{P}_r$ = 2,400 psi
P_{wh} = 120 psi (constant)
γ_g = 0.65
Production = 100% oil

Required: Maximum flow rate possible by gas lift.

Maximum production assuming unlimited gas available (pressure-flow rate diagram procedure)

(1) Using a sheet of rectangular coordinate graph paper, label depth vs. pressure coordinates (Fig. 3.49).

(2) From data given, plot on Fig. 3.49 the following:
Operating pressure (P_{so}): 900 psi at zero depth
Wellhead pressure (P_{wh}): 120 psi at zero depth

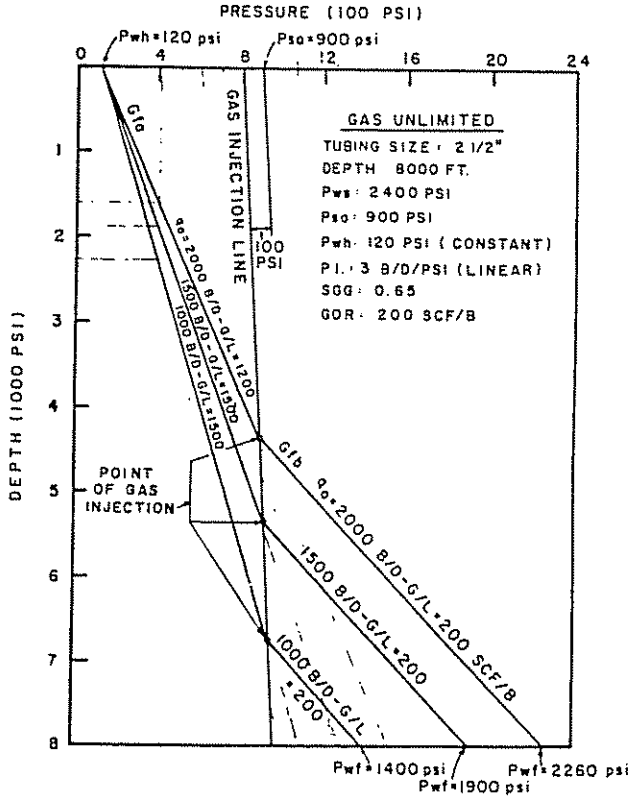


Fig. 3.49 Pressure Depth Diagram for Maximum Production Rate (Hagedorn Correlation)

(3) The operating pressure of 900 psi is extended to the total depth, considering the weight of the gas column as follows:

From Fig. 3A.1 (Appendix) for 900 psi and a specific gravity of 0.65, the gas gradient is 21.3 psi/1,000 ft. The weight of the column at total depth is determined as follows:

Weight of gas column = depth × gas gradient

= 8,000 ft × 21.3 psi/1,000 ft
= 170 psi

Operating pressure at 8,000 ft = 1,070 psi

(4) Production rates assumed: 1,000, 1,500, and 2,000 B/D.
(5) Flowing gradient curves are used in this solution and are selected for each flow rate and tubing size from Appendix 3B.
(6) With the flowing gradient curves, the injection depth point and the corresponding flowing bottom-hole pressures for each flow rate are obtained. The minimum gradient curve was used in each flow rate as the total gas-liquid ratio (For unlimited gas).

The results are tabulated in Table 3.43 and show values for both the Hagedorn and Ros correlation for comparison.

TABLE 3.43
FLOWING BOTTOM-HOLE PRESSURE

Assumed rate	TGLR	P_{wf} (Hagedorn)	Depth	P_{wf} (Ros)	Depth
1,000	1,500	1,400	6,680	1,005	7,800
1,500	1,500	1,900	5,350	1,600	6,050
2,000	1,200	2,260	4,300	2,100	4,500

(7) A pressure-flow rate diagram is prepared as noted in Fig. 3.410.
(8) The inflow performance curve is constructed assuming two rates: zero and 3,000 B/D, and the established static bottom-hole pressure $\overline{P}_r$ of 2,400 as follows:
With the following equation,

$$P_{wf} = (\overline{P}_R - q/PI) \quad \text{(For Constant PI case)}$$

for zero B/D, the flowing pressure is equal to the static bottom-hole pressure of 2,400 psi; for 3,000 B/D:

$$P_{wf} = (\overline{P}_R - q/PI) = (2{,}400 - 3{,}000/3) = 1{,}400 \text{ psi}.$$

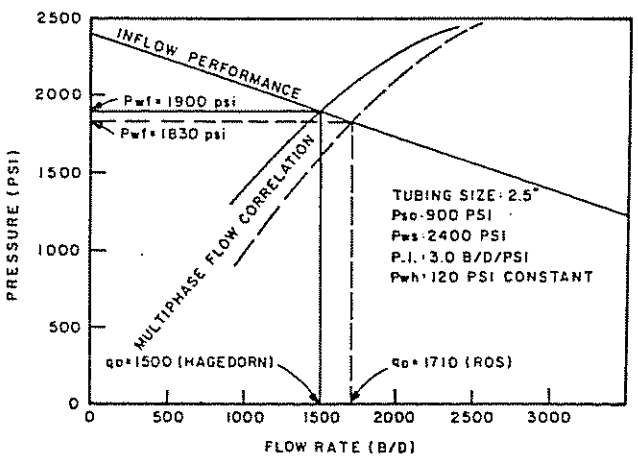


Fig. 3.410 Predicting Maximum Flow Rate-Constant Wellhead Pressure, Unlimited Gas Available (Hagedorn and Ros Solutions)

The two points are plotted in Fig. 3.410 and a straight line is drawn between them. This is the inflow curve.
(9) The flowing pressure data in Table 3.43, Step 6, are plotted on Fig. 3.410.
(10) From the intersection of the two curves obtained in Steps 8 and 9, (Fig. 3.410) the maximum possible flow rate is determined, and the injection depth for this flow rate is found from a plot of the depth-flow rate diagram.

The results are shown in Table 3.44.

TABLE 3.44
MAXIMUM FLOW RATE-UNLIMITED GAS AVAILABLE

Correlations	Possible flow rate	P_{wf}	Depth
Hagedorn and Brown	1,500	1,900	5,350
Ros	1,710	1,830	5,325

Solution by the equilibrium curve procedure

(1) Using a sheet of rectangular coordinate graph paper, label depth vs. pressure coordinates (Fig. 3.411).
(2) Production rates assumed: 1,000, 1,500, and 2,000 B/D.
(3) Flowing gradient curves selected for each flow rate are Figs. 3B.15, 3B.16a, and 3B.17a (Appendix 3B).
(4) For each production rate, determine the flowing

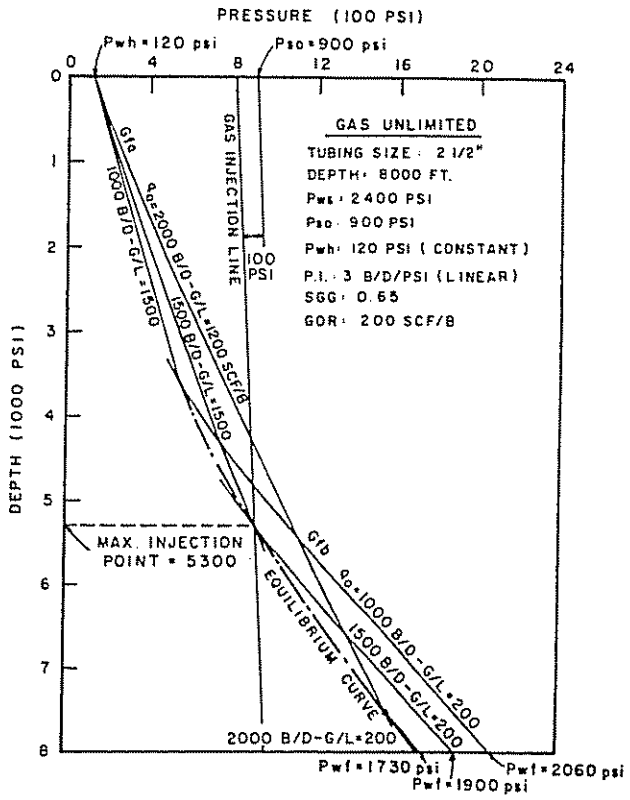


Fig. 3.411 Predicting Maximum Flow Rate—Equilibrium Curve Procedure, Unlimited Gas Available-Constant Wellhead Pressure (Hagedorn Correlation)

pressure from the inflow ability of the well. (Assume constant PI).

$$P_{wf} = (\overline{P}_R - q_L/PI)$$

The results are tabulated in Table 3.45.

TABLE 3.45
FLOWING BOTTOM HOLE PRESSURE FROM WELL INFLOW ABILITY (CONSTANT PI)

Assumed rate	P_{wf}
1,000	2,060
1,500	1,900
2,000	1,730

(5) The point of gas injection for each flow rate is determined as described previously. These points represent the equilibrium points. The results are tabulated in Table 3.46.

TABLE 3.46
POINT OF GAS INJECTION FOR UNLIMITED GAS AVAILABLE

Assumed rate	Injection point (Hagedorn)	Injection point (Ros)
1,000	3,600	2,260
1,500	5,300	4,500
2,000	7,525	6,825

(6) The connection of the equilibrium points determines the equilibrium curve, shown in Figure 3.411.
(7) Plot the surface gas operating pressure at the surface and determine the gas gradient. Draw the operating gas line downwards as described previously.
(8) Subtract 100 psi from the actual P_{so} of 900 psi, and draw a parallel line to the operating gas line. Where this line (gas injection line) intersects the equilibrium curve is the point of gas injection for this P_{so}.
(9) Extend a curve from the point of gas injection to the bottom of the well and, by interpolation, the flowing bottom-hole pressure is determined.
(10) The flow rate is determined for the flowing bottom-hole pressure found in Step 9, with the following equation:

$$q_L = (\overline{P}_R - P_{wf})\ PI \text{ (For Constant PI)}$$

The results are tabulated in Table 3.47.

TABLE 3.47
FLOW RATE FOR UNLIMITED GAS AVAILABLE (EQUILIBRIUM CURVE)

Correlations	Flow rate	P_{wf}	Depth
Hagedorn and Brown	1,500	1,900	5,300
Ros	1,710	1,830	5,350

Limited gas available pressure flow rate diagram procedure

The same problem is worked, except we now assume that limited volumes of gas are available.

The limited gas volumes are assumed to be 600,000, 400,000, and 200,000 Scf/D. The problem is worked for 600,000 Scf/D available.

(1) The solution is the same for Steps 1 through 3 as for the solution when unlimited gas is available. See Figure 3.412 for 600,000 Scf/D available.
(2) Same as Step 4 in solution for unlimited gas available. Assumed rates include 1,000, 1,500, and 2,000 B/D.
(3) Same as Step 5 in solution for unlimited gas available. Gradient curves selected for each flow rate are from Figs. 3B.2, 3B.3, and 3B.4 (Appendix 3B).
(4) For each flow rate assumed, determine the total gas-liquid ratio available above the point of gas injection. This includes gas in solution (R_s) plus the gas liquid ratio available from injection gas. For 1,000 B/D:

$$\text{Injection G/L} = \frac{600{,}000 \text{ Scf/D}}{1{,}000 \text{ B/D}} = 600 \text{ Scf/B}$$

$$= \text{GLR inj.}$$

$$R_s = 200 \text{ Scf/B (given data)}$$

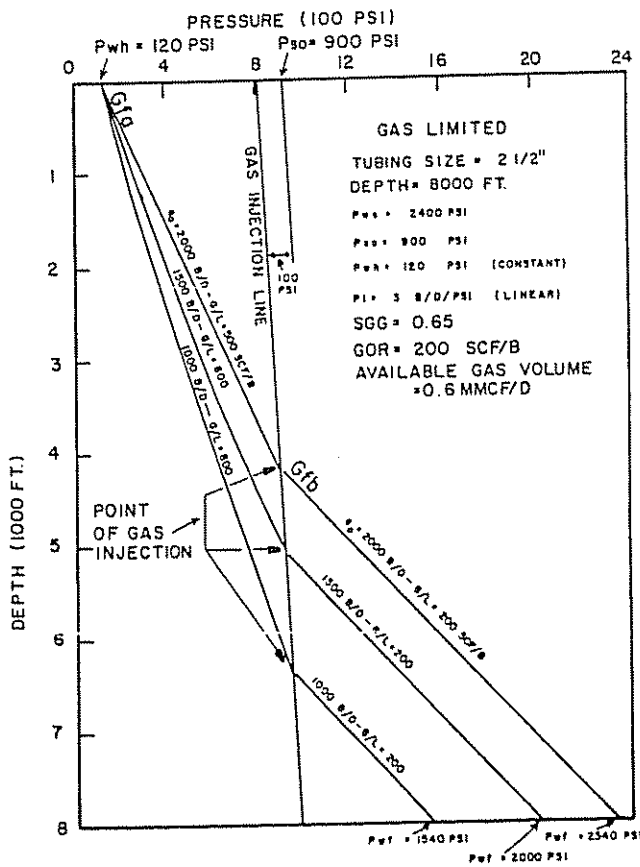


Fig. 3.412 Pressure Depth Diagram for Maximum Production Rate, Limited Gas Available-Constant Wellhead Pressure

Total G/L = 600 Scf/B + 200 Scf/B
= 800 Scf/B

Tabulate each total G/L (TGLR) for each flow rate.

TABLE 3.48
TOTAL GLR FOR LIMITED GAS AVAILABLE

Assumed rate	GLR_{inj} (600,000)	TGLR for 600,000 inj	TGLR for 400,000 inj	TGLR for 200,000 inj
1,000	600	800	600	400
1,500	400	600	467	333
2,000	300	500	400	300

(5) Determine the injection point depth and the corresponding flowing bottom-hole pressure for each production rate using the total gas-liquid ratio in Step 4, Table 3.48, for each flow rate. The results are tabulated in Table 3.49.

TABLE 3.49
FLOWING BOTTOM-HOLE PRESSURE FOR LIMITED GAS AVAILABLE (.6 MMcf/d)

Assumed rate	P_{wf} (Hagedorn)	P_{wf} (Ros)	Injection depth (Hagedorn)	Injection depth (Ros)
1,000	1,540	1,220	6,350	7,225
1,500	2,000	1,800	5,050	5,450
2,000	2,340	2,160	4,125	4,350

(6) A pressure-flow rate diagram is prepared as in Figure 3.413.
(7) The inflow performance curve is constructed as described previously. The results are plotted in Fig. 3.413 with the inflow performance curve as a straight line. (Constant PI).
(8) The flowing bottom-hole pressure data in Table 3.49, step 5, is plotted in Figure 3.413.

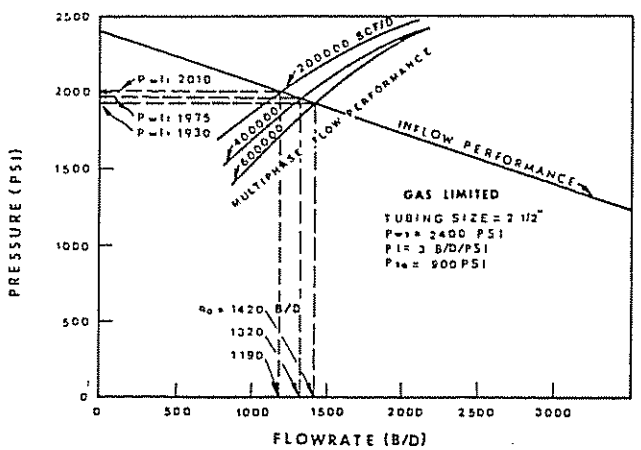


Fig. 3.413 Predicting Maximum Flow Rate-Constant Wellhead Pressure, Limited Gas Available (Hagedorn Correlation)

(9) From the intersection of the two curves obtained in Steps 7 and 8, (Fig. 3.413) the flow rate is determined. The results are tabulated in Table 3.410.

TABLE 3.410
FLOW RATES—LIMITED GAS AVAILABLE

Correlations	q_g	oil Flow rate (q_o)	P_{wf}
Hagedorn and Brown	600,000	1,420	1,930
Hagedorn and Brown	400,000	1,320	1,975
Hagedorn and Brown	200,000	1,190	2,010
Ros	600,000	1,580	1,875
Ros	400,000	1,440	1,920
Ros	200,000	1,260	1,980

Limited gas available
Solution by equilibrium curve procedure

The limited gas volume in this problem is assumed to be 600,000 Scf/d.

(1) Prepare pressure depth plot as in previous examples
(2) Production rates assumed: 1,000 and 1,500 B/D.
(3) Flowing gradient curves selected for each flow rate are found in Appendix 3B
(4) For each production rate, determine the flowing bottom-hole pressure from the inflow ability of the well

$$P_{wf} = \left(P_{ws} - \frac{q}{PI}\right) \qquad \text{(for constant PI)}$$

(5) For each flow rate and for R_s below the point of injection, determine the total G/L available for the pressure traverse above the point of gas injection.
(6) Determine the point of gas injection for each flow rate in the manner described previously. The results are tabulated in Table 3.411.

TABLE 3.411
POINT OF GAS INJECTION FOR LIMITED GAS AVAILABLE

Assumed flow rate (B/D)	Injection point (Hagedorn)	Injection point (Ros)
1,000	3,580	2,260
1,500	5,660	4,550

(7) Draw the equilibrium curve (see Fig. 3.414).

This solution is completed in the same manner as the previously described equilibrium curve procedure in extending the gas injection line downwards until intersecting the "equilibrium curve," extending on to the flowing pressure at total depth and determining the rate. The results are tabulated in Table 3.412.

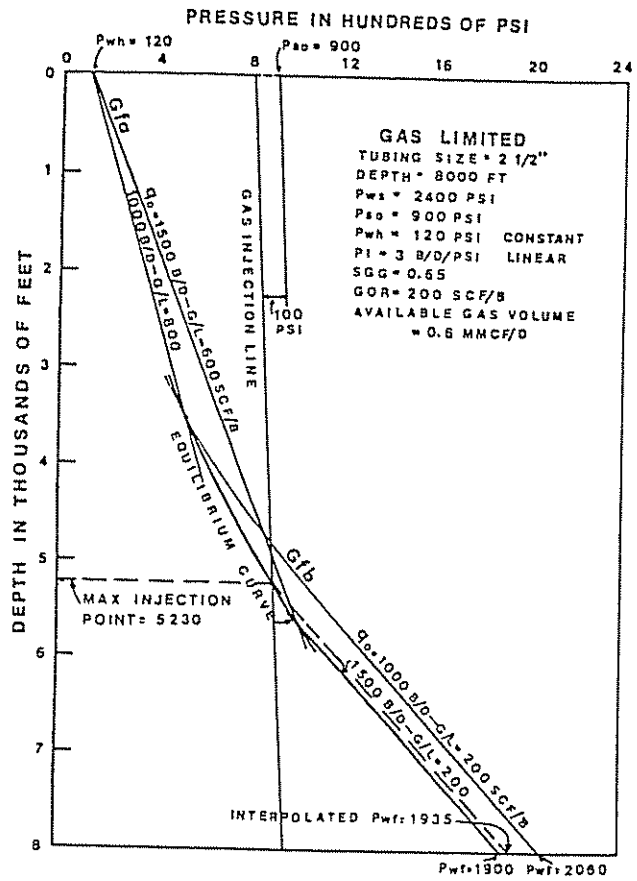


Fig. 3.414 Predicting Maximum Flow Rate, Limited Gas Available-Constant Wellhead Pressure, Equilibrium Curve Procedure (Hagedorn Correlation)

TABLE 3.412
MAXIMUM FLOW RATE FOR LIMITED GAS AVAILABLE
(EQUILIBRIUM CURVE SOLUTION)

Correlations	Possible flow rate	P_{wf}	Injection depth
Hagedorn and Brown	1,395	1,935	5,230
Ros	1,560	1,880	5,220

CLASS PROBLEMS—(constant wellhead pressure) to determine flow rates possible by continuous flow gas lift

(1) Given:

2½" Tubing × 7" casing
PI = 4 (assume constant)
$\gamma_g = 0.7$
$\overline{P}_{wh} = 120$ psi (constant)
$\overline{P}_R = 2500$ psi
$P_{so} = 950$ psi
$R_s = 200$ scf/b
100% oil

Find:
(a) The flow rate possible by continuous flow gas lift for tubing flow.
(1) Unlimited gas available
(2) Gas available = 500,000 scf/d
(3) Gas available = 200,000 scf/d

(2) Given:

$\overline{P}_R = 3000$ psi
PI = 5
$R_s = 300$ scf/b
4" Tubing
q_o = All oil.
$P_{wh} = 120$ psi (constant)
$\gamma_g = 0.8$
9000 ft.
$P_{so} = 1100$ psi

Find:
(a) Rate by natural flow
(b) Rate by gas lift

(3) Given:

$P_{so} = 900$ psi
2½" tubing
PI = 3
$\gamma_g = 0.65$
$\overline{P}_{wh} = 120$ psi (constant
$\overline{P}_R = 2400$ psi
Depth = 8000 ft
G/O = 200 scf/b
All oil

Find flow rate possible by gas lift
(a) Pressure flow rate diagram
(b) Equilibrium curve procedure

(4) Given:

$P_{wh} = 120$ psi (constant)
2½" tubing
G/O = 300 scf/B
$\gamma_g = 0.7$
$P_{so} = 1000$ psi
Depth = 6000 ft
50% oil
$\overline{PI} = 2.5$
$\overline{P}_R = 1600$ psi

Find flow rate by:
(a) Natural flow
(b) Gas lift

(5) Given:

$\overline{P}_{so} = 1100$ psi
$\overline{P}_R = 3000$ psi
PI = 6
G/O = 300 scf/b
$\gamma_g = 0.8$
Temp Top = 120°F.
Temp Bottom = 180°F.
Depth = 9500 ft
(All oil)
$P_{wh} = 160$ psi (constant)

Find flow rate:
(a) Natural flow
(b) Gas lift
(1) Unlimited gas
(2) Gas volume = 2,000,000 scf/day
(3) Gas volume = 500,000 scf/day

(6) Given:

Depth = 7250 ft
4" Tubing

100% oil
$R_s = 400$ scf/B
$P_{wh} = 200$ psi (constant)
$P_{so} = 1000$ psi
$\gamma_g = 0.7$

Year	$\overline{P}_R$	PI
1	3100	30
2	2400	15
3	1400	8
4	1000	4

Find flow rates at each year by:
(a) Natural flow
(b) Gas lift
(c) Cumulative production at end of year 4.

(7) Given:

2½" tubing
8500 ft
$\overline{P}_R = 3200$ psi
PI = 6
$P_{wh} = 120$ (constant)
100% oil
$\gamma_g = 0.7$
$P_{so} = 1000$ psi
$R_s = 300$ scf/B

Find flow rate by:
(a) Natural flow
(b) Gas lift

(8) Given:

A 10,000 ft well is to be placed on continuous flow gas lift with 2½ in. nominal tubing.

$\overline{P}_R = 2400$ psig
PI = 2
$P_{wh} = 130$ psig (constant for all rates)
$P_{so} = 1000$ psig (operating pressure)
90% salt water
GOR = 1000/1
Gas volume available = unlimited
$R_s = 400$ scf/B
$\gamma_g = 0.8$

Find:
The maximum production rate possible from this well.

(9) Given:

An 11,500-ft well is to be placed on gas lift in 2-in. tubing.

Gas operating pressure = 850 psig
PI = 8
$\overline{P}_R = 2500$ psig
Surface pressure constant at 110 psig
95% salt water
GOR = 1500 scf/b
Gas volume available = 1,500,000 scf/d
$\gamma_g = 0.7$
$R_s = 600$ scf/b

Find:
The maximum production possible from this well.

3.42323 Solution for a variable wellhead pressure, by Hugo Marin

To predict the flow rate possible from a gas lift well when a change in wellhead pressure occurs, it is necessary to utilize both vertical and horizontal multiphase flow correlations. This solution assumes no wellhead restrictions. The same variables that affect a well with constant wellhead pressure are considered, but some new variables must be considered to use the horizontal correlations: separator pressure and the length and diameter of the flow line. Therefore, as the rate is varied, the wellhead pressure will change because of the movement of the fluids from the wellhead to the separator.

The gas-liquid ratio will remain constant up to the point of injection. But above this point, the gas-liquid ratio increases due to the gas injected. To obtain the maximum production rate, there exists an optimum total gas-liquid ratio to maximize production. The injection of more or less gas than is required to give this optimum value results in a decreased production rate.

This gas-liquid ratio may be less than the gas-liquid ratio required to obtain the minimum flowing pressure traverse in the tubing string. This is because the pressure required at the wellhead to move the gas and liquid through the surface flowline to the separator increases as the gas volume increases. That is, extra gas in the surface flowline causes additional frictional pressure loss and offers no benefit in lightening the mixture density, because no fluids are lifted vertically. In the vertical tubing string, the additional gas also causes added frictional loss. However, it also lightens the mixture density, causing an overall pressure decrease (up to the minimum gradient).

In the prediction of the maximum flow rate possible by continuous flow gas lift with a variable wellhead pressure, two solution procedures are offered:
(1) wellhead pressure-flow rate diagram procedure
(2) flowing bottom-hole pressure-flow rate diagram procedure

In the wellhead pressure-flow rate diagram procedure, both the vertical and horizontal correlations determine wellhead pressures. From the plot of these values vs. flow rate, production rate is obtained.

In the flowing bottom-hole pressure-flow rate diagram procedure, the horizontal multiphase correlation determines the wellhead pressure, after which the vertical multiphase correlation determines the flowing bottom-hole pressure, considering the wellhead pressures obtained from the horizontal correlation. From a plot of flowing bottom-hole pressure vs. flow rate, the flow rate is determined.

3.423231 Wellhead pressure-flow rate diagram procedure

The maximum flow rate possible from a well by continuous flow gas lift is predicted by using a depth-pressure diagram to find permissible wellhead pressures and a length-pressure diagram to determine required wellhead pressures for various assumed rates, considering the variable total gas-liquid ratio for the different flow rates assumed. For each assumed flow rate, the

flowing bottom-hole pressure is determined from the ability of the well:

$$P_{wf} = (\overline{P}_R - q_L/PI). \text{ (For Constant PI)}$$

The allowable variable wellhead pressures are determined with the flowing bottom-hole pressure and for each flow rate in the depth-pressure diagram. From the flowing bottom-hole pressure at total depth, the flowing gradient curve below the injection point is calculated or traced upward for each flow rate, considering the formation gas-liquid ratio only until it reaches the injection point. From this point, the gradient curve above the injection point is calculated or traced for the different total gas-liquid ratios assumed until it reaches zero depth. At this point, the allowable wellhead pressure is determined.

The required wellhead pressure is determined in the length-pressure diagram. From the separator pressure at zero depth, the flowing horizontal gradient curve is traced and the wellhead pressure is determined at the total flowline length for the different total gas-liquid ratios assumed and for the different flow rates.

The wellhead pressures (vertical and horizontal) are plotted in a wellhead pressure-flow rate diagram. From this plot, the flow rate is obtained with an optimum gas-liquid ratio for maximum production.

The final production rate is a function of the flowing wellhead pressure, but this wellhead pressure is unknown until the final rate is established.

Procedure

(1) Using a sheet of rectangular coordinate paper, prepare a graph of depth vs. pressure.

(2) Plot depth on the ordinate (vertical) with zero depth at top and maximum depth at bottom.

(3) Plot pressure on the abscissa (horizontal) with pressure increasing from zero at the origin to a maximum. The graph should have the same scale as the vertical flowing gradient curves if they are to be used. For a computer approach to the problem, the computer plotter may be used.

(4) Plot the gas surface operating pressure (P_{so}) at the surface.

(5) Extend the gas surface operating pressure down the well, taking into account the static gas column weight determined from the gas specific gravity and appropriate correlation (Appendix 3A.1).

(6) Subtract 100 psi from the gas surface operating pressure and plot this value at the surface. Draw a line parallel to the gas surface operating pressure line extended to the total depth (gas injection line).

(7) Assume various flow rates.

(8) Select the proper vertical and horizontal flowing gradient curves for each flow rate assumed or prepare for computer calculations.

(9) For each flow rate, assume various gas-liquid ratios (total gas-liquid ratio). Use one value of gas-liquid ratio near the minimum gradient for each flow-rate.

(10) For each flow rate, determine the required flowing bottom-hole pressure from the inflow ability of the well:

$$P_{wf} = (P_r - q_L/PI) \text{ (For Constant PI)}$$

(11) Determine the permissible wellhead pressure for each flow rate and for the selected gas-liquid ratios for each rate using the vertical flowing gradient curves or from computer calculations:

(a) For an assumed flow rate, mark the flowing bottom-hole pressure obtained in Step 10 at the total depth of the depth-pressure diagram.

(b) Overlay and align the depth-pressure graph on the flowing gradient curve chosen.

(c) Slide the graph upward along the baseline (depth in feet) until the flowing bottom-hole pressure intercepts the formation gas-liquid ratio. From this point, and in an upward manner, trace the flowing gradient traverse below the point of gas injection until intersecting the gas injection line obtained in step 6. This intersection is the point of gas injection (depth of injection).

(d) From the point of gas injection obtained in the previous step, slide the graph up or down until it intercepts the total gas-liquid ratio selected. From this point, trace the flowing gas lift gradient curve until it intercepts zero depth. This intersection gives the permissible wellhead pressure.

Repeat this procedure for the same flow rate but with the other assumed gas-liquid ratios.

(12) Repeat the entire procedure used in Step 10 for the other assumed flow rates and gas-liquid ratios.

(13) Determine the necessary wellhead pressures to move the same assumed flow rates through the surface flowline to the separator for the same assumed gas-liquid ratios using computer calculations or the horizontal flowing gradient curves as follows:

(a) Prepare a length-pressure diagram, following Steps 1, 2, and 3 in this procedure. Mark the separator pressure (P_{sep}) at the surface.

(b) Overlay and align the length-pressure graph on the flowing gradient curve chosen; slide downward along the baseline (depth in feet) until the separator pressure intercepts the gas-liquid ratio selected. From this point, trace the horizontal flowing gradient curve until it intercepts the length of the flowline. This intersection gives the necessary wellhead pressure.

(c) Repeat this procedure for the same flow rate and the assumed gas-liquid ratios and for the other assumed flow rates and gas-liquid ratios.

(14) Tabulate the results of the wellhead pressures obtained from the vertical and horizontal multiphase flow correlations.

(15) Prepare a wellhead pressure-flow rate diagram and plot the results tabulated in Step 14. The intersection of a horizontal curve with a vertical curve of the same gas-liquid ratio represents the rate possible for that gas-liquid ratio.

(16) Draw a curve through the locus of gas-liquid ratio points. Obtain the maximum possible flow rate from the intersection farthest to the right

on the production rate scale, and determine the corresponding gas-liquid ratio and the required wellhead pressure.

3.423232 Flowing bottom-hole pressure-flow rate diagram procedure

The maximum flow rate possible from a well by continuous flow gas lift is predicted using the depth-pressure diagram and a length-pressure diagram to determine the flowing bottom-hole pressures from vertical flow correlations with the wellhead pressure first obtained from the horizontal flow correlations for the same assumed rate and gas-liquid ratio.

From the separator pressure at zero depth, the flowing horizontal gradient curve is traced and the wellhead pressure is determined for the different total gas-liquid ratios assumed and for the different flow rates.

With the wellhead pressures determined from the horizontal flow correlations and for each flow rate and different total gas-liquid ratio, the flowing bottom-hole pressures are determined from the vertical flow correlations. With these values and considering the inflow performance relationship of the wells in a pressure-flow rate diagram, the maximum flow rate is determined. The flow rate and corresponding gas-liquid ratio can be easily determined from a flow rate—gas-liquid ratio or flow rate-gas volume plot.

Stepwise procedure:

(1) Assume various flow rates.
(2) Assume various total gas-liquid ratios for each flow rate.
(3) Select the proper horizontal and vertical flowing gradient curves for each flow rate assumed or prepare for computer calculations. (See Appendix 3B)
(4) Determine the wellhead pressures necessary to move the assumed flow rates through the surface flowline to the separator, using the horizontal flow correlations.
(5) From the wellhead pressures of Step 4, determine the flowing bottom-hole pressures for each flow rate and for each gas-liquid ratio. Tabulate the results.
(6) Prepare a flowing bottom-hole pressure-flow rate diagram which includes the inflow performance curve. Construct the inflow performance curve for a constant PI by assuming two rates and finding the corresponding flowing pressures by the following equation:

$$P_{wf} = (\overline{P}_r - q_L/PI) - \text{(for constant PI)}$$

(7) Plot the results tabulated in Step 5 on the diagram of Step 6. The intersection of the multiphase flow performance curves for each gas-liquid ratio with the inflow performance curve gives the possible flow rates for each separate gas-liquid ratio.
(8) Obtain the maximum possible flow rate from the intersection furthest to the right on the production rate scale, and obtain the corresponding total gas-liquid ratio and the required flowing bottom-hole pressure from the diagram.
(9) Prepare a flow rate vs. gas-liquid ratio plot to select the maximum rate and the corresponding gas-liquid ratio.

3.423233 Example problems

For the following example problems, the Eaton horizontal multiphase flow correlation, the Hagedorn and Brown, and the Ros vertical flow correlations were used.

Work the problem using the two solution procedures described previously. The following information is given:

EXAMPLE PROBLEM #10

Depth of well = 7,000 ft
Flowline length = 3,000 ft
Tubing size = 3 in.
Flowline size = 3 in.
$\overline{P}_r$ = 2,600 psi
P_{so} = 900 psi
P separator = 80 psi
PI = 5 bpd/psi (assume linear)
GOR = 400 Scf/b
G/L = 200 Scf/b
Production = 50% oil - 50% water
Specific gravity gas = 0.70
TGLR = Variable
P_{wh} = Variable
Required: Maximum flow rate possible.

Maximum production assuming unlimited gas available. (Solution by wellhead pressure-flow rate diagram procedure.)

(1) Figure 3.415 is a graph of depth vs. pressure for the Hagedorn vertical flow correlation (Step 1 of stepwise procedure).
(2) From data given, plot on Figure 3.415 operating pressure = 900 psi
(3) Extend the operating pressure of 900 psi to total depth, considering the weight of the gas column as follows:

From Fig. 3A.1 (Appendix) the gas gradient for 900 psi and specific gravity of 0.7 gas is 23.7 psi/1,000 ft

The weight of the column at total depth is as follows:

weight of gas column = depth × gas gradient
= 7,000 ft × 23.6 psi/1,000 ft
= 165.2 psi

Operating pressure at total depth = 1065.2 psi. This pressure is noted at 7,000 ft and a straight line drawn between 900 psi at the surface and 1,065 psi at 7,000 ft.

(4) Subtracting 100 psi from the actual P_{so} = 900 psi, draw a parallel line to the operating gas pressure line. This represents the point of gas injection line.
(5) Assumed production rates include 1,000, 1,500, 2,000, and 3,000 B/D.
(6) Select vertical gradient curves for each flow rate and note in Figure B (Appendix 3B).

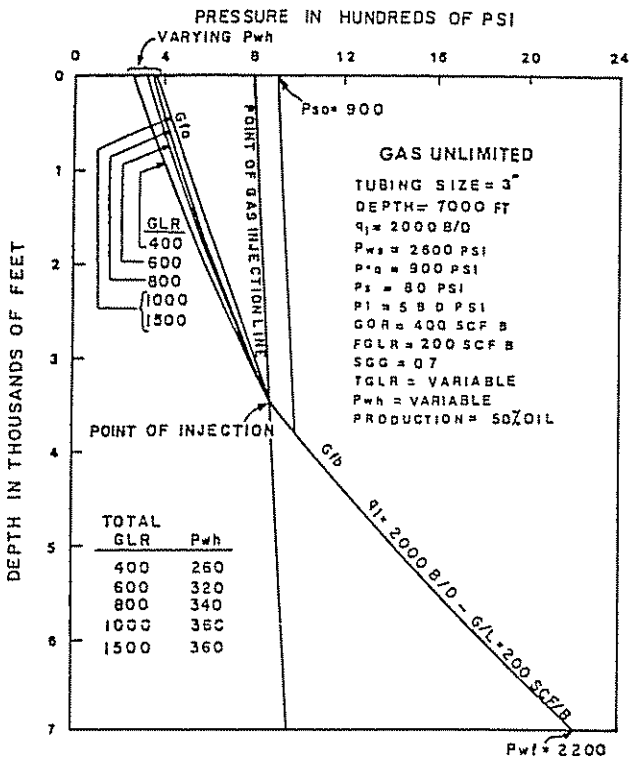


Fig. 3.415 Typical Depth-Pressure Plot That Must Be Made for Each Liquid Rate (Brown Vertical Curves)

(7) Total gas-liquid ratios assumed for each flow rate are 400, 600, 800, 1,000, and 1,500 Scf/b.

(8) For each flow rate, determine the required flowing bottom-hole pressure from the ability of the well as follows:

$$P_{wf} = (\overline{P}_R - q_L/PI). \quad \text{(Constant PI)}$$

For 2,000 B/D and given data of PI = 5 (linear) and P_{ws} = 2,600 psi,

$$P_{wf} = \left(2{,}600 - \frac{2{,}000}{5}\right) = 2{,}200 \text{ psi}$$

The other values of P_{wf} for the assumed flow rates are calculated in the same manner and are tabulated in Table 3.413.

(9) From the vertical gradient curves selected, determine the permissible wellhead pressure using the procedure described previously. Figure 3.415 shows the plot for an assumed rate of 2,000 B/D and for gas-liquid ratios of 400, 600, 800, 1,000 and 1,500 Scf/B. The same procedure is used for the other assumed flow rates using the same selected GLR's.

TABLE 3.413
FLOWING BOTTOM-HOLE PRESSURE FROM ABILITY OF THE WELL

Assumed rate	P_{wf}
1,000	2,400
1,500	1,500
2,000	2,200
3,000	2,000

(10) From a horizontal flow correlation or from the selected horizontal gradient curves, the required wellhead pressures are determined. Figure 3.416 shows the plot for an assumed rate of 2,000 B/D and gas-liquid ratios of 400, 600, 800, 1,000 and 1,500 Scf/B. Use the same procedure for the other assumed flow rates using the same selected GLR's. (Refer to Appendix 3B).

(11) The final tabulation of required and permissible wellhead pressures is found in Table 3.414.

(12) With the values shown in Table 3.414, make a wellhead pressure-flow rate diagram (refer to Figs. 3.417A and 3.417B). The maximum possible flow rate is obtained (Figs. 3.417A and 3.417B) by connecting all points of equal gas-liquid ratios and noting the point farthest to the right of the curve connecting these points. The corresponding gas-liquid ratio and flowing wellhead pressure is read directly from this plot.

The results are given in Table 3.415.

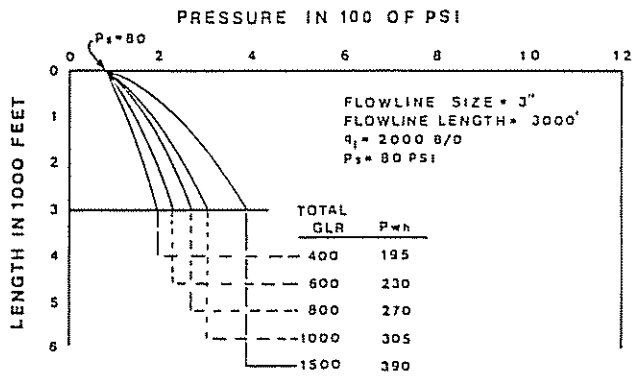


Fig. 3.416 Typical Length-Pressure Plot That Must Be Made for Each Liquid Rate (Brown Horizontal Curves)

TABLE 3.414
WELLHEAD PRESSURES FROM VERTICAL AND HORIZONTAL CORRELATIONS

Assumed rate (B/D)	TGLR	Vertical Correlations P_{wh} (Hagedorn)	Vertical Correlations P_{wh} (Ros)	Horizontal correlation (Eaton) P_{wh}
	400	370	345	145
	600	440	400	140
1,000	800	480	420	160
	1,000	500	460	170
	1,500	540	490	195
	400	325	280	160
	600	390	360	185
1,500	800	420	380	205
	1,000	430	400	225
	1,500	450	440	285
	400	260	220	195
	600	320	280	230
2,000	800	340	300	270
	1,000	360	330	305
	1,500	360	350	390
	400	110	45	280
	600	140	45	320
3,000	800	160	85	360
	1,000	160	90	425
	1,500	160	110	535

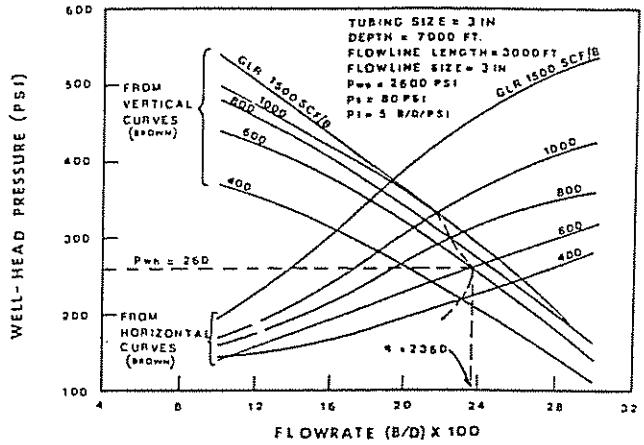


Fig. 3.417-A Predicting Maximum Flow Rate-Variable Wellhead Pressure (Hagedorn and Eaton Correlations)

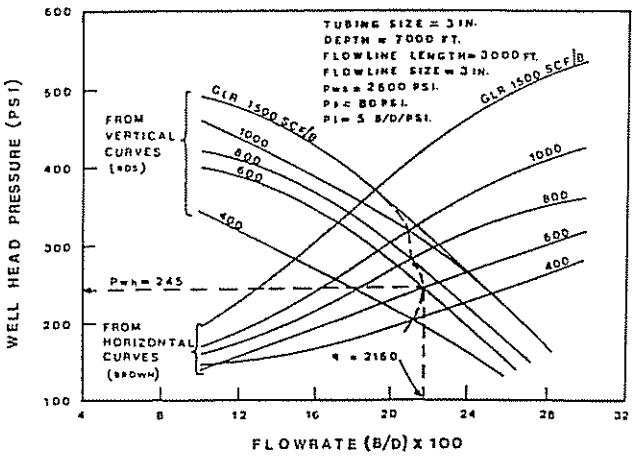


Fig. 3.417-B Predicting Maximum Flow Rate-Variable Wellhead Pressure (Ros and Eaton Correlations)

TABLE 3.415
MAXIMUM FLOW RATES FOR VARIABLE WELLHEAD PRESSURES

Correlations	Maximum flow rate	P_{wh}	TGLR
Hagedorn and Brown	2,360	260	600
Ros	2,160	245	600

Solution by flowing bottom-hole pressure vs. flow rate diagram procedure:

(1) Production rates assumed are 1,000, 1,500, 2,000, and 4,000 B/D
(2) Total gas-liquid ratios assumed are 400, 600, 800, 1,000, and 1,500 Scf/B.
(3) Select the proper horizontal and vertical flowing gradient curves or use a suitable computer calculation.
(4) Determine the wellhead pressures from horizontal curves or correlations as described previously. Figure 3.416 shows the plot for an assumed rate of 2,000 B/D. The same procedure is used for the other assumed flow rates, using the same selected GLR's.
(5) Obtain the flowing bottom-hole pressure from vertical flowing curves or from computer calculations for each flow rate with different total gas-liquid ratios. Figure 3.418 shows a typical plot that must be made for each flow rate. Prepare a tabulation, as noted in Table 3.416, which shows results from two correlations.
(6) Prepare a pressure-flow rate diagram as noted in Figs. 3.419 and 3.420.
(7) In preparing Figures 3.419 and 3.420, construct the inflow performance curve by assuming two rates, 800 and 3,000 B/D. The corresponding flowing bottom-hole pressures are found from the ability of the well as follows:

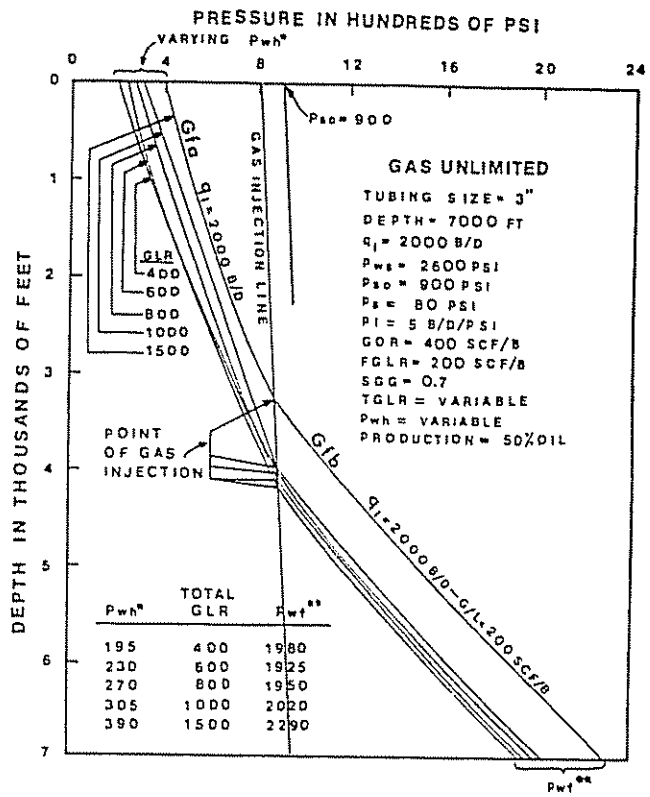


Fig. 3.418 Typical Depth-Pressure Plot That Must Be Made for Each Liquid Rate (Hagedorn Correlations)

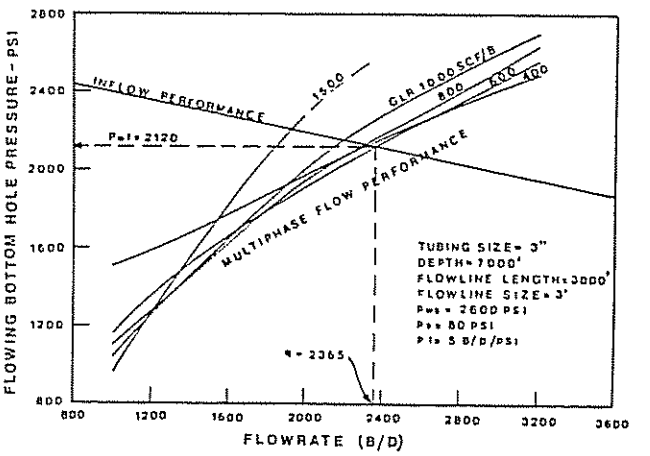


Fig. 3.419 Predicting Maximum Flow Rate-Variable Wellhead Pressure (Hagedorn Correlation)

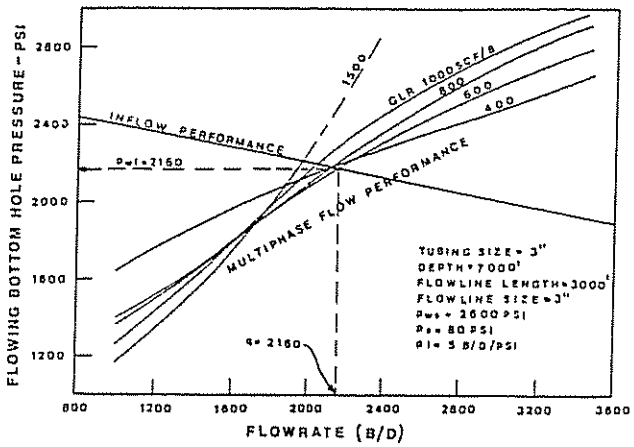


Fig. 3.420 Predicting Maximum Flow Rate-Variable Wellhead Pressure (Ros Correlation)

For 800 B/D,

$$P_{wf} = (\overline{P}_R - q_L/PI)$$
$$= 2{,}600 - 800/5 = 2{,}440 \text{ psi. (constant PI)}$$

For 3,000 B/D,

$$P_{wf} = (2{,}600 - 3{,}000/5) = 2{,}000 \text{ psi.}$$
(constant PI)

Plot the two pressures in Figs. 3.419 and 3.420 as straight lines for the constant PI case.

(8) Plot the flowing bottom-hole pressures from Table 3.416 step 5, in Figs. 3.419 and 3.420.

(9) Obtain the flow rate is from the intersection of the inflow performance curve and the multiphase flow curve furthest to the right on the flow-rate scale. Note the flowing bottom-hole pressure and the corresponding total gas-liquid ratio also on the same plot. The results are tabulated in Table 3.417.

TABLE 3.416
FLOWING BOTTOM-HOLE PRESSURES FROM VERTICAL AND HORIZONTAL CORRELATIONS

Assumed rate (B/D)	TGLR	Horizontal (Eaton) P_{wh}	Vertical P_{wf} (Hagedorn)	Vertical P_{wf} (Ros)
	400	145	1,500	1,645
	600	140	1,160	1,400
1,000	800	160	1,100	1,360
	1,000	170	1,045	1,260
	1,500	195	965	1,170
	400	160	1,730	1,920
	600	185	1,590	1,720
1,500	800	205	1,535	1,710
	1,000	225	1,550	1,700
	1,500	285	1,690	1,630
	400	195	1,980	2,135
	600	230	1,925	2,080
2,000	800	279	1,950	2,090
	1,000	305	2,020	2,160
	1,500	390	2,290	2,260
	400	390	2,860	2,840
	600	465	2,985	2,940
4,000	800	515	3,090	3,044
	1,000	560	3,180	3,150
	1,500	660	3,180	3,365

TABLE 3.417
FLOW RATE FOR VARIABLE WELLHEAD PRESSURES

Correlations	Flow rate	P_{wf}	G/L
Brown	2,365	2,120	600
Ros	2,160	2,160	600

(10) To more easily determine the maximum flow rate and the corresponding gas-liquid ratio, a plot of rate vs. gas-liquid ratio, such as Figure 3.421, is prepared. Data for this plot is obtained in Figs. 3.419 and 3.420 at the intersection of each multiphase flow curve with the inflow performance curve. Data from Figs. 3.419 and 3.420 are tabulated in Table 3.418.

TABLE 3.418
FLOW RATE VS. GAS-LIQUID RATIO

GLR	Flow rate (Hagedorn)	Flow rate (Ros)
400	2,290	2,120
600	2,365	2,160
800	2,280	2,140
1,000	2,180	2,040
1,500	1,930	2,000

From Fig. 3.421, the maximum flow rate is determined. These values are the same as noted in Table 3.417.

EXAMPLE PROBLEM #11

One other problem of this type as worked by Fonseca is shown in Figures 3.422 and 3.423.[8]

Given:

$\overline{P}_R$ = 2400 psi
PI = 3 (constant)
P_{sp} = 900 psi
γ_g = .65
°API = 35
Separator pressure = 80 psi
R_s = 200 scf/B

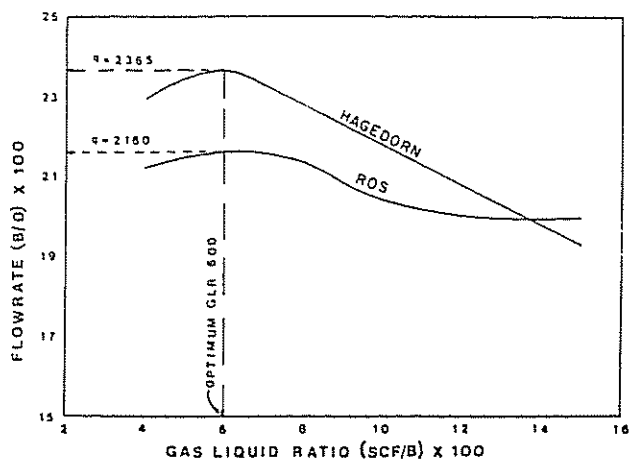


Fig. 3.421 Predicting Maximum Flow Rate From Flow Rate-Gas-Liquid Ratio Plot

Depth = 8000 ft
Flowline length = 4000 ft
Flowline diameter = 3"
Tubing size = 2⅞" O.D.

Figure 3.422 shows a pressure flow rate diagram on a large scale which shows the effect of various gas injection volumes. Values from this figure are tabulated in Table 3.419. The production rate decreases after an injection gas volume of 1,000,000 scf/D, and the incremental increase in production is very low after 600,000 scf/D.

Fig. 3.423 shows the data of Table 3.419 in a plot which illustrates the performance for this well. The maximum rate is 1412 B/D at a gas injection depth of 4985 feet and for an injection gas-liquid ratio of 850 scf/B.

CLASS PROBLEMS (VARIABLE WELLHEAD PRESSURE)

(1) Given:
Tubing = 8000 ft
Flowling = 4000 ft
$\overline{P}_r$ = 2400 psi
G/O = 200 scf/B

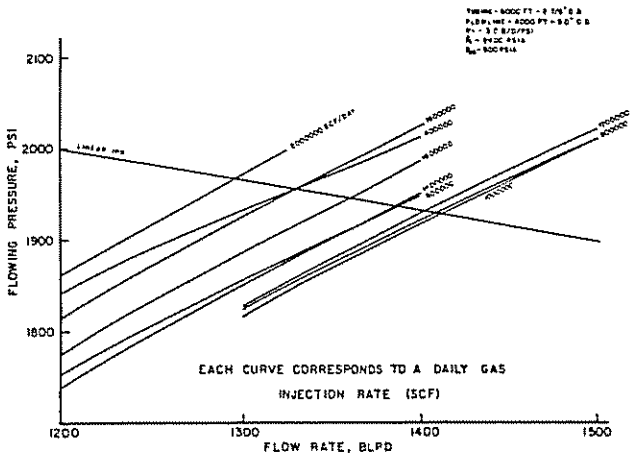


Fig. 3.422 Pressure Flow Rate Diagram (after Fonseca)

TABLE 3.419
RESULTS OF EXAMPLE PROBLEM #2 (FIG. 3.422)

Gas flow rate Scf/D	GLR	q_L	P_{wf}	Injection depth	P_{wh}
100,000	298	1024	2059	4553	112
200,000	369	1180	2007	4724	127
300,000	436	1273	1976	4833	140
400,000	501	1330	1957	4894	150
500,000	566	1366	1945	4937	160
600,000	633	1387	1938	4968	170
700,000	700	1400	1934	4973	178
800,000	768	1408	1931	4981	187
900,000	838	1411	1930	4983	195
1,000,000	909	1411	1930	4987	203
1,200,000	1056	1402	1933	4977	219
1,400,000	1211	1385	1939	4957	236
1,600,000	1376	1360	1947	4930	249
1,800,000	1554	1330	1957	4895	264
2,000,000	1742	1297	1968	4853	278

PI = 3
$P_{separator}$ = 80 psi
2½" tubing
2½" flowline
γ_g = 0.65
P_{so} = 900 psi

Find:
Flow Rate for a variable wellhead pressure in natural flow and gas lift.

(2) Given:
3" tubing - 9000 ft
3" flowline - 3000 ft
PI = 5 (constant)
G/O = 300 scf/b
50% water
γ_g = 0.7
Separator pressure - 100 psi
(Variable wellhead pressure)
$\overline{P}_R$ = 3000 psi
P_{so} = 1000 psi

Find:
flow rate possible by:
(a) Natural flow
(b) Gas Lift
(1) Gas unlimited
(2) Gas Volume = 1,000,000 scf/d
(3) Gas Volume = 400,000 scf/d

(3) Given:
2⅜" O.D. tubing
7000 ft depth
$\overline{P}_R$ = 2950 psi
GOR = 400 scf/b
γ_g = 0.7
P_{so} = 750 psi

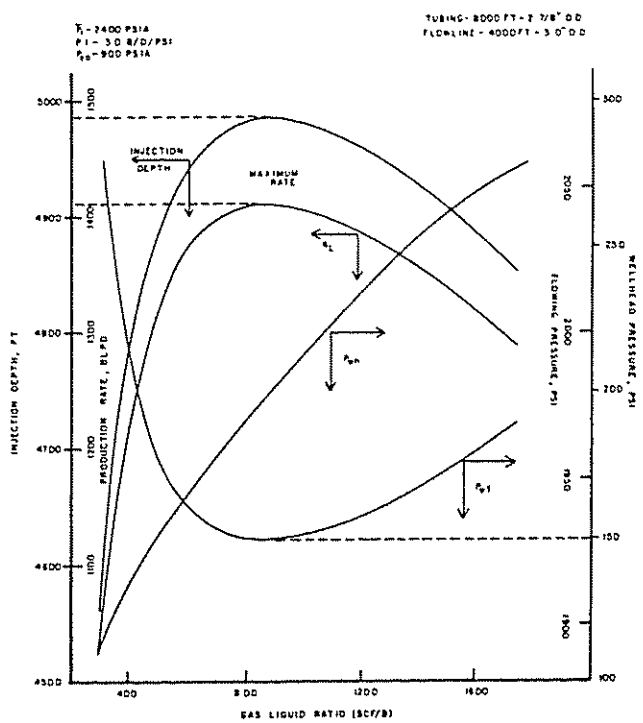


Fig. 3.423 Gas Lift Performance Curves (after Fonseca)

2" I.D. flowline
4000 ft in length
PI = 2
Separator pressure = 80 psi
(Variable wellhead pressure)

Find:
flow rate possible by:
(a) Natural flow
(b) Gas lift
(1) Unlimited gas available
(2) Gas volume = 250,000 scf/day
(c) Find (a) and (b) if PI = 8.
(d) Find (a) and (b) if we have 4000 ft of 4" I.D. Flowline.
(Use pressure flow-rate diagram and equilibrium curve procedures).

(4) Given:
Depth = 10,000 ft
$\overline{P}_R$ = 3000 psig
Tubing size = 2 in. nominal
Production = 90% salt water
GOR = 1000 scf/b
Separator pressure = 80 psig
Surface flowline = 3500 ft of 2-in. line
Gas volume = unlimited
γ_g = 0.70
Injection operating pressure = 900 psig
PI = 4

Find:
(a) The maximum production rate possible from this well, assuming no Christmas tree restrictions.
(b) The minimum size surface flowline to obtain maximum production from this well for a set 2-in. tubing size. Give rate and flowline size.

(5) Given:
Depth = 8000 ft
$\overline{P}_R$ = 2500 psi
Tubing size = 2 in.
Separator pressure = 60 psig
Injection operating pressure = 900 psig
95% water
γ_g = 0.65
GOR = 1000 scf/b
PI = 20
Surface flowline length and size = 3000 ft × 2 in.

Find:
the maximum production possible to gas lift from this well. An unlimited supply of gas is available.

(6) An initial completion is being decided upon for a 6000 ft well. This well is to produce into a 60 psig separator system and will have 3000 ft of surface flowline. It is expected to be a water drive reservoir with a maximum total liquid production of 700 B/D (10 percent oil).
Given:
$\overline{P}_R$ = 2000 psig.
Available gas pressure = 800 psi
Gas sp gr = 0.70
PI = 10
Kick-off pressure = 850 psi
GOR = 600 scf/b
Select both the vertical tubing size and the horizontal flowline size for this well.

(7) Given:
Depth = 10,000 ft
Tubing size = 2½ in.
Production = 95% salt water
GOR = 500 scf/b
PI = 50
Separator pressure = 80 psig
(2000 ft from wellhead)
Flowline size = 2½ in.
Injection gas pressure = 1200 psig
Static BHP = 2800 psig
Maximum gas available = 1,400,000 scf/D
γ_g = 0.8

Find:
The maximum fluid production rate.

3.424 An economic study of a continuous flow gas lift well

3.4241 Introduction

Regnault did a study whereby he showed the economic considerations in designing a gas lift installation.[9] This study took a present-day flowing well and determined the economic feasibility of placing it on gas lift. Many wells fall in this category, i.e., they are capable of producing by natural flow, but the rates may be increased by placing them on gas lift at the present time. This is particularly true for high productivity wells. To best illustrate this study, an example problem is worked.

3.4242 Example problem

The following information is given, including economic data:

Typical well data:
$\overline{P}_R$ = 2400 psi (all oil)
PI = 4 (original), but changes as follows: PI = 0.00143 $\overline{P}_R$
GOR = 400 scf/b
P_{wh} = Constant at 120 psi
P_{so} = 900 psi
γ_g = 0.65
Depth = 8000 ft
$\overline{P}_R$ decreases 100 psi per each 200,000 bbl of recovery

Bubble point pressure = 2300 psi below which Vogel's equation holds for two-phase flow in the reservoir

The following tubing and casing sizes are available:
Tubing: 2 in., 2½ in., 3 in., and 4 in. (nominal sizes)
Annular flow is also possible for 2 × 5½ in. and 2½ in × 7 in. sizes.

Economic Data:
Oil price U.S. $11.00/bbl (owner's share of gross)

Operating injection cost from central system	$ 0.05/Mscf
Production costs	$ 0.20/bbl
	$0.10/bbl
Maintenance cost	$0.15/bbl using a compressor
Pulling the well	$150,000.00
New equipment (valves)	$ 18,000.00
Other equipment	$ 5,000.00
Interest rate	10%
Royalty	15%
Local production tax	5%
Capacity cost if using a gas compressor	$ 73,000/MMSCF
Additional supervision cost if using a gas compressor	$ 1,000 initially $ 500/month

Two different tubing sizes will be worked, namely 2½" and 4". Assume sufficient gas is available to obtain maximum production rates in each case, and the gas is not optimized for maximum profit in this case.

Required:

PROJECT A (using 2½-in. tubing)

(1) Determine the maximum production rate by natural flow from this well.
(2) Determine the maximum production rate possible by gas-lift.
(3) Perform an economic analysis in placing the well on gas-lift using gas from a central system and a compressor.

PROJECT B (using 4-in. tubing):

(1) Determine the maximum production rate by natural flow from this well.
(2) Determine the maximum production rate possible by gas-lift.
(3) Perform an economic analysis in placing the well on gas-lift using gas from a central system and a compressor.

Solution Procedure: PROJECT A

The following solution procedure is given without the use of a compressor.

(1) Pressure-flow rate diagrams are constructed as in Figures 3.424 and 3.425, for both flowing and gas lift conditions. This procedure has been described previously.

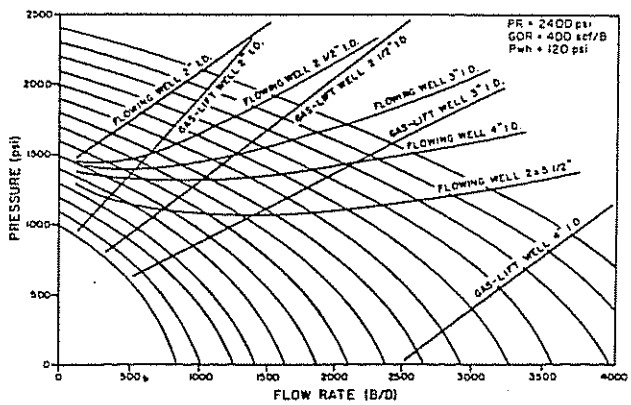


Fig. 3.424 Comparison of Flow Configuration Sizes

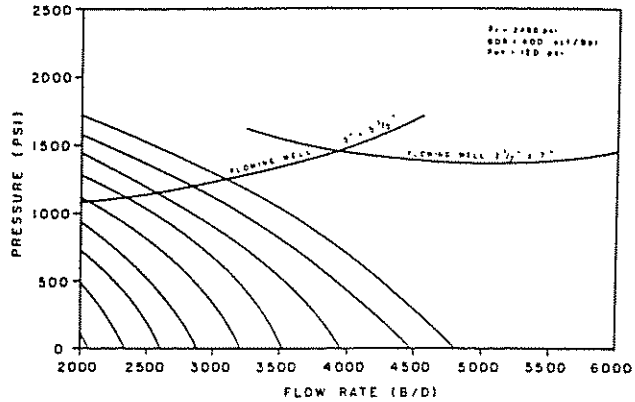


Fig. 3.425 Comparison of Flow Configuration Sizes

(2) Tables 3.420 and 3.421 are prepared showing flow rates possible for both natural flow and gas lift for 2½" tubing. Similar tables are prepared for 4" tubing. This information is obtained from Figure 3.424 and from given data stating that $\bar{P}_R$ decreases 100 psi for each 200,000 b recovery.
(3) Fig. 3.426 shows the cumulative production vs. time for both natural flow and gas lift.
(4) Table 3.422 shows the production from natural flow.

TABLE 3.420
FLOWING WELL PRODUCTION
PROJECT A (2.5" Tubing)

Time (month)	Average rate (B/D)	Cumulative production (bbl)
4.46	1,475	200,000
9.43	1,325	400,000
15.03	1,175	600,000
21.30	1,050	800,000
28.61	900	1,000,000
37.28	750	1,200,000
48.34	600	1,400,000
62.96	450	1,600,000
84.89	300	1,800,000

TABLE 3.421
GAS-LIFT PRODUCTION
PROJECT A (2.5" Tubing)

Time (month)	Average rate (B/D)	Cumulative production (bbl)
3.96	1,660	200,000
8.20	1,550	400,000
12.77	1,440	600,000
17.68	1,340	800,000
23.07	1,220	1,000,000
28.95	1,120	1,200,000
35.53	1,000	1,400,000
42.84	900	1,600,000
52.23	700	1,800,000
63.02	610	2,000,000
75.92	510	2,200,000

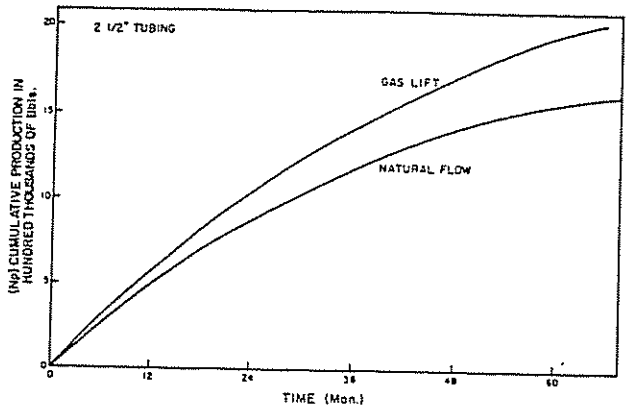


Fig. 3.426 Cumulative Production (Np) vs. Time—"Project A"

TABLE 3.422
FLOWING WELL—PROJECT A
(2½ in.)

Year	Avg. rate (B/D)
0-1	1450
1-2	1100
2-3	850
3-4	675
4-5	540

(5) Table 3.423 shows the production for gas lift—Project A (2½" tubing).
(6) Table 3.424 shows the increased production from natural flow to gas lift.
(7) A cumulative net cash flow vs. time curve is constructed in Fig. 3.427 for money at 10% from which a relatively short pay-out time of 4.8 months is found. (See Table 3.425)
(8) To look at other economic measures of value such as "rate of return" and net present value, Table 3.426 shows discounted cash flows. Net present worth values are in table 3.426.
(9) The data of Table 3.426 is plotted on Figure 3.428 from which the rate of return (ROR) is extrapolated to be 368% and extremely favorable project.

TABLE 3.423
GAS LIFT WELL—PROJECT A
(2½ in.)

Avg. rate (B/D)	Injection gas (MMscf/yr)
1600	666
1320	622
1080	578
880	538
700	490

TABLE 3.424
INCREASED PRODUCTION PROJECT A

Avg. rate (B/D)	Yearly increase (B)
150	54750
220	80300
230	83950
205	74825
160	58400

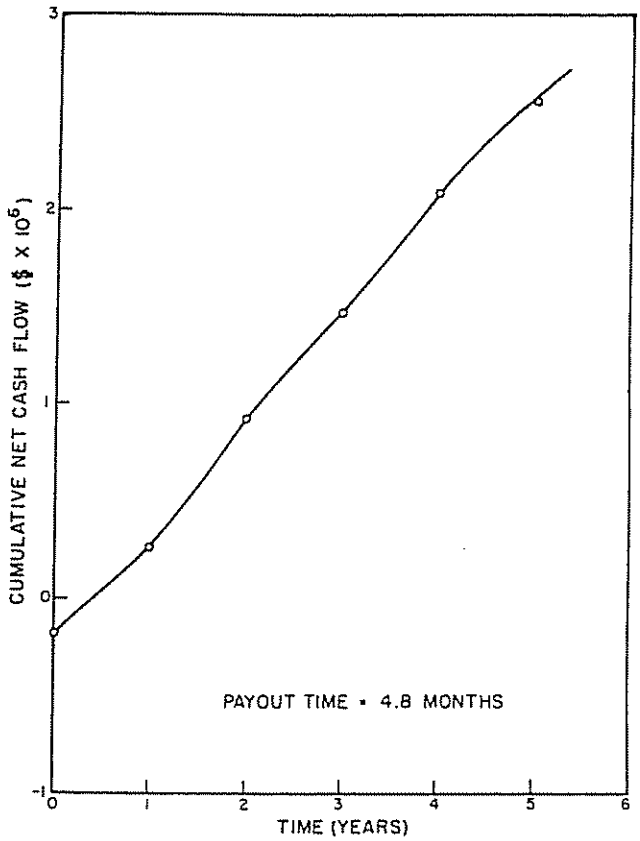


Fig. 3.427 Cumulative Net Cash Flow vs. Time—"Project A"

The solution procedure is the same as for a compressor, and a pay-out time of nine months is found with a rate-of-return of 330% for an initial investment of $306,200.

Solution: PROJECT B (4" tubing)

Since the solution procedure is the same, only a few of the pertinent figures and tables will be shown.

Tables 3.427 and 3.428 show the comparative rates and cumulative production by natural flow and gas lift for 4" tubing.

Figure 3.429 shows cumulative production vs. time for Project B for both natural flow and gas lift.

Without a compressor the pay-out time is only one month with an ROR that is greater than 400%.

If a compressor is used, the pay-out time is two months and the ROR is also greater than 400%.

Summary:

Table 3.429 summarizes the economic study of this well for 2½" and 4" tubing sizes. Other flow configurations would be studied in the same manner.

Obviously, this represents a very profitable project and its immediate initiation would be recommended.

Only three "measures of value" are studied, namely (1) pay-out, (2) rate-of-return, and (3) net present value.

TABLE 3.425
PRODUCTION AND PROJECTED NET CASH FLOW FOR PROJECT A (2½" TUBING)

Year	Injection gas (MMscf/yr)	Yearly increase of prod. (B)	Annual gross revenue (@$11/bbl)	Opeating injection gas cost (@$0.05/Mscf)	Production cost (@$0.20/bbl)	Maintenance cost (@$0.10/bbl)	Annual net revenue*	Future expenditure	Net cash flow	Cum. Net Cash Flow
0	—	—	—	—	—	—	—	—	−173,000	−173,000
1	666	54750	602,250	33,300	10,950	5,475	436,592	—	436,592	263,592
2	622	80300	883,300	31,100	16,060	8,030	658,075	—	658,075	921,667
3	578	83950	923,450	28,900	16,790	8,395	691,600	150,000	541,600	1,463,267
4	538	74825	823,075	26,900	14,965	7,482	615,286	—	615,286	2,078,553
5	490	58400	642,400	24,500	11,680	5,840	476,718	—	476,718	2,555,271

*Annual net revenue = annual gross revenue (1-0.15) (1-0.05) − Operating injection cost − Production cost − Maintenance cost

Royalty 15% = 0.15
Local production tax 5% = 0.05

TABLE 3.426
NET PRESENT WORTH FOR PROJECT A

Year	Net Cash Flow	Discount Factor (mid-year) 10%	20%	50%	100%	200%	Discounted Cash Flow 10%	20%	50%	100%	200%
0	−173,000	1.00000	1.00000	1.00000	1.00000	1.00000	−173,000	−173,010	−173,000	−173,000	−173,000
1	436,592	0.95346	0.91287	0.81650	0.70711	0.57735	416,273	398,552	356,477	308,718	252,066
2	658,075	0.86678	0.76073	0.54433	0.35355	0.19245	570,406	500,617	358,210	232,662	126,646
3	541,600	0.78799	0.63394	0.36289	0.17673	0.06415	426,775	343,342	196,541	95,717	34,744
4	615,286	0.71635	0.52828	0.24192	0.08839	0.02138	440,760	325,043	148,850	54,385	13,155
5	476,718	0.65123	0.44023	0.16128	0.04419	0.00713	310,453	209,866	76,885	21,066	3,399
				Net present worth:			1,991,667	1,604,420	963,963	539,548	257,010

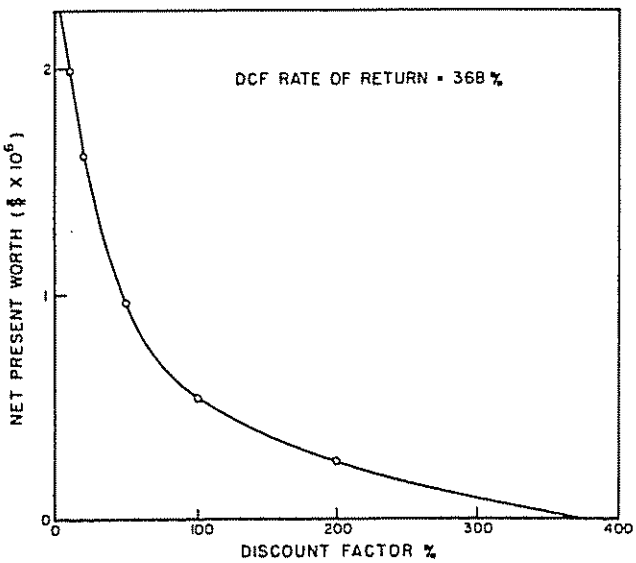


Fig. 3.428 Net Present Worth vs. Discount Factor—"Project A"

There are certainly sufficient to initiate the project. Refer to Volume 1, Chapter 3, of this series for a review on economics.

In this section, no attempt has been made to optimize the gas volume for maximum profit. The goal was to find maximum rate and to show the economic return of placing this well on gas lift. Optimization procedures will be discussed in Section 3.426.

TABLE 3.427
FLOWING WELL PRODUCTION
PROJECT B (" tubing)

Time (Month)	Avg. Rate (B/D)	Np (Bbl × 10⁵)
2.70	2450	2
6.00	2200	4
10.30	1925	6
15.82	1663	8
23.08	1425	10
33.60	1175	12
51.20	900	14
77.97	675	16
118.42	500	18

TABLE 3.428
GAS-LIFT PRODUCTION
PROJECT B (4" tubing)

Time (Month)	Avg. Rate (B/D)	Np (Bbl × 10⁵)
1.81	3625	2
3.80	3462	4
6.00	3300	6
8.50	3100	8
11.34	2900	10
15.80	2500	12

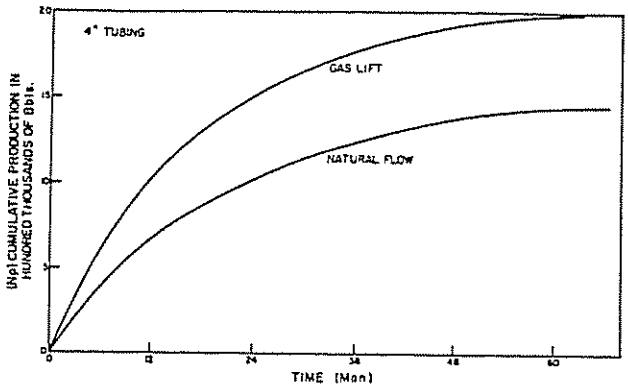


Fig. 3.429 Cumulative Production (Np) vs. Time—Flowing Well Project B (4" Tubing)

TABLE 3.429
RESULTS OF PROBLEM NO. 2

Production analysis	Project A (2½")	Project B (4")
Maximum production rate by natural flow	1550 B/D	2575 B/D
Maximum production rate possible by gas lift	1725 B/D	3700 B/D
Economic analysis		
(a) Using a central system		
@ 10% Discount rate payout	0.40	0.08 year
(NPV) Net present value	$1,991,667	
(ROR) Rate of return	368%	>400%
(b) Using a compressor		
@ 10% Discount rate payout	0.75 year	0.17 year
(ROR) Rate of return	330%	>400%

3.425 Design of continuous flow gas lift installations based on the most economical volume of gas to be injected, by Victor Mitchell and Jesus Pacheco

3.4251 Introduction

The most economical total gas-liquid ratio exists when the profit obtained from lifting a quantity of oil by a unit increment of gas is just equal to the cost of that unit of gas. The profit per barrel of oil varies from place to place, and the operator must evaluate this figure. It is essentially the selling price of oil less operating costs. The cost of gas also varies from installation to installation and depends on such variables as cost of makeup gas, cost of fuel for compressors, size and efficiency of compressors, differential pressure across compressors, etc. and must be determined by the operator

3.4252 Determination of the most economical total gas-liquid ratio for constant well-head pressure

Mitchell made the following study to show how to obtain the most economical gas-liquid ratio for continuous flow gas lift for certain given conditions.[10] His solution is for an assumed constant wellhead pressure.

Consider the well and the associated pressure vs. TGLR plot in Fig. 3.430. The pressure vs. TGLR curve for the depth of injection becomes the pressure in the tubing at the injection point $(P_{tub})_{inj}$ vs. TGLR curve.

Assume the most economical TGLR occurs at a point on the $(P_{tub})_{inj}$ vs. TGLR curve where the slope is X psi/TGLR (psi bbl/scf),

therefore:

$$\frac{(d\,P_{tub})_{inj}}{d\,TGLR} = X \tag{3.414}$$

The flow rate of this well is q B/D.

The productivity index of this well at a flow rate of q B/D is defined as PI (B/D/psi). This can be represented as the rate of production of liquids from the reservoir into the tubing per psi drop of flowing bottom hole pressure in the tubing ($-dq/dP_{wf}$).

This solution assumes that the decrease in tubing pressure at the injection point caused by a unit increase in TGLR is approximately equal to the net decrease in flowing bottom hole pressure at the sandface depth, that is,

$$d(P_{tub})_{inj} = dp_{wf}$$

For most gas lift wells this will be true. For normal PI's a small decrease in bottom hole pressure will result in a small increase in inflow (dq) of fluids into the well.

therefore:

$$\frac{dq}{dTGLR} = \left(\frac{d(P_{tub})_{inj}}{dTGLR}\right)\left(\frac{dq}{dp_{wf}}\right) = X(PI) \tag{3.415}$$

At the most economical injection rate the cost of a unit increase in TGLR is just equal to the profit obtained from the resultant production of oil.

The gas injected per unit increase of TGLR is:

$$q(B/D) \times 1(scf/bbl)$$

The oil lifted per unit TGLR injected is:

$$(PI \cdot X)$$

Hence, if gas costs are G $/scf and the profit/bbl of oil is P $/B, then at the economical point:

$$\$(q\ G) = \$(PI \cdot X \cdot P)$$
$$\text{and } X = (qG)/(PI \cdot P) \tag{3.416}$$

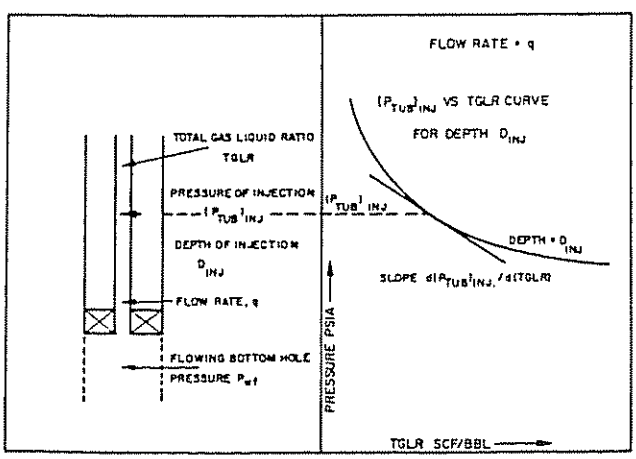


Fig. 3.430 Relationship Between The Flowing Well and The Pressure vs. TGLR Curve

If a tangent of the slope X is fitted to the $(P_{tub})_{inj}$ vs. TGLR curve, the point of tangency defines the most economical TGLR. The tangent of zero slope intersecting this curve defines the maximum TGLR and corresponds with the minimum gradient TGLR.

For a given well the pressure vs. TGLR curves may be drawn for varying values of depth and the economical TGLR determined at each depth. A curve joining these economical TGLR points defines the most economical TGLR curve as shown in Figures 3.431, 3.432 and 3.433 for different rates.

A short-cut method is also presented to determine the economical TGLR based on the maximum TGLR curve and is given in Section 3.4254.

3.42521 Procedure for the most economical gas lift design

The principal diagram used in this design is the vertical multiphase flow plot of pressure vs. TGLR for different depths. This can either be transferred from existing gradient curves or calculated from a suitable multiphase flow correlation. One figure is necessary for each rate as noted in Figures 3.431, 3.432 and 3.433 for rates of 1500, 2000, and 2500 B/D.

Figure 3.434 is a cross plot used to prepare the most economical gas lift design. This comprises four plots side by side. These plots are best made on transparent graph paper for tracing gradient curves if desired.

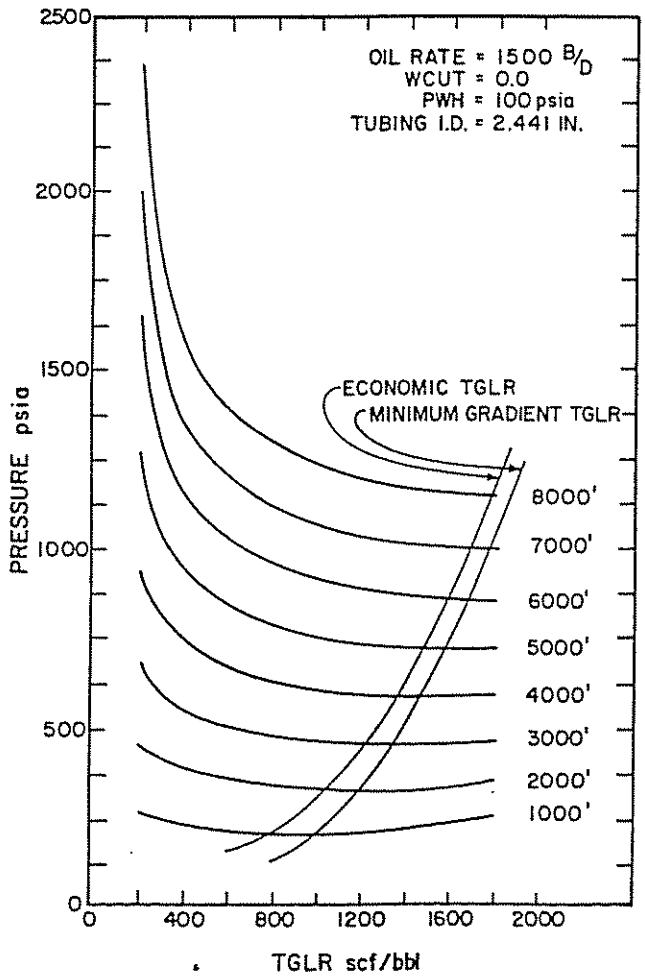


Fig. 3.431 Pressure vs. TGLR Curves for 1500 BBL/D

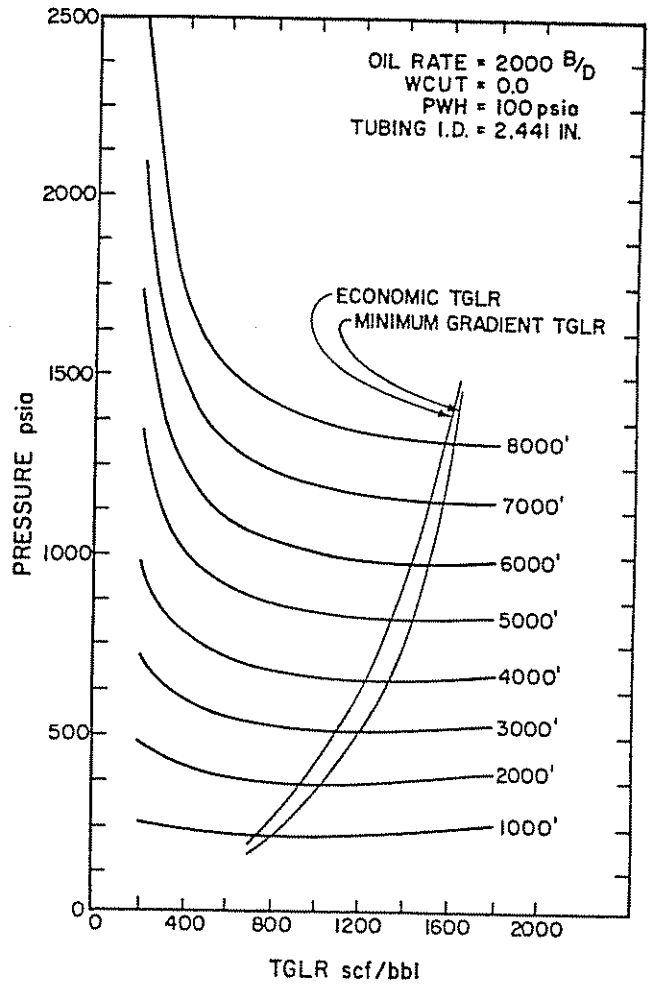


Fig. 3.432 Pressure vs. TGLR Curves for 2000 BBL/D

These plots are:

Fig. 3.434 (A)—A pressure vs. depth plot which shows gas column weight effects and determines the bottom hole pressure by means of extending the flowing gradient curve to well depth after the point of injection is obtained.

Fig. 3.434 (B)—Pressure vs. TGLR plot which shows the interrelation of flow rate, depth, and the most economical total gas-liquid ratio.

Fig. 3.434 (C)—A pressure vs. flow rate plot where the bottom hole pressure determined in plots 3.434 (A) and 3.434 (B) is cross plotted against inflow performance.

Fig. 3.434 (D)—A pressure vs. flow rate plot where the surface operating pressures are plotted vs. flow rate. These values are obtained from first obtaining P_{wf} from Figure 3.434 (C) and then determining the corresponding values of P_{wh}.

The following procedure applies to the most economical design method for different values of surface operating pressures. It permits the selection of the most economical gas volumes to be used for set design parameters.

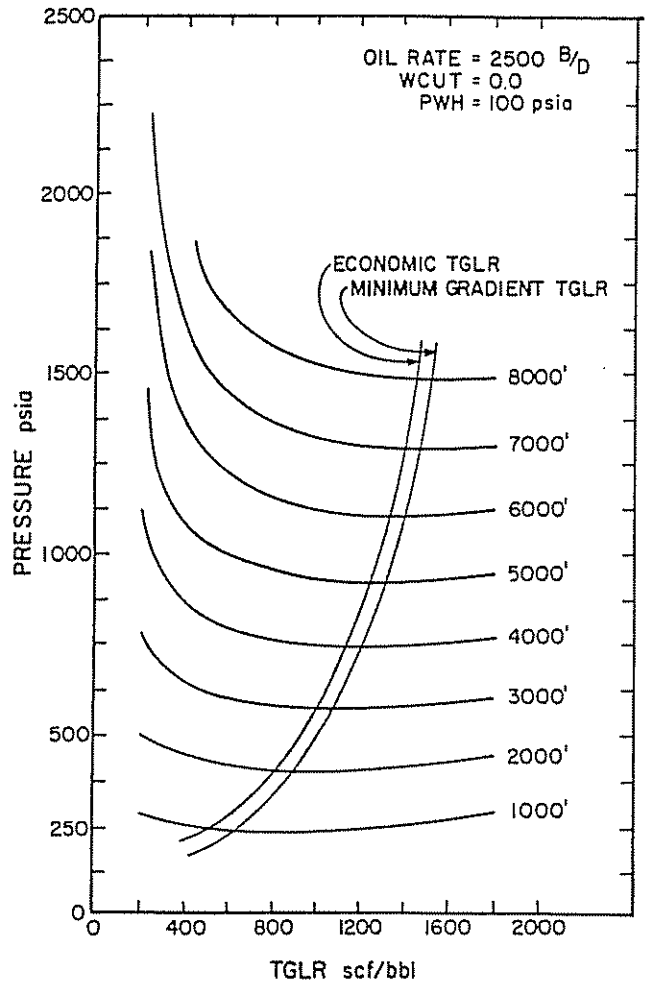


Fig. 3.433 Pressure vs. TGLR Curves for 2500 BBL/D

(1) Prepare the multiphase flow plot of pressure vs. TGLR for different depths and determine the economical and maximum TGLR as follows:
  (a) Assume various flow rates.
  (b) Plot the vertical multiphase flow data with pressure on the ordinate and TGLR on the abscissa for various depths, and different assumed rates (Figs. 3.434 A, B, and C).
  (c) Determine the minimum gradients, i.e. when the slopes of the pressure vs. TGLR curves are zero. These points are then connected as the maximum TGLR curves. This line can be calculated or constructed by observation.
  (d) Calculate the economical slope (qG/PI/P) and determine the economical TGLR curve on the pressure vs. TGLR plot. This curve is obtained by joining the most economical TGLR points determined at each depth. The most economical TGLR is calculated as previously described.

(2) (a) With pressure on the ordinate and depth on the abscissa, plot the surface operating pressure (P_{so}) at zero depth and extend down the well. [Fig. 3.434(A)] The effect of gas column weight must be included. A line parallel to the above is drawn beginning at a value of (P_{so} – 100 psi) at zero depth and extending to the well depth. This is the gas operating line. This 100 psi difference accounts for the pressure drop accurring during flow through the gas lift valve and represents the tubing pressure available at any injection point throughout the well.

  (b) With pressure on the ordinate and TGLR on the abscissa, trace onto Figure 3.434(B) the most economical TGLR curves for different rates obtained in Step 1 and plotted in Figures 3.434 A, B and C. Mark off depths beginning at 1000 ft in units of 1000 ft up to the total well depth for each flow rate assumed. Draw a smooth curve between points of equal depth. Since the pressure scale on the pressure vs. depth and the pressure vs. TGLR curves [Figs. 3.434(A) and 3.434(B)] are the same, a straight-edge laid across the two curves gives the injection depth and injection pressure where the depths on each figure correspond.

  With the injection point located on the gas operating line, [Fig. 3.434(A)], the pressure gradients below the points of injection are determined from the multiphase flow plot at the producing formation gas liquid ratio and assumed flow rates. The flowing bottom hole pressures can then be determined at the chosen flow rates by tracing the gradient curve at the formation GLR from the point of injection to the well depth. The flowing bottom hole pressures determined in this step are the bottom hole pressures which are required for the assumed rates.

  Tabulate the results of surface operating pressure, flow rate, pressure of injection, depth of injection, TGLR and injection lift gas.

(3) (a) Prepare the pressure vs. flow rate plot by plotting pressure on the ordinate and flow rate on the abscissa [Fig. 3.434(C)]. Draw in the IPR curve, (flowing bottom hole pressure vs. flow rate). Either of the many ways of handling IPR may be considered, including Vogel, Standing, Weller and Fetkovitch, or constant PI.

  (b) The bottom hole pressure and flow rate data obtained from Step 2(b) in Figure 3.434(C) for a single surface operating pressure are plotted. The intersection of the curves from Step 3(a) and (b) represents the most economical oil flow rate for that surface operating pressure.

  (c) Other values of surface operating pressure may be assumed and Steps 2(a) to 3(b) repeated, giving economical flow rates for each assumed surface operating pressure. Flow rate vs. surface operating pressure is next plotted, giving the relationship between surface operating pressure and flow rate for the gas lift system while maintaining the most economical gas liquid ratio [see Fig. 3.434(D)].

3.42522 Short-cut method for determining the most economical gas-liquid ratio

Determining the most economical gas-liquid ratio by fitting a calculated slope to the pressure vs. TGLR multiphase flow curve as explained in the previous section

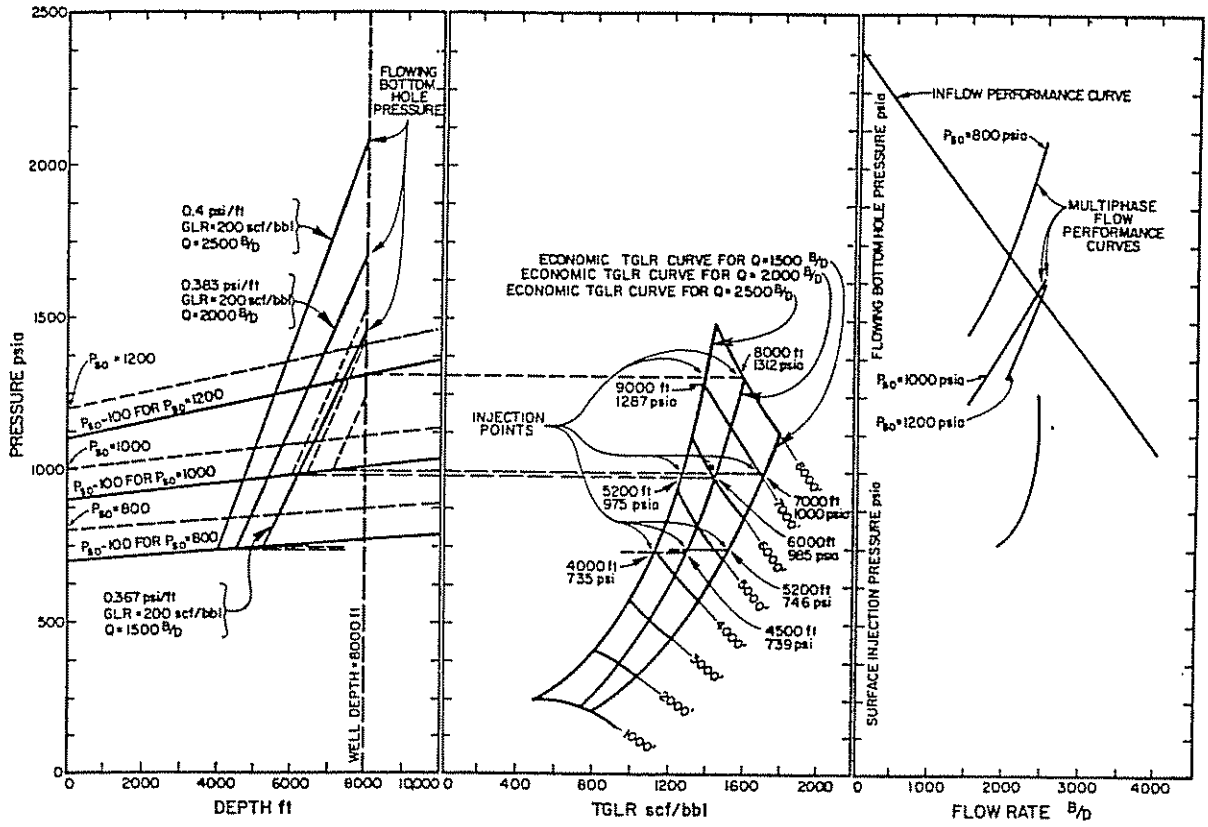


Fig. 3.434 Overall Cross Plot for Most Economical Gas Lift Design

may be difficult and time-consuming in some cases. As a result a fairly accurate, simplified method has been devised to determine the most economical total gas-liquid ratio.

The pressure vs. TGLR curve for a given depth shown in Figure 3.435 indicates that beginning with zero TGLR the pressure decreases with increasing TGLR up to a minimum gradient and then begins to increase with increasing TGLR. This minimum gradient corresponds to a minimum gradient TGLR and occurs where any increase in TGLR at that depth results in an increase in multiphase flow pressure drop and a decrease in well production. This point is noted on the pressure vs. TGLR curve as the point of zero slope and can readily be observed.

Observation of the pressure vs. TGLR curves for a given flow rate and pipe size (Fig. 3.435) shows that the minimum gradient occurs at a different value of TGLR for different depths but follows a smooth curve with the minimum gradient TGLR increasing with increasing depth. The economical TGLR point which occurs at the calculated slope also follows a smooth curve with the economical TGLR increasing with increasing depth. The economical TGLR is always less than the minimum gradient TGLR.

A computer program can be written which generates the pressure vs. TGLR curves for different flow rates and pipe sizes and interpolates these curves for specified values of slope. The difference in TGLR required to move from the minimum gradient TGLR to the economical TGLR is then calculated.

Table 3.430 shows typical results for 3½" pipe and a flow rate of 3500 B/D.

The economical TGLR for a given injection depth occurs on the pressure vs. TGLR curve where the slope is equal to the value (q G/PI/P).

The results of Table 3.430 indicate that the difference between the minimum gradient TGLR and the economical TGLR is different for each depth. For example, at 0.01 slope the difference in TGLR varies from 104 scf/bbl at 1000 feet to 57 scf/bbl at 10,000 feet; at .05 slope, the difference in TGLR is 353 scf/bbl at 1000 feet compared to 278 scf/bbl at 10,000 feet. The differences in TGLR between the minimum gradient and the most economical TGLR's change faster for changing values of slope than for changing values of depth. Therefore, a single mean value of the TGLR difference should be used for all depths, corrected to the nearest 50 scf/bbl. The curve connecting the most economical TGLR's is always parallel to the curve connecting the minimum gradient TGLR's.

Using this approximation of parallel slopes, Table 3.431 was prepared which gives the difference between the most economical and the minimum gradient TGLR curves for different values of slope.

Use this short-cut method for calculating the economical TGLR:

EXAMPLE PROBLEM #12

Tubing diameter = 2½" in.
P_{wh} = 100 psia
Flow rate = 2000 B/D
Water-cut = 0
PI = 1.0 B/D psi
Profit on oil = $4.00/bbl
Gas cost = $.2/mcf
Depth of injection = 5000 ft

Determine the most economical TGLR for this well.

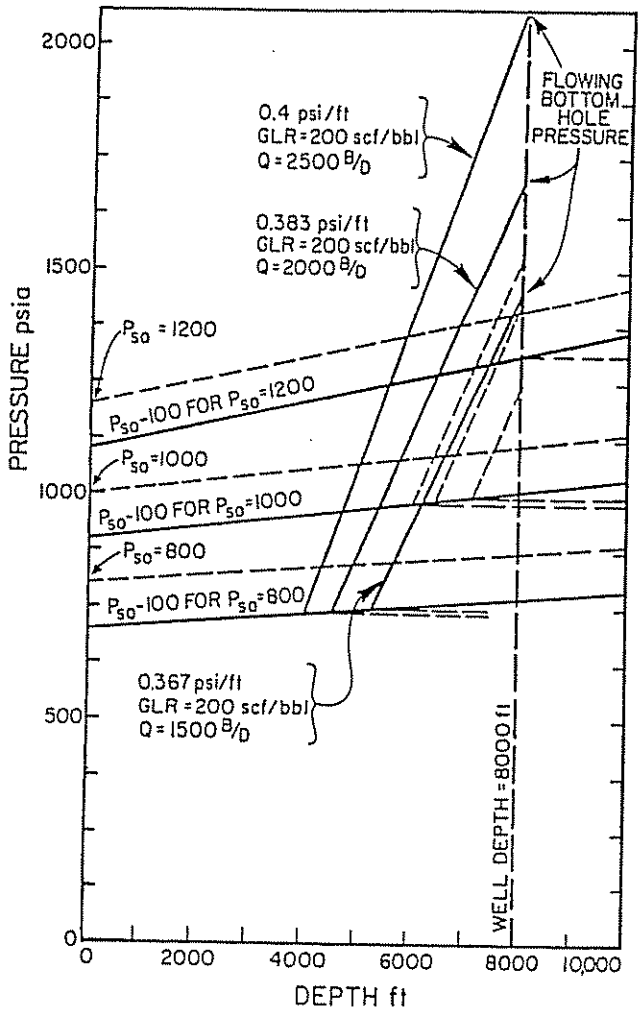


Fig. 3.434-A Pressure vs. Depth Portion of Economic Cross Plot

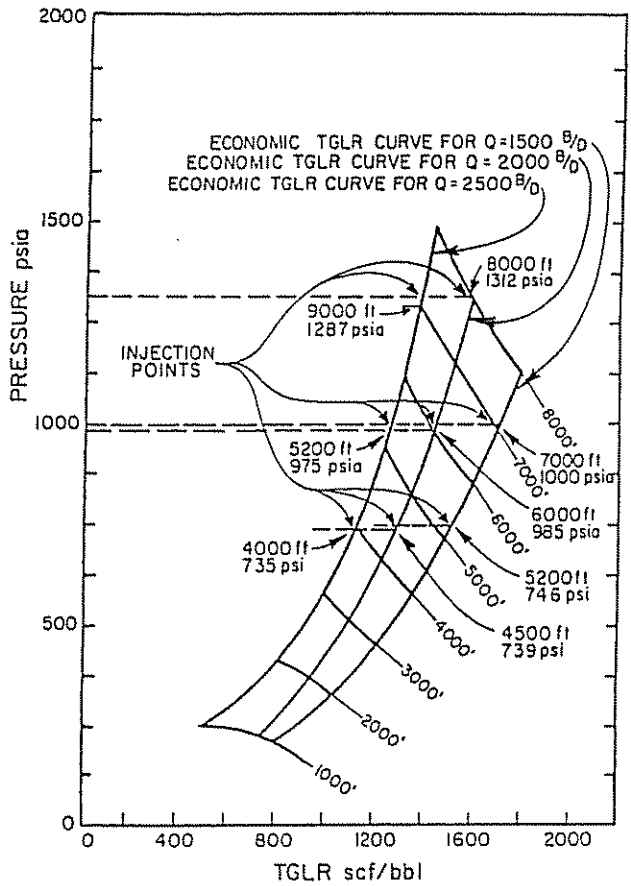


Fig. 3.434-B Pressure vs. TGLR Portion of Economic Cross Plot

Solution:

Draw the pressure vs. TGLR curve shown in Figure 3.435 and determine the points of zero slope for a depth of 5000 ft to be approximately 1450 scf/bbl. Determine the most economical slope to be:

$$\text{Slope} = q \times G/PI/P$$
$$= 2000 \times \frac{0.2}{1000} \times \frac{1}{1} \times \frac{1}{4}$$
$$= 0.1$$

From Table 3.431 the difference in TGLR between the economical slope and the minimum gradient TGLR's is 200 scf/bbl. Hence, the economical TGLR occurs at 1450 − 200 = 1250 scf/B.

3.42523 Example problem to illustrate the most economical gas lift design

Given:

Tubing I.D. (in.) = 2.441
Depth = 8000 ft
Static bottom hole pressure = 2400 psi
P_{wh} = 100 psia
PI = 3 B/D/psi (linear)
Oil API gravity = 30°
R_s = 200 scf/bbl
P_{so} = 800, 1000 and 1200 psi
Profit/bbl oil = $6.00
Gas cost/mcf = $.1

Determine the flow rates possible from this well at the most economical TGLR for the given values of surface operating pressure.

Solution:

(1) (a) Assume oil flow rates of 1500, 2000, 2500 B/D and plot the multiphase flow data in the form of pressure vs. TGLR curves for depth increments of 1000 ft. to the well depth of 8000 ft. Refer to Figs. 3.431, 3.432 and 3.433. Mark off the points of zero slope for the curves plotted in Figs. 3.431, 3.432 and 3.433. These points are determined by observation, in this case.

(b) Drawn the minimum gradient curve for each assumed flow rate by joining the points of zero slope. The economical TGLR's are calculated as follows:

Cost of gas = $.1/mscf
Profit on oil = $6.00/bbl

As previously described, the economical slope

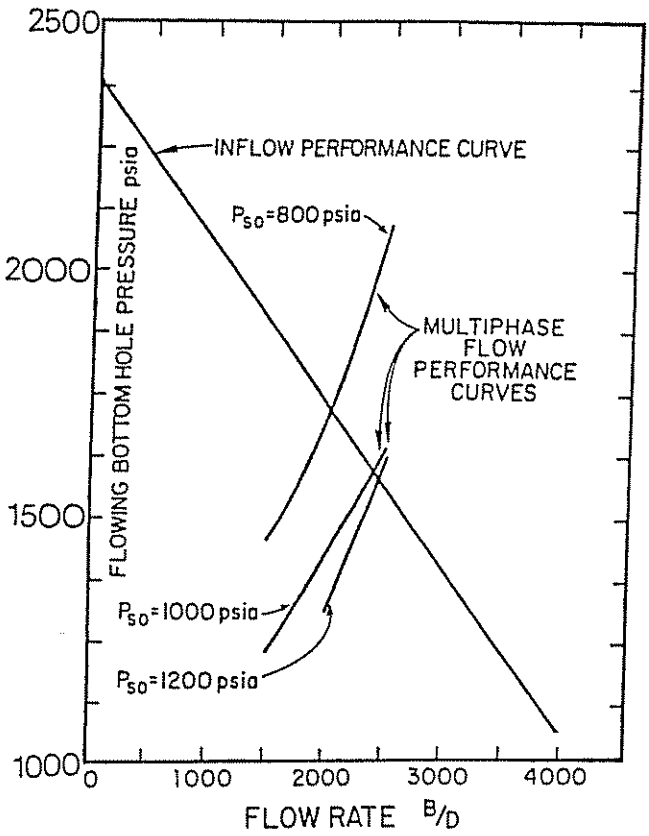


Fig. 3.434-C Bottom Hole Pressure vs. Flow Rate Portion of Economic Cross Plot

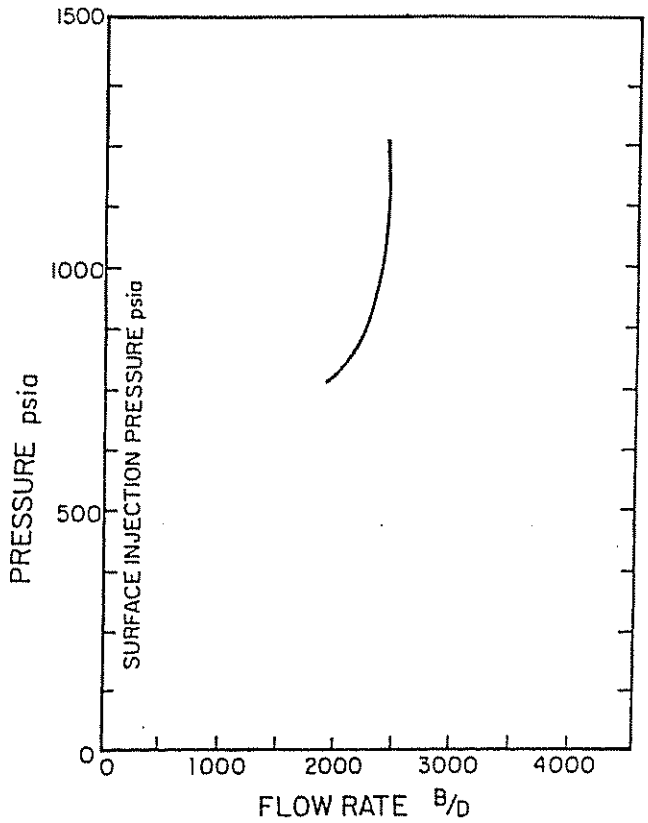


Fig. 3.434-D Surface Operating Pressure vs. Flow Rate Portion of Economic Cross Plot

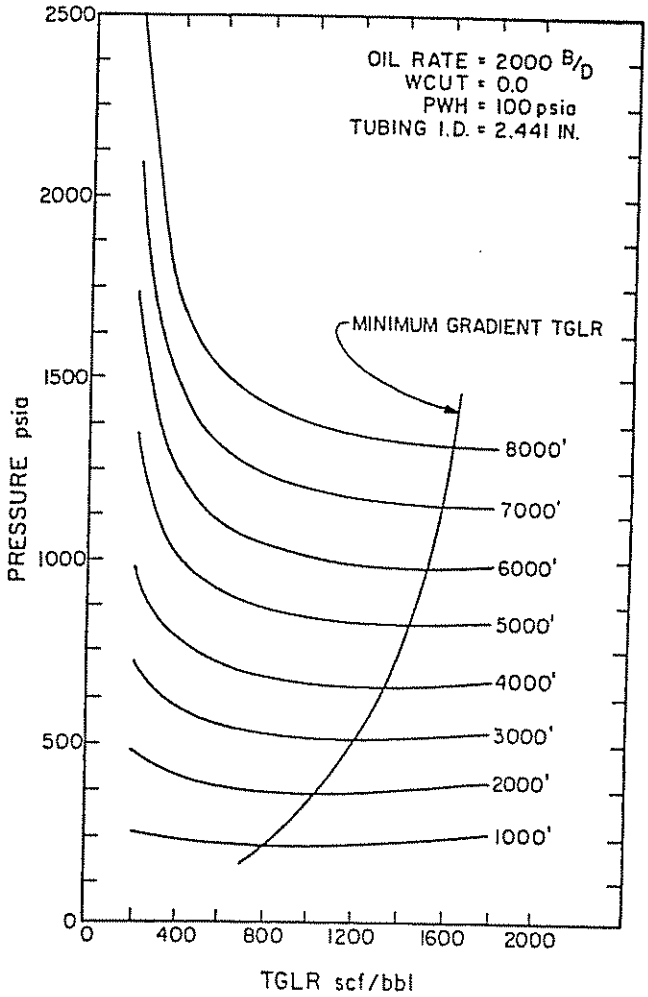


Fig. 3.435 Pressure vs. TGLR Curves for 2000 BBL/D

TABLE 3.430
VARIATION OF DIFFERENCE BETWEEN MINIMUM GRADIENT TGLR AND MOST ECONOMICAL TGLR WITH DEPTH

Depth of injection (ft)	Max GLR (scf/bbl)	ΔTGLR = Minimum gradient TLGR (scf/bbl) − Economical slope TGLR (scf/bbl) for slopes .01	.05	.1	.15	.2	.25
1000	873	104	353	520	594	624	654
2000	1100	100	360	510	634	750	805
3000	1258	91	340	537	633	717	793
4000	1415	87	337	551	687	771	842
5000	1518	67	329	538	675	768	853
6000	1595	56	281	502	682	761	832
7000	1688	79	277	510	665	800	558
8000	1760	69	276	490	651	788	882
9000	1818	63	276	469	643	763	884
10000	1867	57	278	453	630	747	848

TABLE 3.431
DETERMINING ECONOMICAL TGLR FROM MINIMUM GRADIENT TGLR

		Minimum gradient TGLR (scf/bbl) — economical slopes TGLR (scf/bbl) for slopes					
I.D. (in.)	rate (B/D)	.01	.05	.1	.15	.2	.25
1.995	500	200	600	850	950	1150	1200
	100	100	200	350	450	500	600
	1500	50	150	250	300	350	400
	2000	50	50	200	250	300	300
	2500	50	50	100	150	200	200
	3000	50	50	100	100	150	150
2.441	1000	200	700	950	1150	1250	1350
	1500	100	400	600	750	850	950
	2000	100	300	450	550	600	700
	2500	50	200	350	450	550	600
	3000	50	200	250	300	400	450
	3500	50	150	200	250	300	300
	4000	50	100	150	200	250	300
	5000	50	100	100	150	200	200
2.952	1500	400	1100	1350	1450	1700	1900
	2000	250	750	1000	1150	1250	1350
	2500	200	500	750	950	1050	1100
	3000	100	400	600	750	850	950
	3500	100	350	500	600	700	800
	4000	100	300	450	550	650	700
	5000	50	200	300	400	500	600
	6000	50	100	200	300	350	400
3.548	2500	600	1500	1800	2100	2200	2400
	3000	350	1300	1500	1600	1700	1800
	3500	300	850	1100	1200	1300	1400
	4000	200	700	900	1050	1150	1200
	5000	100	500	700	800	900	1000
	6000	100	400	550	650	700	800
	7000	100	300	450	550	650	700
	8000	50	250	350	450	500	600
3.958	2500	1100	2500	2800	2900	3000	3100
	3000	1000	1900	2300	2500	2700	2700
	3500	1000	1900	2300	2500	2700	2700
	4000	450	1100	1350	1600	1700	1900
	5000	200	700	1050	1150	1250	1400
	6000	200	600	850	950	1000	1100
	7000	100	450	650	750	850	950
	8000	100	350	550	650	750	800
	9000	100	300	450	550	650	700

$$\chi = qG/PI/P$$

$$\text{substituting } \chi = \frac{1500 \times .1}{3 \times 6 \times 1000}$$

$$\chi = .008 \text{ psi/TGLR}$$

The economical slopes for flow rates of 2000 and 2500 B/D are calculated in a similar manner and are 0.01 and 0.012 psi/TGLR respectively.

Tangents of slopes equal to the economical slope values determined above are drawn to pressure vs. TGLR curves for each depth and corresponding flow rates. This is accomplished with the aid of triangles. A smooth curve is drawn through these points. Table 3.431 (short-cut method) gives an economical TGLR of 50 scf/bbl less than the minimum gradient TGLR for a flow rate of 2500 B/D and a slope of .01. This economical TGLR curve gives a relationship between pressure, economical TGLR, and depth. For example, at a rate of 2500 B/D, and assuming a point in the tubing where the pressure is 500 psia, the most economical TGLR required is 920 scf/bbl at a depth of injection of 2500 ft. This means that the gas pressure at the gas lift valve outlet to the tubing is 500 psia, and at the inlet (casing) is 600 psia. For an injection depth of 5000 ft a TGLR of 1230 scf/bbl is needed at a tubing pressure of 925 psia or a casing gas pressure of 1025 psia. Until this point, the inflow properties of the well have not been considered and the well's ability to produce fluids (inflow performance) may not match the multiphase flow performance considered up to this point. This is covered in the following steps.

(2) (a) On the pressure vs. depth graph of the cross plot (Fig. 3.434A), plot the surface operation pressures (1200, 1000, 800 psia) at zero depth. Draw in the gas static gradients down the well. The gas gradients are 26.1, 23.6 and 19 psi/1000 ft, respectively, for values of P_{so} of 1200, 1000, 800 psia. Plot gas pressures of 1409, 1189, and 952 psia at 8000 ft and connect them by dashed lines to 1200, 1000, and 800 psia, respectively. Draw in the gas operating lines (P_{so} – 100) by plotting (P_{so} – 100) psia at zero depth and extending them them down the well parallel to the above dashed lines. These are solid lines in Figure 3.434 A.

(b) On the pressure vs. TGLR portion (Fig. 3.434 B) of the cross plot, trace the economical TGLR curves obtained from step 1 and plotted in Figures 3.431, 3.432 and 3.433 for each flow rate assumed. Be sure that the pressure and TGLR scales match. Connect the points of the same depth. With a straightedge laid horizontally across Figures 3.434 A and B —i.e., keeping the same pressure between the gas operating line in Fig. 3.434 A and 3.434 B—move vertically until the depth in Fig. 3.434 A corresponds with the depth in Fig. 3.434 B. This point marks the injection depth and pressure required so the multiphase flow conditions occurring uphole from the injection point are satisfied at the economical TGLR.

(c) Determine the pressure gradient below the point of injection from the multiphase flow curves prepared in Step 1. This gradient is determined from the pressure vs. TGLR curves (Figs. 3.431, 3.432, and 3.433). For a gas-liquid ratio equal to the formation gas-liquid ratio (200 scf/bbl) and a flow rate of 1500 B/D (Fig. 3.431), calculate the pressure at 8000 ft. to be 2350 psia and the pressure at the approximate injection depth of 5000 ft to be 1250 psia. The pressure gradient below the point of injection for 1500 B/D is therefore 0.367 psi/ft. The gradients for 2000 and 2500 B/D are determined in the same manner to be 0.383 and 0.400 psi/ft, respectively. If more accuracy is required, a pressure depth curve for the flowing gas-liquid

ratio can be calculated by computer or traced from gradient curves beginning at the injection point and extending to the sandface depth. The point of intersection of the flowing gradient curve and the well depth line give the value of flowing bottom hole pressure required to satisfy the requirement of multiphase flow at the economical TGLR. This procedure is repeated for each assumed flow rate and injection pressure. Tabulate the flow rates, surface operating pressure, depth of injection, pressure of injection, and TGLR as in Table 3.432.

TABLE 3.432
RESULTS FROM ECONOMICAL CROSS PLOT

FGLR	$\bar{P}_r$	P_{wh}	P_{so}	q	D_{inj}	$(P_{tub})_{inj}$	P_{wf}	TGLR
200	2400	100	800	1500	5200	746	1456	1520
200	2400	100	800	2000	4500	739	1705	1300
200	2400	100	800	2500	4000	735	2087	1120
200	2400	100	1000	1500	7000	1000	1230	1700
200	2400	100	1000	2000	6000	985	1430	1440
200	2400	100	1000	2500	5200	975	1655	1250
200	2400	100	1200	1500	-	-	-	-
200	2400	100	1200	2000	8000	1312	1312	1600
200	2400	100	1200	2500	7000	1287	1625	1380

(3) (a) On the pressure vs. flow rate diagram (Fig. 3.434 C), plot the inflow performance, i.e. flowing bottom hole pressure vs. rate. Plot 2400 psia at zero flow rate. At 1000 B/D and for a PI of 3, the drawdown is 333 psia. Plot 2400 − 333 = 2067 psia at 1000 B/D. Join these two points with a straight line and extend to a flow rate of 4000 B/D.

(b) Plot the values of P_{wf} vs. flow rate determined from Step 2 and Figure 3.434 A for each value of surface operating pressure and flow rate assumed. Join the P_{wf} vs. flow rate points. This gives a P_{wf} vs. flow rate curve for each value of surface operating pressure. Intersection of these curves with the inflow performance curve gives the producing rate possible from this well for each value of surface operating pressure.

The results of Figure 3.434 show that flow rates of 2030, 2400, 2450 B/D are possible from this well for the assumed values of surface operating pressures. These injection pressures are 800, 1000, and 1200 psia, respectively. (Results are plotted in Fig. 3.434 D.) The designer can now evaluate the effect of surface operating pressures. The depth and pressure of injection can then be obtained by proceeding backwards in the cross plot as follows:

(5) From the flow rate decided upon, determine P_{so} from Fig. 3.434 D. Alternatively, from the surface operating pressure decided upon, determine the flow rate possible from the same figure.

(6) Move vertically to Fig. 3.434 C to intersect the inflow curve and determine the bottom hole pressure (P_{wf}).

(7) Move horizontally to Fig. 3.434 A to intersect the vertical line corresponding to well depth (8000 ft). The flowing gradient curve below the point of injection corresponding to the flow rate being used is then traced to intersect the gas operating line corresponding to the surface operating pressure selected in step A. If a value of surface operating pressure is to be used other than one already present, a new operating line corresponding to the new value of P_{so} must be drawn in as explained in Step 2 a of the economical design procedure. The flowing gradient curve may be traced from existing curves, interpolated between the curves already drawn, or calculated by computer. The intersection of the flowing gradient curve with the gas operating line curve gives the point of injection. The depth and pressure may be read off the scales of Figure 3.434 A.

(8) Move horizontally to Figure 3.434 B to the same pressure/depth point or to an interpolated curve for the flow rate considered in Step 5. The economical TGLR can now be read from the abscissa of Figure 3.434 B.

The economical cross plot shows, in Figure 3.434 D, that for P_{so} values greater than 1000 psia, no great increase in flow rate is obtained for additional increases in surface operating pressure at the economical TGLR. This plot assumes the same gas cost for all values of P_{so}. This may not be true in practice due to more stages of compression. The economical cross plot also gives the value for the most economical TGLR for any flow rate within the range of the ones assumed at any depth of injection or pressure of injection.

3.4253 Determination of the most economical gas-liquid ratio for variable wellhead pressure

3.42531 Introduction

Pacheco proposed two procedures for determining the most economical gas liquid ratio for gas lift design.[11] In addition, he extended the work of Mitchell to include the effect of horizontal flow lines; therefore, his solution is for a variable wellhead pressure.[10] The first method uses the vertical multiphase curves of pressure vs. TGLR for different depths to determine the most economical TGLR based on the concept of an economical slope.

The second method uses a continuous flow gas lift program to determine the maximum total liquid production where the slope of the gas lift performance curve is equal to zero, then proceeds backwards until the point at which the cost of injection of an incremental quantity of gas is equal to the profit obtained from the additional recovery of oil. This point corresponds to the most economical TGLR.

The modified Hagedorn and Brown correlation for multiphase vertical flow[4] and Dukler's friction factor correlation in combination with Eaton's holdup correlation for multiphase horizontal flow are used in this study.[7]

In the design of continuous flow gas lift installations for the variable wellhead pressure method, there exists

an optimum TGLR with which maximum production will occur; it will probably be less than that ratio required to create the minimum flowing gradient in the vertical tubing string. However, in any gas lift installation, a most economical gas injection rate exists which occurs when the profit obtained from an additional incremental recovery of oil is equal to the cost of injection of the incremental amount of gas.

3.42532 Description of the economical slope method

Mitchell discussed the economical slope method for the case of constant well head pressure.[10] Commonly, maximum production is designed by assuming that the minimum gradient can be reached in the vertical tubing string. This solution shows that this is unlikely except for extremely large surface flow lines. The low pressure surface flow lines cannot handle the large volume of expanded gas without increasing the wellhead pressure. The flowing bottom hole pressure increases, and production decreases before reaching the minimum gradient. Therefore, Mitchell's economical slope method was modified for application to variable well head pressure conditions.

Figure 3.436 is a plot of a pressure vs. TGLR curve. The pressure vs. TGLR curve for the depth of injection becomes the pressure in the tubing at the injection depth $(P_{tub})_{inj}$ vs. TGLR curve.

Assume that the most economical TGLR occurs at a point on the $(P_{tub})_{inj}$ vs. TGLR curve where the slope is X psi/TGLR (psi/scf/bbl)

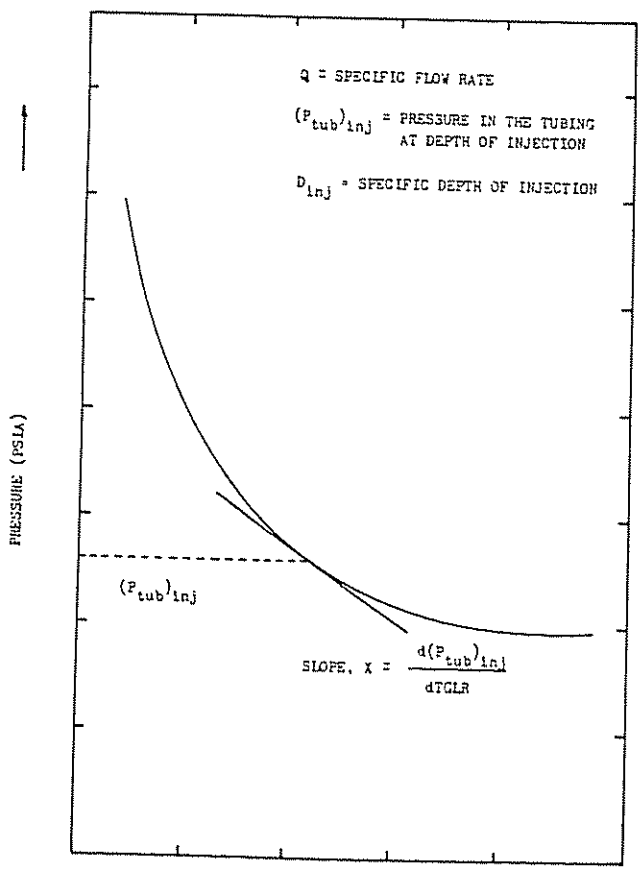


Fig. 3.436 Pressure vs. TGLR Curve

Then:

$$X = \frac{d(P_{tub})_{inj}}{d\,TGLR} \tag{3.414A}$$

For a flow rate of q B/D, the productivity index PI(B/D/psi) can be represented as the surface rate of production of liquids from the reservoir into the tubing per psi pressure drop of flowing bottom hole pressure in the tubing, (dq/dBHP).

The decrease in tubing pressure at the injection point caused by a unit increase in TGLR is associated with a net decrease in flowing bottom-hole pressure at the sand face depth which results in a small increase of inflow (dq) of fluids into the well. Therefore,

$$X\ PI = \frac{d(P_{tub})_{inj}}{dTGLR} \times \frac{dq}{dBHP} \tag{3.415}$$

At the most economical injection rate, the cost of a unit increase in TGLR (dTGLR = unity) is equal to the profit obtained from the resultant production of oil (dq). The gas injected per unit increase of TGLR is q (q B/D × 1 scf/bbl = q). If the gas cost is G $/scf and the profit on oil is PO $/bbl, at the economical point

$$dq \times PO = q \times G$$

or:

$$dq = \frac{q \times G}{PO}$$

Substituting for dTGLR = unity:

$$X\ PI = \frac{q \times G}{PO} \times \frac{d(P_{tub})_{inj}}{d\ BHP} \tag{3.416A}$$

Assuming the decrease in tubing pressure at the injection point caused by a unit increase in TGLR is approximately equal to the net decrease in flowing bottom-hole pressure at the sand face depth, the most economical slope is expressed by the formula:

$$X = \frac{q \times G}{PI \times PO} \tag{3.416B}$$

Then, if a tangent of slope X is fitted to the $(P_{tub})_{inj}$ vs. TGLR curve, the point of tangency defines the most economical slope. The tangent of zero slope fitted to this curve also defines the maximum TGLR at which the maximum production is reached.

Fig. 3.437A is a vertical multiphase flow plot of pressure vs. TGLR for different depths. Using the value of the economical slope given by Equation 3.416B, the economical TGLR can be determined at each depth. A curve joining these economical TGLR points defines the most economical TGLR curve. A curve joining the points of zero slope also defines the maximum TGLR curve.

The procedure applying to the most economical design method for varying values of surface operating pressures is:

(1) Assume various flow rates
(2) Assume various TGLR's
(3) Assume various well depths
(4) With the value of flow rate assumed in Step 1, determine the wellhead pressure for a single value of TGLR assumed in Step 2.
(5) Determine the flowing bottom-hole pressures for the values of wellhead pressures calculated in Step 4 and all depths assumed in Step 3

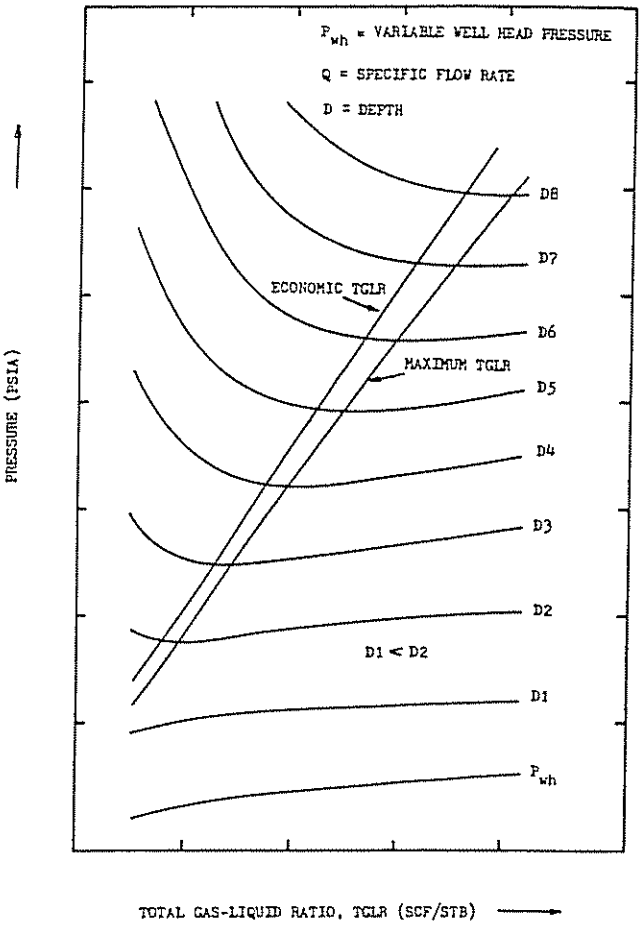


Fig. 3.437-A Pressure vs. TGLR Curves

(6) Repeat Steps 4 and 5 for all the values of TGLR
(7) Calculate the economical slope from equation 3.416B
(8) With pressure on the ordinate and TGLR on the abscissa plot the values of flowing bottom-hole pressure and TGLR from Step 5 for various depths (Fig. 3.437A).
(9) Determine the most economical TGLR and flowing bottom-hole pressure corresponding to the economical slope calculated in Step 7. Fit the most economical slope to the curve of TGLR vs. flowing bottom-hole pressure drawn in Step 8 at each depth. Calculate the maximum TGLR corresponding to a value of zero slope in the same way. Use a cubic spline interpolation scheme in combination with the Lagrange formula to accomplish these calculations.
(10) In Figure 3.437A, draw the economic TGLR curve, joining the most economical TGLR points determined in Step 9 at each depth. Determine the maximum TGLR curve by joining the maximum TGLR points.
(11) Repeat Steps 4 to 10 for the various values of flow rate
(12) Assume various surface operating pressures
(13) Calculate the gas pressure at well depth, including the effect of gas column weight for every surface operating pressure assumed less 100 psi.
(14) With pressure on the ordinate and depth on the abscissa, plot the values of surface operating pressure less 100 psi at zero depth. Also, plot the values of gas pressure calculated in Step 13 at well depth. A line joining the points for the same operating pressure defines the gas operating line (Fig. 3.437B).
(15) In Figure 3.437B, plot the values of flowing bottom-hole pressure corresponding to the economical TGLR calculated in Step 9 for a single flow rate. Repeat for all flow rates. Join the points corresponding to the same flow rate with a straight line. The intersection of these lines with the gas operating lines defines the depth of injection and the pressure of injection for the given flow rate and surface operating pressure.
(16) With pressure on the ordinate and TGLR on the abscissa, plot the values of economical TGLR calculated in Step 9 for each depth and flow rate (Fig. 3.437C). Draw a smooth curve to join points of the same depth. This plot can be used to calculate the most economical TGLR given the flow rate, injection depth, and injection pressure from Figure 3.437B.
(17) Calculate the flowing-bottom pressure for the injection points determined in Step 15 (Fig. 3.437B), using the formation gas-liquid ratio. The flowing bottom-hole pressures determined

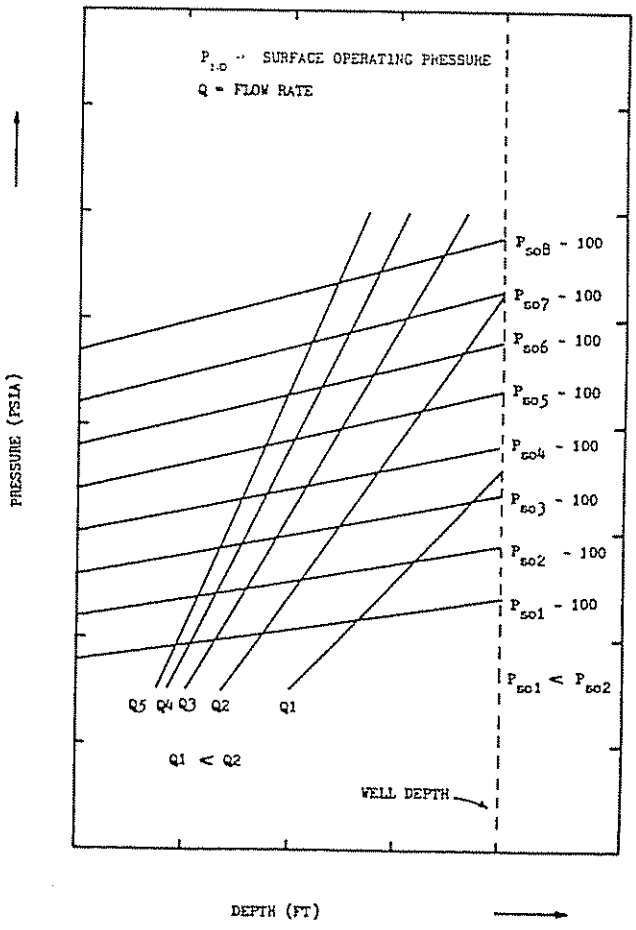


Fig. 3.437-B Determination of Injection Point

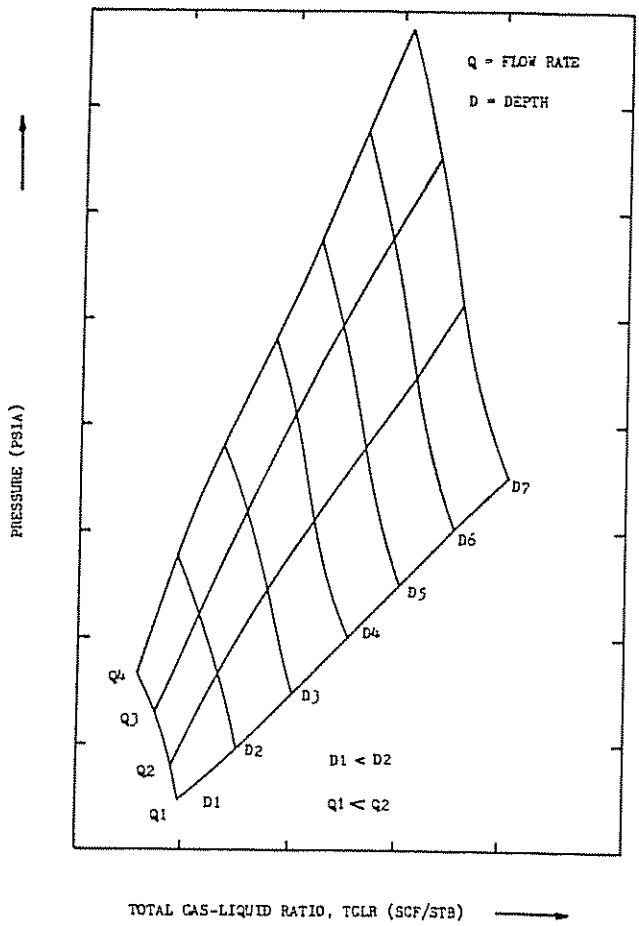


Fig. 3.437-C Determination of The Most Economical TGLR

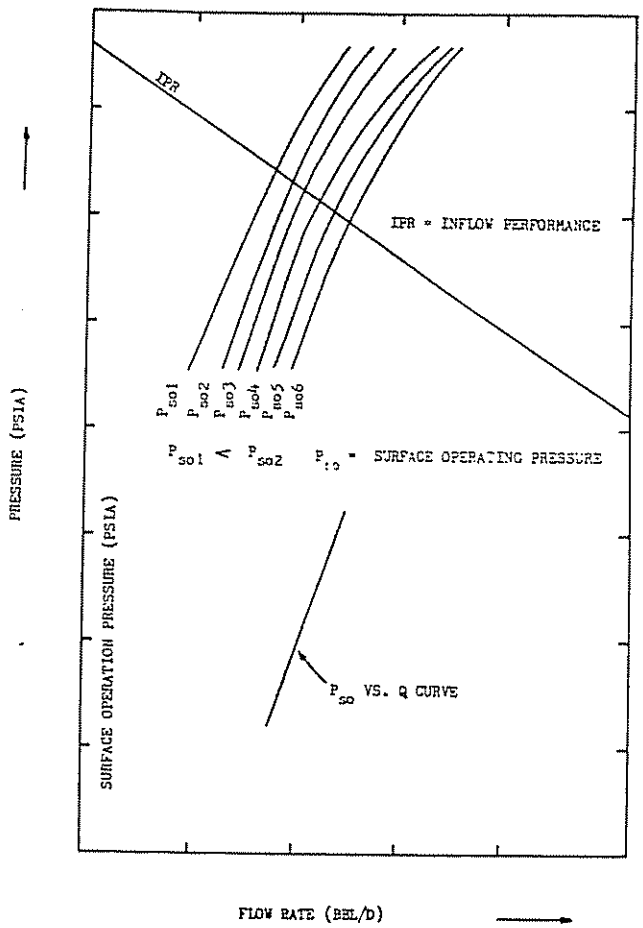


Fig. 3.437-D Determination of the Most Economical Flow Rate

in this step are the bottom-hole pressures which are required for the well to flow up the tubing at the assumed rates.

(18) With pressure on the ordinate and flow rate on the abscissa, plot the inflow performance curve (IPR) of the well (Fig. 3.437D).

(19) In Figure 3.437D, plot the values of flowing bottom-hole pressure obtained in Step 17 for a single value of surface operating pressure vs. flow rate. The intersection of the inflow performance curve with the surface operating curve represents the most economical flow rate for that surface operating pressure. Repeat for all operating pressures.

(20) Plot the most economical flow rates from Step 19 vs. surface operating pressure in Fig. 3.437D. This curve gives the relationship between surface operating pressure and flow rate for the gas lift system, corresponding to the most economical TGLR.

3.42533 Profit on oil vs. cost of gas injected method

In this method the combination of Dukler's friction factor correlation and Eaton's holdup correlation for horizontal flow is used to determine the wellhead pressure, and the Hagedorn and Brown correlation for vertical flow is used to determine the flowing bottom-hole pressure based on the wellhead pressure obtained from the horizontal correlation. The maximum flow rate is predicted by using the depth-pressure diagram and a length-pressure diagram to determine the flowing bottom-hole pressure from vertical correlations with the wellhead pressures obtained from the horizontal correlations.

From the separator pressure at zero depth, the flowing horizontal pressure traverse is calculated and the wellhead pressure is determined for the different total gas-liquid ratios assumed and for the different flow rates. With the wellhead pressures determined for each flow rate and different TGLR's, the flowing bottom-hole pressures are determined from the depth-pressure diagram. With these values and taking into consideration the inflow performance relationship of the well in a pressure-flow rate diagram, the maximum flow rate is determined for the maximum TGLR.

In using the profit on oil vs. cost of gas injected method, the flow rate is determined for incremental TGLR's until the maximum flow rate is reached. As shown in Figure 3.438A the flow rate increases up to a maximum rate, as gas injection volumes increase; afterwards, the flow rate decreases with additional gas injected. When the maximum flow rate is reached, a comparison—starting from the first gas increment—is made between the profit on oil obtained for every additional oil increment and the cost of gas required to produce this increment of oil recovery. The TGLR at which the cost of injecting an incremental amount of gas is equal

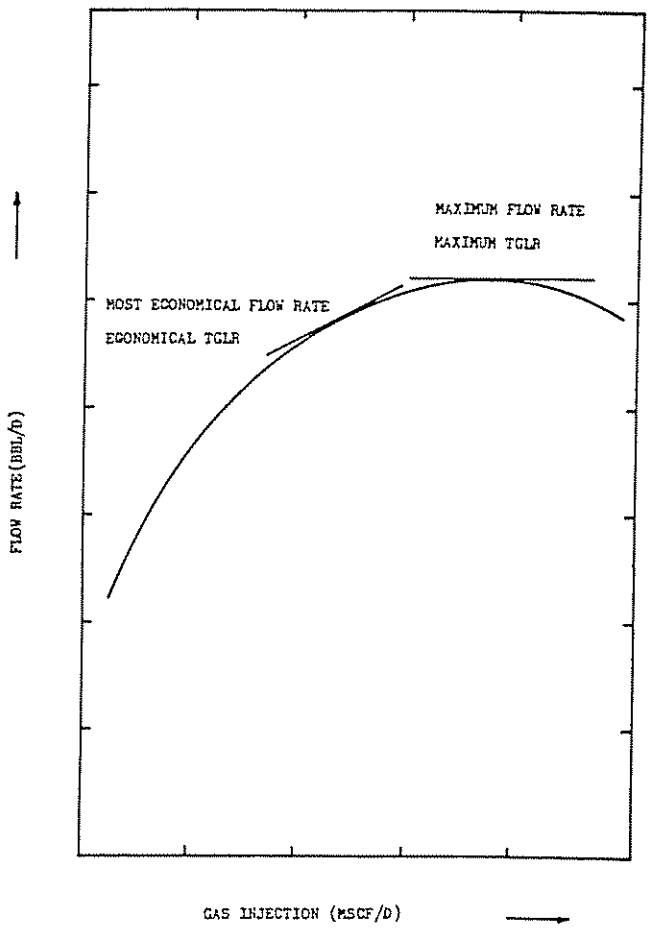


Fig. 3.438-A Continuous Flow Gas Lift Performance Curve

to the profit obtained from the additional recovery of oil is taken as the most economical TGLR.

The following procedure applies to the most economical design for varying values of surface operating pressure when the profit on oil vs. cost of gas injected procedure is used:

(1) Determine whether or not the well will flow naturally and, if so, at what rate. Use the following procedure:
  (a) Assume a flow rate
  (b) Calculate the flowing bottom-hole pressure for the assumed flow rate and for the given static bottom-hole pressure and productivity index
  (c) Calculate the wellhead pressure for the assumed flow rate using a horizontal correlation
  (d) Calculate the flowing bottom-hole pressure for the calculated wellhead pressure using the vertical correlation
  (e) Compare the flowing bottom-hole pressure calculated in (b) and (d). If the difference between them is less than 1 psi, the assumed flow rate is the natural flowing rate of the well. If the difference is greater than 1 psi, another flow rate is calculated for the computed flowing bottom-hole pressure in (d) and for the given static bottom-hole pressure and productivity index. This procedure is repeated until convergence is obtained.

(2) After determining natural flowing conditions, gas is injected into the well to determine the maximum flow rate for the given TGLR and surface operating pressure. The procedure is as follows:
  (a) Assume a flow rate greater than the natural flowing rate
  (b) Calculate the TGLR adding the formation gas-liquid ratio to an incremental gas-liquid ratio. The incremental gas-liquid ratio was selected as 50 scf/STB for this study.
  (c) Calculate the wellhead pressure for the assumed flow rate and TGLR using the horizontal correlation
  (d) Calculate the flowing bottom-hole pressure for the assumed flow rate, TGLR, given static bottom-hole pressure and productivity index
  (e) Calculate the casing gas pressure at the bottom from the known surface operating pressure less 100 psi, considering the gas column weight
  (f) Compare the calculated flowing bottom-hole pressure in (d) and the calculated casing gas pressure plus 100 psi at the bottom. If the casing gas pressure at the bottom is greater or equal to the flowing bottom-hole pressure, the injection point is located at the bottom.
  (g) If the casing gas pressure at the bottom is less than the flowing bottom-hole pressure, the injection point is calculated as indicated in (h) and (i)
  (h) Divide the well depth into increments of approximately 200 feet each
  (i) Calculate the casing gas pressure (dynamic for high gas rates) and the flowing bottom-hole pressure for the TGLR at the same depth. Do this down the hole for each vertical increment until the casing gas pressure is greater than or equal to the flowing bottom-hole pressure. A linear interpolation determines the depth of gas injection and the injection pressure where the flowing gradient curve intersects the gas column gradient. Use the vertical correlation to calculate the flowing bottom-hole pressure (see Fig. 3.438B).
  (j) Use the formation gas-liquid ratio to calculate the traverse below the point of injection and determine the flowing bottom-hole pressure using the vertical correlation.
  (k) Compare the flowing bottom-hole pressure calculated for the assumed flow rate and given static bottom-hole pressure and productivity index with the flowing bottom-hole pressure calculated in (j). If the difference between them is greater than 1 psi, calculate another flow rate using the flowing bottom-hole pressure from (j) and the given static bottom-hole pressure and productivity index, and repeat the procedure from step (c) until convergence is obtained. If the difference is less than or equal to 1 psi, the assumed flow rate is the correct flow rate for the assumed TGLR.
  (l) Repeat the procedure from Step (a) for new

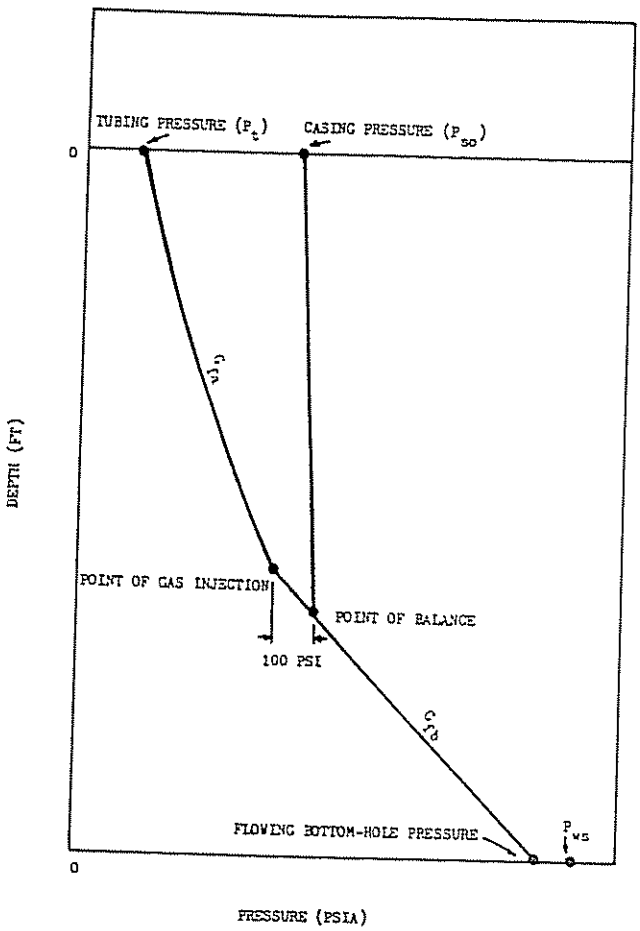


Fig. 3.438-B Continuous Flow Well Illustration

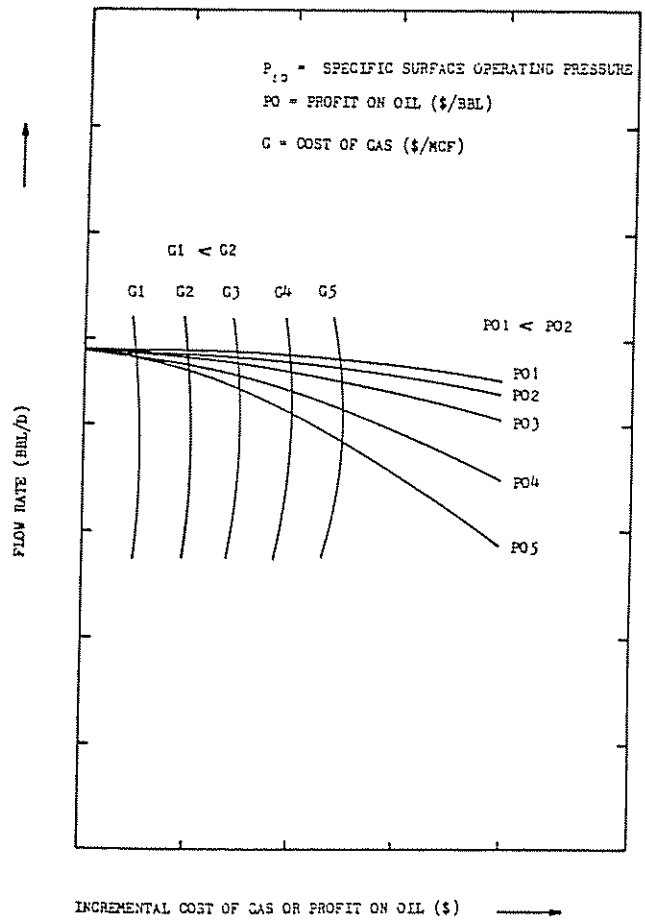


Fig. 3.438-C Determination of The Most Economical Flow Rate

gas-liquid ratio increments until the flow rate does not increase with additional gas liquid ratio increments.

(3) Assume various values of profit on oil ($/bbl)

(4) Assume various values of gas cost ($/Mcf)

(5) For a single value of profit on oil ($bbl) and all the values of gas cost ($/Mcf) determine the flow rates and the gas-liquid ratios where the profit on oil ($) obtained from an increase in gas injected is equal to the cost of gas ($) required to obtain the additional oil recovery. Do this by comparing the values of profit on oil ($) and the values of gas cost ($) for every gas-liquid ratio increment. The flow rates and TGLR's determined in this manner are the most economical flow rate and the most economical TGLR, respectively.

(6) Repeat Step 5 for all the values of profit on oil assumed in Step 3

(7) With flow rate on the ordinate and profit on oil ($) or cost of gas ($) on the abscissa, plot the values of profit on oil ($) and cost of gas ($) vs. flow rate, determined in Step 5 for the comparison procedure. The intersection of the different combinations of gas cost ($/Mcf) and the profit on oil ($/bbl) are the most economical flow rates which correspond to the most economical TGLR's (Fig. 3.438C).

(8) Assume other values of surface operating pressure and repeat the procedure from Step 2.

(9) With flow rate on the ordinate and TGLR on the abscissa, plot the values of TGLR vs. flow rate determined in (i) for each value of surface operation pressure (Fig. 3.438D.) From the most economical flow rate determined in Figure 3.438C, the most economical TGLR can be determined from Figure 3.438D.

(10) With flow rate on the ordinate and injection depth on the abscissa, plot the values of injection depth determined in (i) for each value of surface operating pressure and different TGLR's corresponding to each gas-liquid ratio increment. Join the points corresponding to the same TGLR with a smooth curve (Fig. 3.438E). From the most economical TGLR determined in Figure 3.438D and the corresponding surface operating pressure, the injection depth can be determined in Figure 3.438E.

(11) With flow rate on the ordinate and injection pressure on the abscissa, plot the values of injection pressure determined in (i) for each value of surface operating pressure and different TGLR's corresponding to each gas-liquid ratio increment. Join the points corresponding to the same TGLR with a smooth curve (Fig. 3.438F). From the most economical TGLR determined in Figure 3.438D and the appropriate surface operating pressure, the pressure at the injection point can be determined in Figure 3.438F.

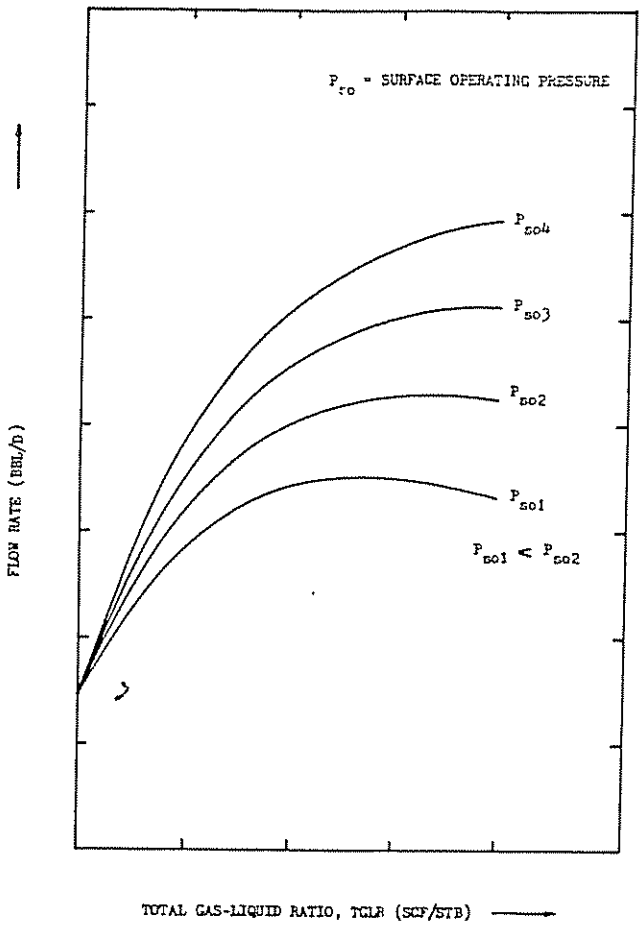


Fig. 3.438-D Determination of The Most Economic TGLR

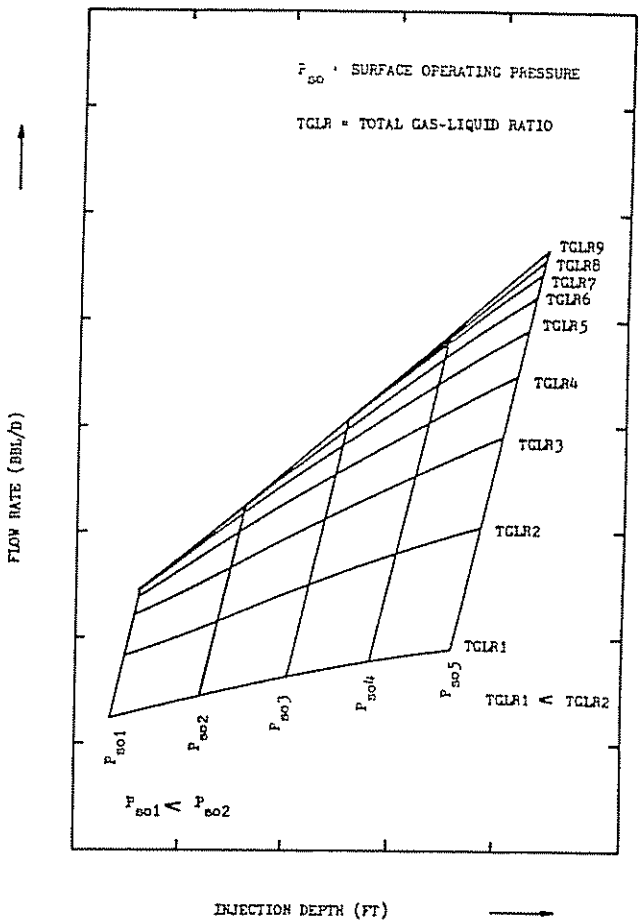


Fig. 3.438-E Determination of The Depth of Injection

3.42534 Example problems for the most economical gas lift design—variable wellhead pressure

An example problem illustrates the application of both procedures by Pacheco for continuous flow gas lift design.[11]

The following well information is used:

Depth of well = 8000 ft
Tubing size = 2.441 in. (I.D.)
Flowline length = 4000 ft
Flowline size = 2.5 in.
Static bottom hole pressure = 2400 psi
Surface operating pressures = 800, 1000, 1200 psi
Separator pressure = 80 psi
Separator temperature = 100°F.
Productivity index = 3 B/D psi
Formation gas-liquid ratio = 200 scf/STB
Water cut = 0.0 %
Specific gravity of gas = 0.65
Oil API gravity = 35°
Specific gravity of water = 1.07
Surface temperature = 120°F.
Bottom-hole temperature = 180°F.
Profit on oil = $ 4/bbl
Gas cost = $ 1/mcf

Required: Flow rates possibles from this well at the most economical TGLR.

3.425341 Economical slope method

Solution:

(1) Assume flow rates of 500, 1000, 1500, 2000 STB/D
(2) Assume TGLR's from 100 to 1300 scf/STB with increments of 100 SCF/STB
(3) Assume depth increments of 1000 ft to the well depth of 8000 ft
(4) For a flow rate of 500 STB/D and a TGLR of 100 scf/STB, calculate the wellhead pressure for a constant separator pressure of 80 psi and a flowline length of 4000 ft using the horizontal correlation
(5) Calculate the flowing bottom-hole pressures corresponding to the value of wellhead pressures from Step 4 for depth increments of 1000 ft to the well depth, using the vertical correlation
(6) Repeat steps 4 and 5 for values of TGLR from 200 to 1300 scf/STB with increments of 100 scf/STB
(7) Calculate the economical slope for a flow rate of 500 STB/D, a PI of 3 bpd/psi, a profit on oil of $ 4/bbl and a gas cost of $ 1/bbl using Equation 3.416 B as follows:

$$\chi = \frac{q \times G}{PO \times PI} = \frac{500 \times 1}{4 \times 3 \times 1000} = 0.0417 \text{ psi/TGLR}$$

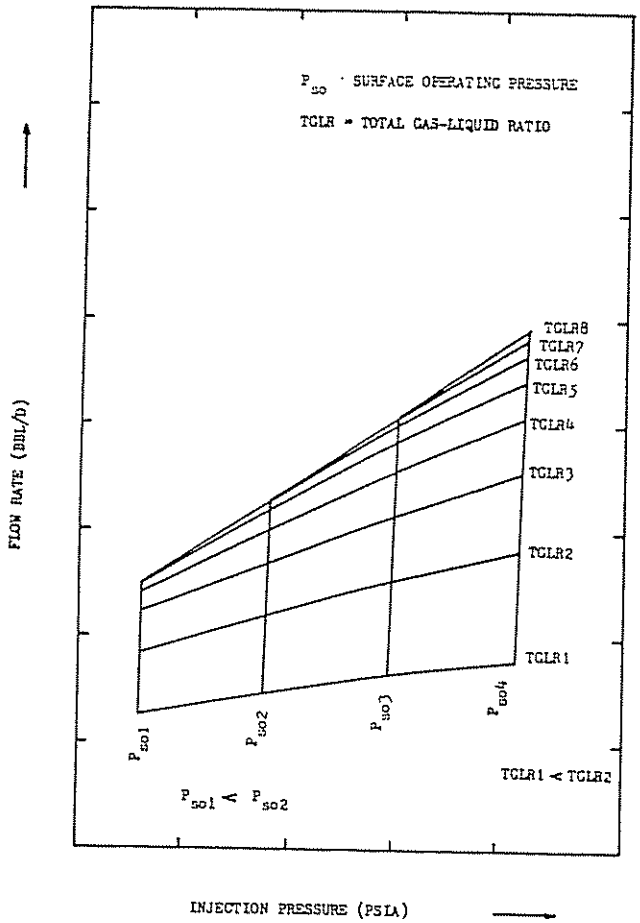


Fig. 3.438-F Determination of Injection Pressure

(8) Plot the values of flowing bottom-hole pressures from Step 5 vs. TGLR for depth increments of 1000 ft to the well depth of 8000 ft (Fig. 3.439A)

(9) Estimate values for the most economical TGLR and corresponding flowing bottom-hole pressure from Fig. 3.439A; fit a tangent of slope equal to the economical slope value, determined in Step 7, to the curve for each depth increment. Estimate the maximum TGLR and corresponding flowing bottom-hole pressure for a value of zero slope in the same way.

(10) Draw the economic TGLR curve in Fig. 3.439A, joining the most economical TGLR points from Step 9 at each depth. Also, draw the maximum TGLR curve by joining the points of zero slope.

(11) Repeat the procedure from Step 4 to 10 for values of flow rate of 1000, 1500 and 2000 STB/D

(12) Assume surface operating pressures of 800, 1000 and 1200 psi.

(13) Calculate the gas pressures at well depth including the effect of gas column pressure for each surface operating pressure assumed in Step 12. The surface operating pressures of 800, 1000, and 1200 psi give 849.5, 1090.2 and 1331.7 psi at 8000 ft.

(14) Plot the values of gas casing pressure at well depth and zero depth less 100 psi vs. depth. Lines joining the pair of points for the same surface operating pressure define the gas operating lines (Fig. 3.439B)

(15) Plot the values of flowing bottom-hole pressure from Step 9 corresponding to the economical TGLR in Fig. 3.439B for each flow rate. Join points corresponding to the same flow rate. The intersections of these lines with the gas operating pressure lines correspond to the depth and pressure at the injection point for a given flow rate and surface operating pressure.

(16) Plot the values of economical TGLR, calculated in Step 9, vs. pressure in Fig. 3.439C for each depth and flow rate. Join points corresponding to the same flow rate by smooth lines. Draw a smooth curve to join points of the same depth. Calculate the most economical TGLR from Fig. 3.439C, using the results of Fig. 3.439B.

(17) Calculate the flowing bottom-hole pressures for the injection points determined in Step 15, using the vertical correlation and the formation gas liquid ratio.

(18) The inflow performance curve of the well is plotted in Fig. 3.439D.

(19) Plot the values of flowing bottom-hole pressure determined in Step 17 vs. flow rate in Fig. 3.439D for each value of surface operating pressure. Join points corresponding to the same surface operating pressure with a smooth line. The intersection of the inflow performance curve with the surface operating curve represents the

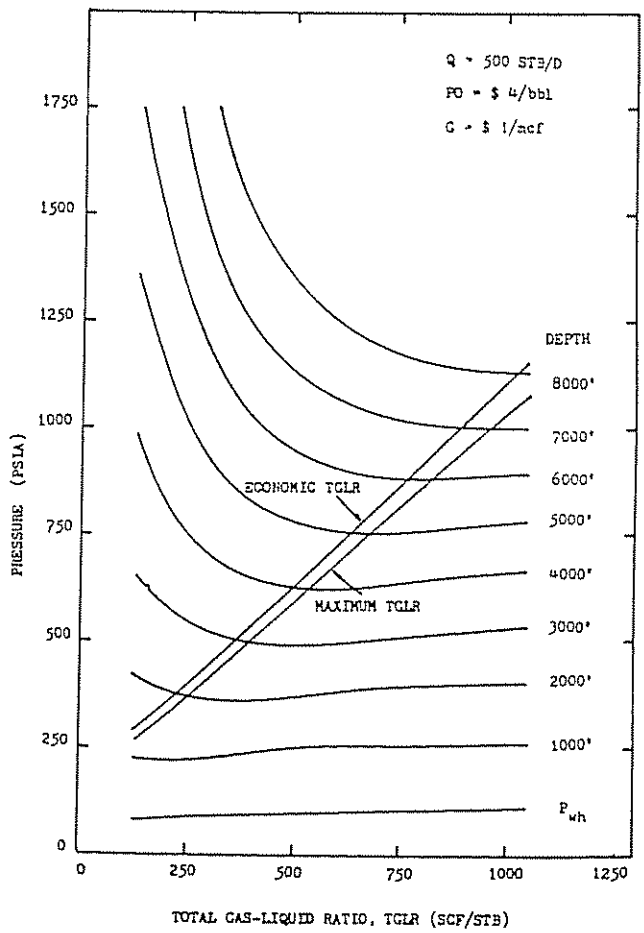


Fig. 3.439-A Pressure vs. TGLR Curves for A Flow Rate of 500 STB/D

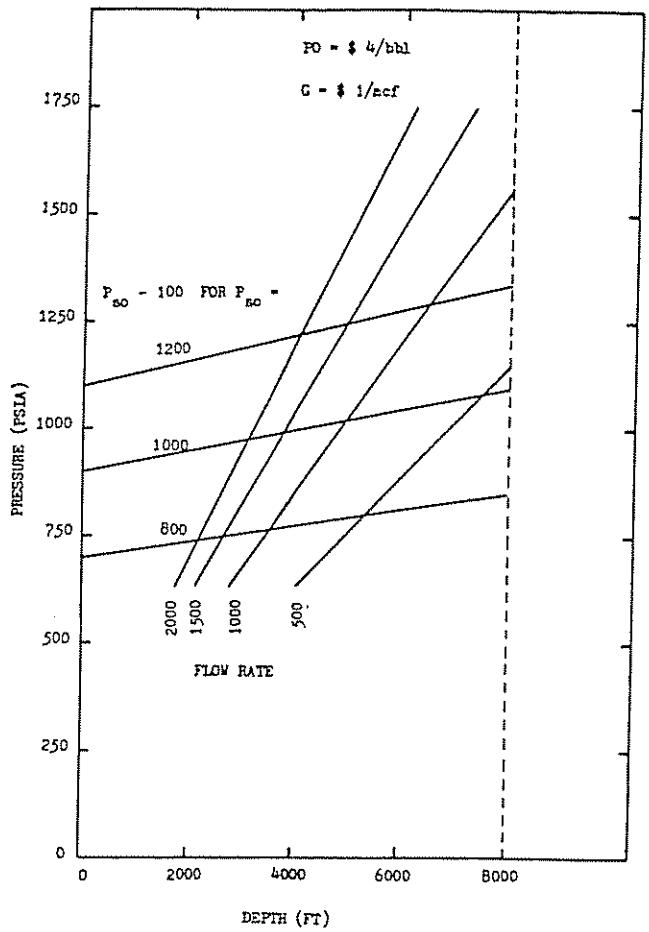


Fig. 3.439-B Determination of Injection Point

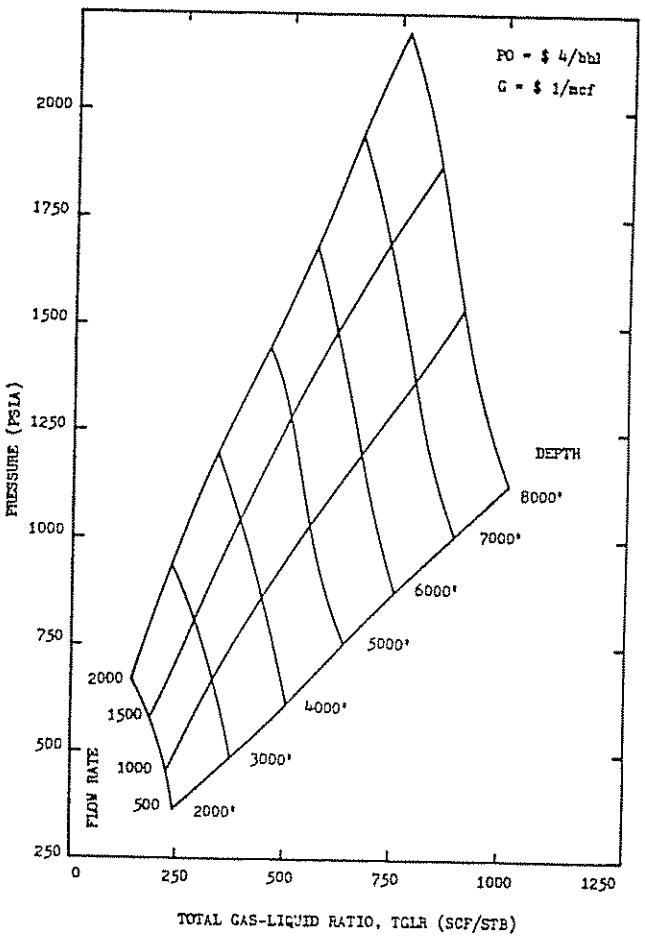


Fig. 3.439-C Determination of The Most Economical TGLR

most economical flow rate for the given surface operating pressure. Results of these calculations give flow rates of 885, 1025 and 1180 B/D for surface operating pressures of 800, 1000 and 1200 psi.

(20) The values of the most economical flow rates from Step 19 are plotted vs. surface operating pressure in Fig 3.439D. This plot gives the relationship between surface operating pressure and flow rate at the most economical TGLR.

A summary of the results obtained for the most important gas lift parameters is presented in Table 3.433.

3.425342 Profit on oil vs. cost of gas injected method

Using the same example problem of the previous section, the application of the profit on oil vs. cost of gas injected method for continuous flow gas lift design is illustrated following the procedure outlined in Section 3.42533.

Solution:

(1) A flow rate of 700 STB/D is assumed and the iterative procedure outlined earlier is followed to determine the natural flowing conditions of the well. These results show a flow rate of 632 B/D for a flowing pressure of 2189.7 psi and wellhead pressure of 91.1 psi.

(2) A surface operating pressure of 800 psi is assumed and the iterative procedure followed to determine the flow rates for incremental TGLR's, until passing the maximum rate. Results of these calculations are in Table 3.434.

(3) Values of profit on oil of 3, 4, 6, 8 and $10/bbl are assumed.

(4) Values of gas of 0.5, 1.0, 1.5, 2.0, and $2.5/Mcf are assumed.

(5) Flow rates and TGLR's are determined for a single value of profit on oil and all gas costs assumed in Step 4 until the profit on oil ($) obtained from an incremental amount of gas injected is equal to the cost of gas ($) required to obtain the additional oil recovery.

(6) Step 5 is repeated for all values of profit on oil. Results from Steps 5 and 6 are in Table 3.435.

(7) The values of flow rate determined in Step 5 are plotted vs. gas cost ($) or profit on oil ($) in Fig. 3.439E. The intersection of the different combinations of profit on oil ($/bbl) and gas cost ($/mcf) are the most economical flow rates which correspond to the most economical TGLR's.

(8) The procedure from Step 2 through 7 is repeated for values of surface operating pressures of 1000 and 1200 psi. Refer to Fig. 3.439E for P_{so} = 800 psi.

(9) The values of flow rate and TGLR determined

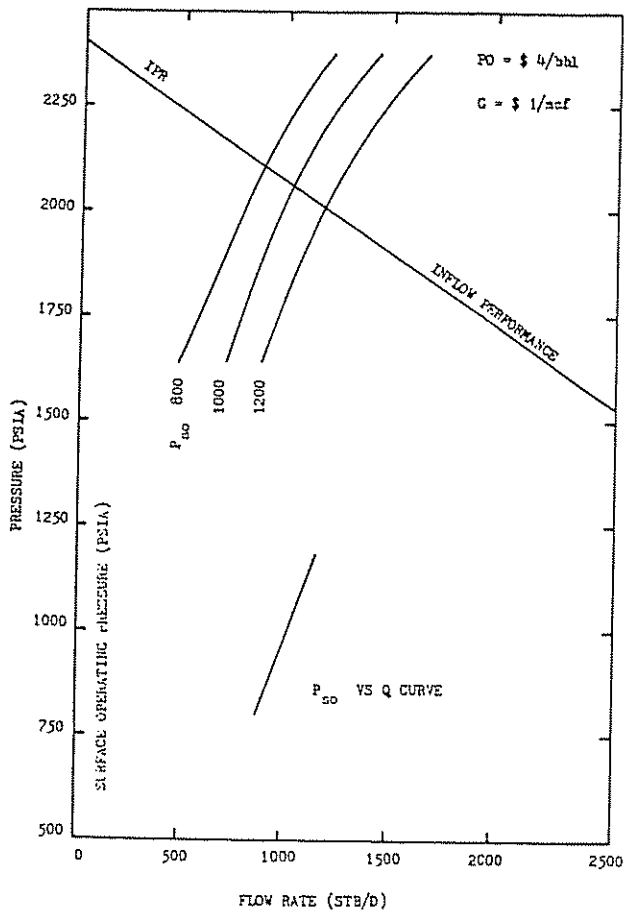


Fig. 3.439-D Determination of The Most Economical Flow Rate

TABLE 3.433
SUMMARY OF RESULTS FOR CONTINUOUS FLOW GAS LIFT DESIGN USING THE ECONOMICAL SLOPE METHOD

Surface operating pressure (psia)	Injection depth (ft)	Injection pressure (psia)	Economical TGLR (SCF/STB)	Economical flow rate (B/D)
800	3950	770	450	885
1000	4900	1015	525	1025
1200	5800	1265	625	1180

TABLE 3.434
GAS LIFT CONDITIONS FOR A SURFACE OPERATING PRESSURE OF 800 PSIA

Flow rate (B/D)	Injection GLR (SCF/STB)	Total GLR (SCF/STB)	Injection pressure (psia)	Injection depth (ft)	Flowing wellhead pressure (psia)	Flowing bottom hole pressure (psia)
728.3	50.0	250.0	758.7	3676.8	98.1	2157.2
788.8	100.0	300.0	759.7	3743.8	104.6	2136.9
826.4	150.0	350.0	760.4	3782.5	110.6	2125.3
845.7	200.0	400.0	760.7	3805.2	115.8	2118.3
853.7	250.0	450.0	760.0	3815.2	120.2	2115.3
853.7	300.0	500.0	761.0	3816.0	123.8	2115.1

TABLE 3.435
RESULTS AT ECONOMICAL TGLR FOR A SURFACE OPERATING PRESSURE OF 800 psia

Profit on oil ($/bbl)	Cost of gas ($/Mcf)	Economical flow rate (STB/D)	Economical TGLR (scf/STB)
3.0	0.5	835.7	450.0
3.0	1.0	848.8	417.1
3.0	1.5	842.2	389.7
3.0	2.0	834.4	368.5
4.0	0.5	853.7	450.0
4.0	1.0	851.5	434.4
4.0	1.5	847.4	409.2
4.0	2.0	842.2	389.7
4.0	2.5	836.3	373.6
6.0	0.5	853.7	450.0
6.0	1.0	853.0	450.0
6.0	1.5	851.5	434.0
6.0	2.0	848.8	417.1
6.0	2.5	846.0	401.6
8.0	0.5	853.7	450.0
8.0	1.0	853.7	450.0
8.0	1.5	853.5	448.5
8.0	2.0	851.5	434.4
8.0	2.5	849.5	421.3
10.0	0.5	853.7	450.0
10.0	1.0	853.7	450.0
10.0	1.5	853.7	450.0
10.0	2.0	853.1	445.6
10.0	2.5	851.5	434.4

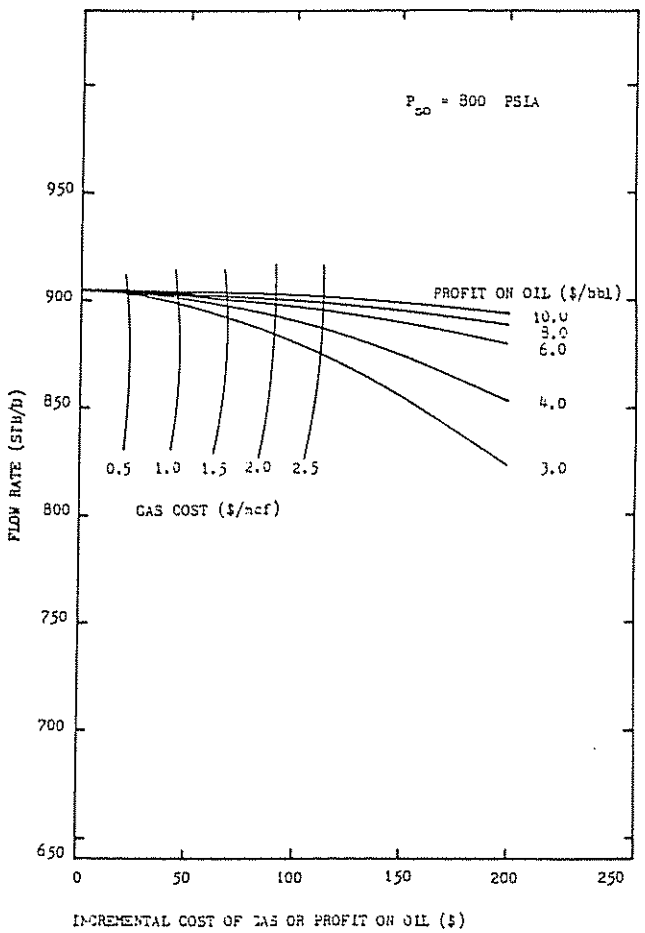


Fig. 3.439-E Determination of The Most Economical Flow Rate

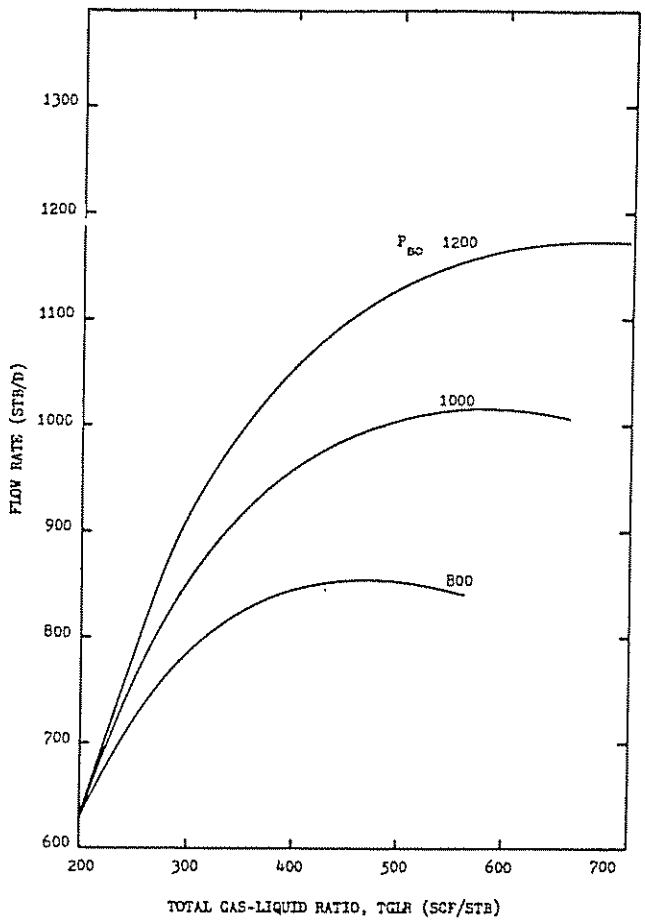


Fig. 3.439-F Determination of the Most Economical TGLR

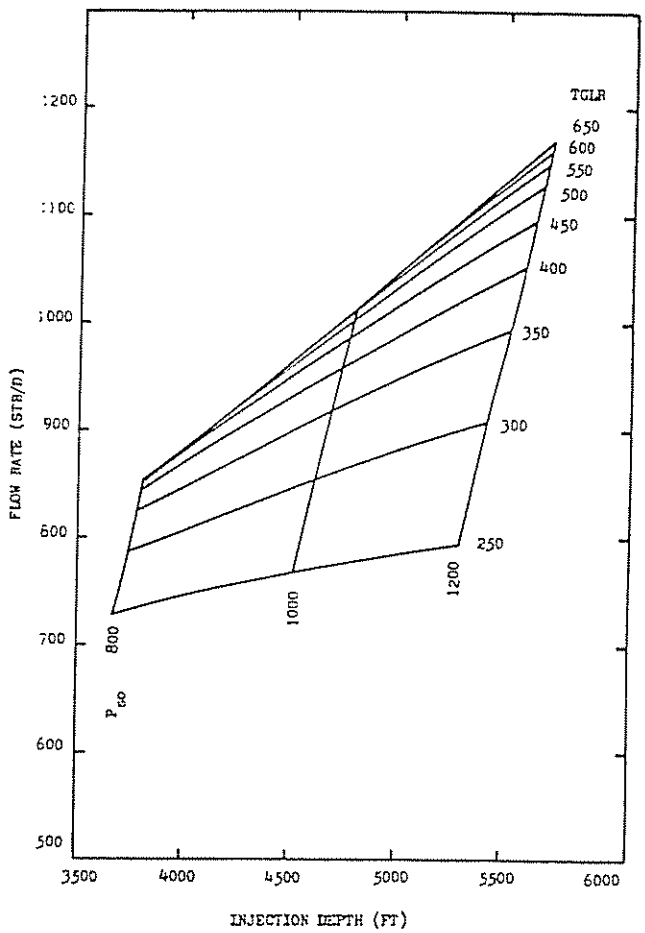


Fig. 3.439-G Determination of the Depth of Injection

in Step 2 are plotted for each value of surface operating pressure in Fig. 3.439F. From the most economical flow rate determined in figures such as 3.439E, the most economical TGLR can be calculated in Fig. 3.439F.

(10) Plot the values of injection depth determined in the iterative procedure of Step 2 vs. flow rate for each value of surface operating pressure and different TGLR's corresponding to each gas liquid ratio increment (Fig. 3.439G). The points corresponding to the same TGLR are joined with a smooth curve. From the most economical TGLR determined in Fig. 3.439F, the depth of injection can be determined from Fig. 3.439G.

(11) Plot the values of injection pressure determined in the iterative procedure of Step 2 vs. flow rate for each value of surface operating pressure and different TGLR's corresponding to each gas liquid ratio increment (Fig. 3.439H). Join the points corresponding to the same TGLR with a smooth curve. From the most economical TGLR determined in Fig. 3.439F and appropriate surface operating pressure, the pressure at the injection point can be determined from Fig. 3.439H.

A summary of the results obtained for the most important gas lift parameters, using the profit on oil vs. cost of gas method for continuous flow gas lift design, shows rates of 852, 1008 and 1164 B/D for total gas-liquid ratios of 434, 515 and 584 for surface operating pressures of 800, 1000 and 1200 psi.

3.42535 Effect of variables

To show how changes in flowline length and flowline diameter affect the flowing characteristics in a continuous flow gas lift system, the example problem illustrating the application of the methods for economical design was solved for different values of flowline length and flowline diameter. Tables 3.436A and Table 3.436B show a summary of the results obtained in these calculations for the most important gas lift parameters.

3.425351 Effect of flowline length

Figure 3.440A shows the effect of flowline length on flow rate and TGLR for a given tubing diameter of 2.441 inches, flowline diameter of 2.5 inches, and a surface operating pressure of 1000 psi. Any increase in flowline length decreases flow rate. For example, a TGLR of 600 scf/STB for a flowline length of 1000 feet gives a flow rate of 1109 STB/D, while a flowline length of 5000 feet gives a flow rate of 995 STB/D.

Figure 3.440B shows the effect of flowline length on wellhead pressure and TGLR for the same conditions given above. For these conditions any increase in flow

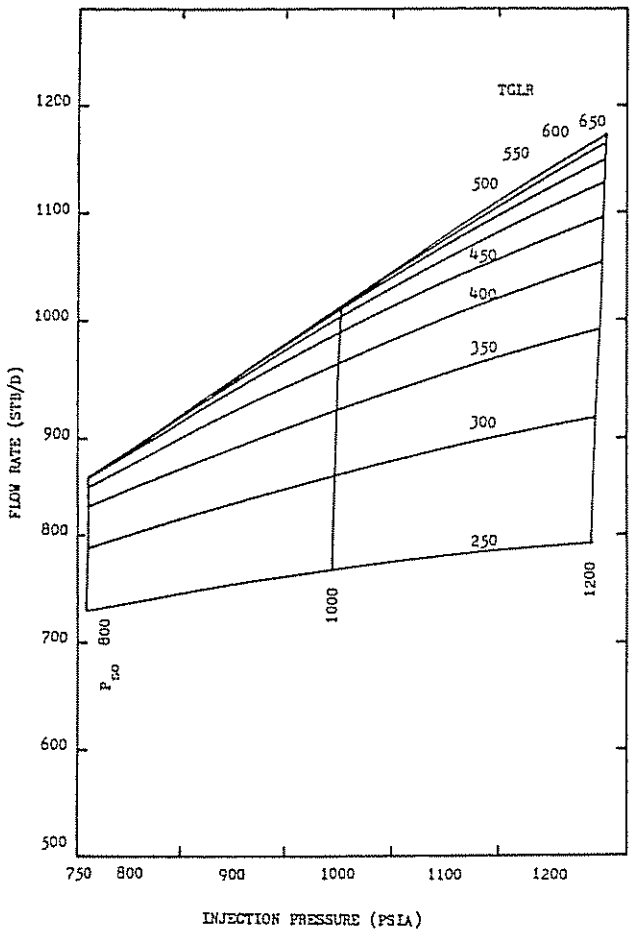


Fig. 3.439-H Determination of Injection Pressure

TABLE 3.436A
SUMMARY OF RESULTS FOR CONTINUOUS FLOW
GAS LIFT DESIGN USING DIFFERENT FLOWLINE LENGTHS

Flowline length (ft)	Surface operating pressure (psia)	Injection depth (ft)	Injection pressure (psia)	Economical TGLR (scf/STB)	Economical flow rate (B/D)
1000	800	3886	762	452	917
1000	1000	4896	1003	537	1101
1000	1200	5845	1253	603	1281
2000	800	3857	762	446	891
2000	1000	4854	1002	528	1063
2000	1200	5789	1251	594	1232
3000	800	3834	761	441	870
3000	1000	4820	1001	520	1033
3000	1200	5747	1250	590	1195
4000	800	3814	761	434	852
4000	1000	4792	1001	515	1008
4000	1200	5712	1249	584	1164
5000	800	3797	761	430	836
5000	1000	4768	1000	512	988
5000	1200	5683	1249	580	1139
6000	800	3781	760	427	822
6000	1000	4748	1000	512	970
6000	1200	5658	1248	575	1117

TABLE 3.436B
SUMMARY OF RESULTS FOR CONTINUOUS FLOW
GAS LIFT DESIGN USING DIFFERENT FLOWLINE SIZES

Flowline size (in.)	Surface operating pressure (psia)	Injection depth (ft)	Injection pressure (psia)	Economical TGLR (scf/STB)	Economical flow rate (B/D)
2.5	800	3813	761	434	851
2.5	1000	4792	1001	515	1008
2.5	1200	5712	1249	498	1132
3.0	800	3868	762	445	901
3.0	1000	4871	1002	533	1079
3.0	1200	5814	1252	601	1254
4.0	800	3909	762	459	938
4.0	1000	4932	1003	545	1133
4.0	1200	5895	1254	615	1326

line length causes an increase in wellhead pressure. For example, a TGLR of 600 scf/STB for a flowline length of 1000 feet corresponds to a wellhead pressure of 104 psi, while a flowline length of 5000 feet gives a wellhead pressure of 157 psi.

3.425352 Effect of flowline diameter

Figure 3.440C shows the effect of flowline diameter on the flow rate and TGLR. For a given tubing diame-

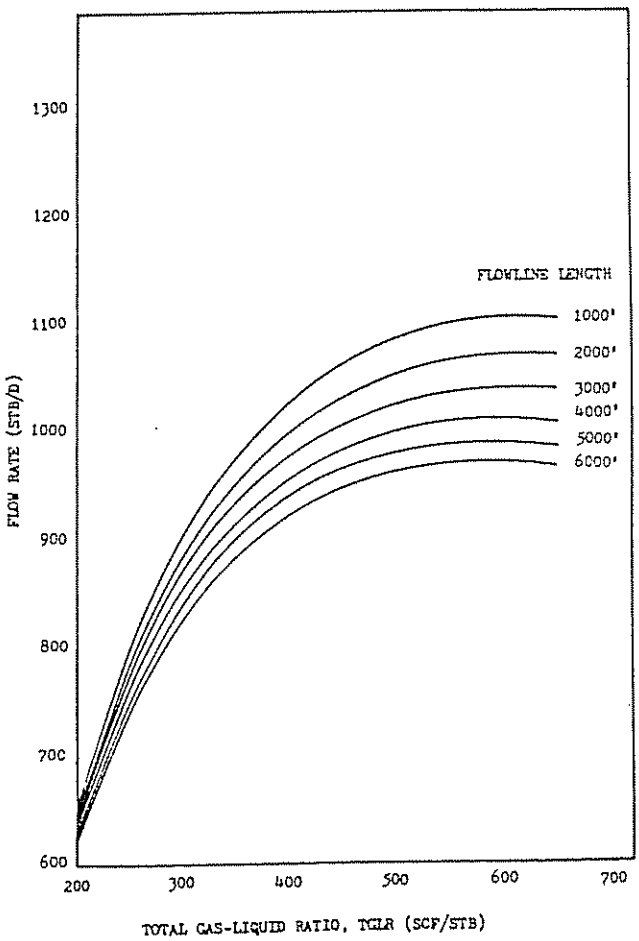


Fig. 3.440-A Effect of Flowline Length on Flow Rate

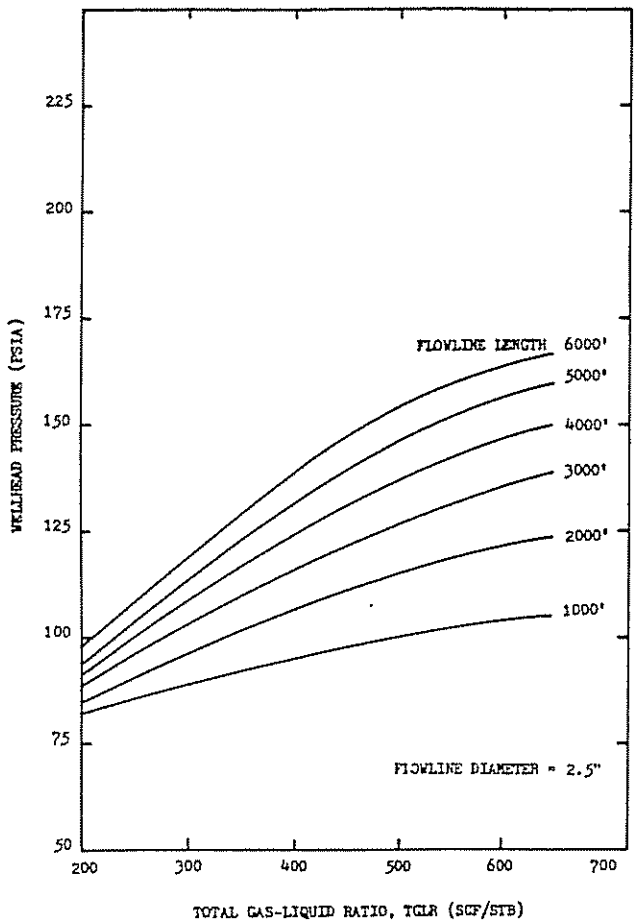


Fig. 3.440-B Effect of Flowline Length on Wellhead Pressure

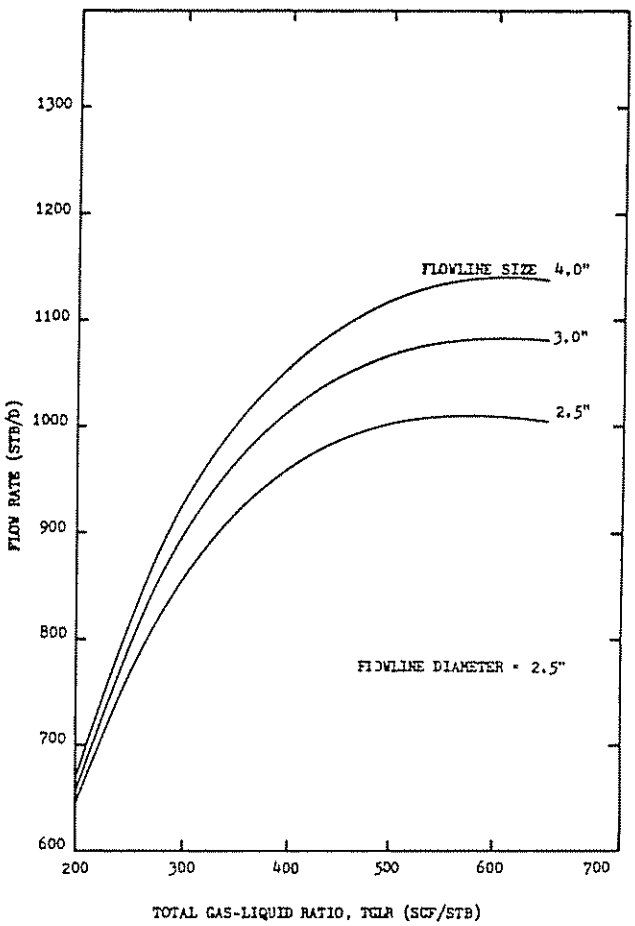


Fig. 3.440-C Effect of Flowline Size on Flow Rate

ter of 2.441 inches, flowline length of 4000 feet, and surface operating pressure of 1000 psi, any increase in flowline diameter increases flow rate. For example, a TGLR of 600 scf/STB in a 2.5-in. flowline gives a flow rate of 1016 STB/D, while a 4.0-in. flowline gives a flow rate of 1141 STB/D.

Figure 3.440D shows the effect of flowline diameter on wellhead pressure and TGLR for the same conditions given above. Any increase in flowline diameter decreases wellhead pressure. For example, a TGLR of 600 scf/STB in a 2.5-in. flowline requires a wellhead pressure of 147 psi, while a 4.0-in. flowline requires a wellhead pressure of 89 psi.

3.42536 Discussion of results

Two different methods were developed to determine the most important gas-lift parameters in continuous flow gas-lift design for variable wellhead pressure conditions. Method I utilizes the concept of an economical slope, and Method II determines the point where the cost of injection of an incremental quantity of gas is equal to the profit obtained from the additional recovery of oil.

An example problem was solved to illustrate the application of both methods. The well in the example problem can produce 632 STB/D by natural flow, with a formation gas-liquid ratio of 200 scf/STB. For Method I and for a surface operating pressure of 1000 psia, the production is increased to 1025 STB/D for an economical TGLR of 525 SCF/STB. When Method II is used, the production is increased to 1008 STB/D for an economical TGLR of 515 scf/STB.

To apply Methods I and II, various figures can be used to relate the parameters of flow rate, TGLR, surface operating pressure, injection pressure, and injection depth at the most economical point.

For Method II the preparation of one graph can be applied for all combinations of profit on oil vs. gas cost, while Method I requires that a different set of curves be prepared for each combination of profit on oil vs. gas cost. Method II appears to be more accurate and simpler to use than Method I. For Method I the value of the most economical flow rate is generated in the computational procedure, requiring an accurate determination of the slope.

3.42537 Conclusions

(1) The economical TGLR can be successfully determined in a complete gas lift system analysis whereby both the vertical and horizontal flow conduits are considered; therefore, a variable wellhead pressure occurs for changes in rates and/or gas-liquid ratios.
(2) The methods of economical slope and profit on oil vs. cost of gas injected can interrelate the most important gas lift parameters, using eco-

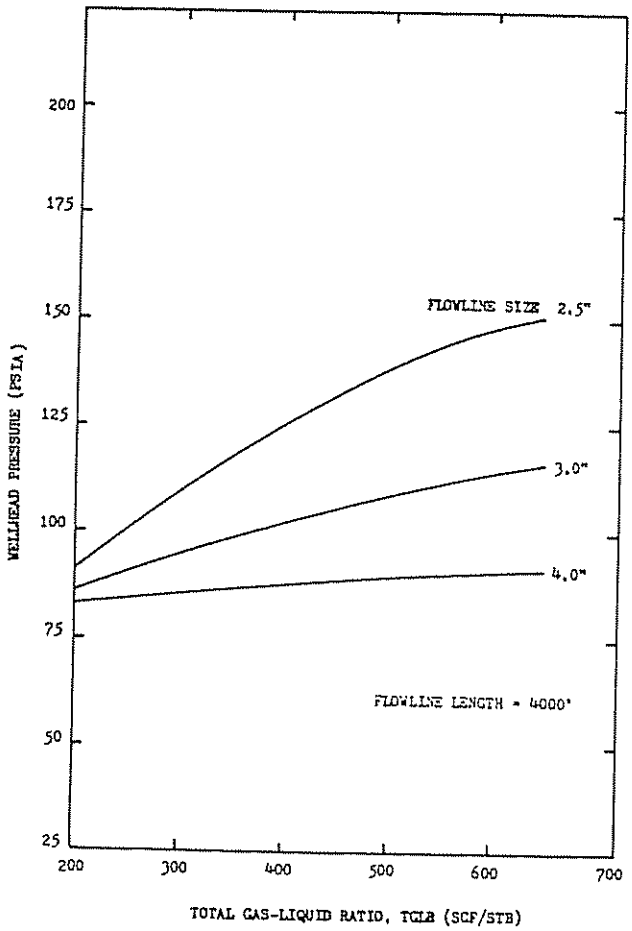


Fig. 3.440-D Effect of Flowline Size on Wellhead Pressure

nomical considerations in continuous flow gas lift design.

(3) Both methods can perform a complete analysis of the gas lift system and can detect poor performance of the gas lift well by comparing the present production with the production obtained from any of the methods. They also allow the change of any design parameter and note the effect of one parameter on the others.

3.426 Optimizing continuous flow gas lift systems

3.4261 Introduction

For many years the term "optimum gas injection" meant that volume of gas injected whereby the production rate was maximum. Today, the use of gas has become very important and the optimum should be assigned to the point where an incremental expense for gas injection is equal to some percentage of the incremental revenue produced at this injection rate. The determination of the optimum gas distribution for each gas lift well in a reservoir has particular significance when the gas available is insufficient to supply all the wells; therefore, the gas should be allocated to those wells which produce the most oil.

An optimization study determines the optimum gas injection rate which must be supplied to a group of continuous flow gas lift wells where the gas supply is restricted to a certain gas volume. To generate what is called by Mayhill "the performance curve" of a gas lift well at a particular time, a reliable correlation which will accurately predict pressure losses in two-phase vertical flow must be used.[12] In this work the modified correlation of Hagedorn and Brown[14] was used, which Lawson and Brill showed to be one of the most accurate in vertical multiphase flow.[13]

A computer program was developed for this purpose. The computer program can determine the maximum total liquid production when there is no gas availability limitation and when there is a limited volume of gas available. It can also detect poor performance of gas lift wells when the actual production is compared to the ideal production given by the performance curve.

Jones and Brown, in analyzing a 20-well platform, increased the production of the system.[14] They suggested the need for an optimization program. They pointed out that the critical design features of a continuous gas lift system are tubing size, gas lift pressure, gas volumes, downhole valve design, and surface back pressure. Correcting any one of these will improve the system. However, optimization of all five will yield the most efficient system. The program can study the effects of variation of all well parameters.

Optimizing an individual well or group of gas lift wells was an overwhelming task when done manually. Today, optimization procedures can be programmed for solution by digital computer and optimum gas injection rates can be accurately predicted. As the gas injection rate increases, the production increases until the maximum is reached. Further increases in the gas injection rates will cause a decrease in the production (see Fig. 3.441). Additional gas continues to decrease the overall pressure gradient up to a point after which any additional gas causes an increase in the pressure gradient. This is due to additional friction and possibly kinetic energy effects.

For example, in Figure 3.441 this occurs at a gas rate of approximately 6,300 Mscf/D. This represents an injection G/L of 1,000 scf/B. Ability to predict pressure losses in two-phase vertical flow is essential, and it was not until accurate correlations for predicting two-phase flow were developed that optimum gas injection could be predicted accurately. Until that time optimization was performed in the field by trial and error procedures. In many cases the maximum production was being obtained, but in others gas was being needlessly wasted in some wells and robbed from others.

Optimization may be done in two ways:

(1) The first is a procedure to obtain maximum production from a group of wells that has an allocated total amount of gas to be used. This is the case, for example, on an offshore platform where the lift gas from the compressor system can only be used for the wells on that specific platform.

(2) The second is based on an economic optimization where the cost of an incremental amount of gas is equal to the profit from the incremental increase in production. This is the case when new compressors are being considered or the gas available can be readily routed to another profitable source.

In practice both approaches may be close to the same total production for a group of wells or may differ considerably depending upon the availability and cost of gas.

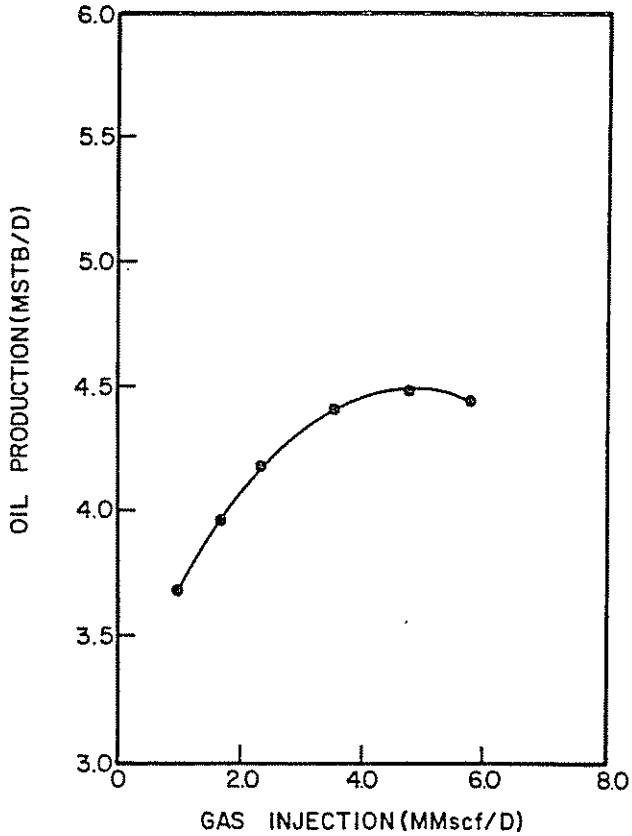


Fig. 3.441 Continuous Flow Gas-Lift Performance Curve

3.4262 Continuous flow gas lift design for optimization based on maximum rate

This section covers optimization based on maximum rates obtainable from a system of wells. To design a continuous-flow gas lift installation that may be optimized requires reliable information about the mechanical conditions of the well, physical properties of the fluid, available gas pressure and capacity of the compressor to supply the required volume of gas, and the prediction of the reservoir's future behavior.

To determine the point of injection, use the standard procedure described in Section 3.4231. For a required rate, given productivity index and static bottom hole pressure, calculate the flowing bottom hole pressure. Using a good multiphase flow correlation, obtain a flowing gradient curve representing the desired rate and the formation gas-liquid ratio for flow in the tubing string below the point of gas injection. The surface operating pressure is 100 psi less from the known available surface injection pressure. Extend a gas gradient line down the hole until it intersects the flowing gradient line. The intersection point is the point of balance between tubing and casing pressure. To insure gas entry and to insure that the well can be unloaded, calculate the point of injection by returning up the hole at a point where the tubing pressure is 100 psi lower. With the known wellhead pressure, a search is made for the appropriate gas-liquid ratio which will connect these two points. Calculate the injection gas-liquid ratio by subtracting the formation gas-liquid ratio from the total gas-liquid ratio.

There are two cases to be considered: constant wellhead pressure and variable wellhead pressure. For wells with long flow lines, the wellhead pressure will vary considerably depending upon the length and size of lines. For wells on offshore platforms with very short flow lines, the wellhead pressure may change only slightly with rate. Computational procedures are given for both cases.

3.4263 Constant wellhead procedure

The continuous gas lift program determines the injection point with an assumed injection pressure sufficient to lift an assumed rate from the well. The following information is input data:

Depth
Static bottom hole pressure
Productivity index
Wellhead pressure (remains constant)
Tubing inside diameter
Available pressure
Wellhead temperature
Bottom-hole temperature
Water cut
Formation gas liquid ratio
Oil API gravity
Specific gravity of gas
Specific gravity of water

Gomez proposed the following procedure to use in these computations:[15]

(I) Determine whether or not the well will flow naturally and, if so, at what rate. For example, one well in a study of 13 wells flowed 3,400 B/D naturally but would produce about 6,000 B/D by gas lift.

Procedure: (constant wellhead pressure)

(1) Assume a value of q_L
(2) Calculate the flowing bottom-hole pressure for the assumed liquid flowrate and for the given static bottom-hole pressure and productivity index
(3) Compute the wellhead pressure for the assumed rate and corresponding P_{wf} of Steps 1 and 2
(4) The calculated wellhead pressure obtained in Step 3 is compared to the given wellhead pressure. If the difference between them is less than or equal to 1 psi, then the assumed q_L is the natural flowing rate of the well. If the calculated P_{wh} does not reach the surface, the well will not flow even for $P_{wh} = 0$. If it falls between 0 and a set P_{wh}, then it will flow, but at a lesser rate.
(5) If the difference is greater than 1 psi, assume another liquid flowrate and repeat the procedure from Step 2 to Step 4 until convergence is obtained.

(II) After the natural flowing conditions of the well are calculated, gas is injected into the well to determine the optimum liquid flowrate for the given GLR injected. Increments of ΔGLR are chosen. These increments may vary from 50-300 scf/B. The procedure is as follows:

(1) From the known surface-operating pressure less 100 psi, calculate the casing gas pressure at the bottom
(2) Calculate the total GLR, adding the formation GLR and an injection GLR of 200 scf/STB or some other value
(3) Assume a liquid flowrate
(4) Calculate the flowing bottom-hole pressure for the assumed liquid flowrate and for the given static bottom-hole pressure and productivity index, or obtain the flowing pressure from an appropriate inflow performance curve
(5) Subtract the calculated flowing bottom-hole pressure from the calculated casing gas pressure at the bottom

The result in Step 5 has two possibilities which are described separately.

POSSIBILITY A

If the result in Step 5 is positive or zero, the casing gas pressure at the bottom is greater than or equal to the flowing bottom-hole pressure; therefore, the injection point is located at the bottom. Proceed as follows:

(6) Using the total GLR, compute the wellhead pressure using a two-phase multiphase flow correlation.
(7) Compare the calculated wellhead pressure obtained in Step 6 to the given wellhead pressure. If the difference between them is greater than 1 psi, assume another flow rate and repeat the procedure from step 4.
(8) If the difference is less than or equal to 1 psi, then the assumed liquid flow rate is the optimum flowrate for the total GLR.
(9) Increase the total GLR by increments (say 200 scf/STB).
(10) Repeat the procedure from Step 3 to Step 8. Repeat the procedure as many times as the total GLR is increased until the last optimum liquid flow rate calculated at time n is less or equal to the optimum flow rate calculated at time n-1.

POSSIBILITY B

If the result in Step 5 is negative, then the flowing bottom-hole pressure is greater than the casing gas pressure at the bottom, and the following procedure is utilized.

(6) Divide the total depth into increments ΔL of approximately 200 ft
(7) Compute the flowing pressure at intervals ΔL using the formation GLR. Simultaneously, compute the casing gas pressure at the same depth up the hole until the casing gas pressure is greater than or equal to the flowing tubing pressure. A linear interpolation gives the point of gas injection where the flowing gradient curve intersects the gas column gradient.
(8) Using the total GLR, compute the pressure traverse above the point of injection and determine the wellhead pressure.
(9) Compare the wellhead pressure calculated in Step 8 with the given wellhead pressure. If the difference between them is greater than 1.0 psi, assume another flow rate and repeat the procedure from Step 4.
(10) If the difference is less than or equal to 1.0 psi, then the assumed liquid flow rate is the optimum flow rate for the total GLR.
(11) Increase the total GLR in increments (say 200 scf/STB)
(12) Repeat the procedure from Step 3 to Step 10. Repeat the procedure as many times as the total GLR is increased until the last optimum liquid flow rate calculated at time n is less or equal to the optimum flow rate calculated at time n-1.

3.42631 Description of curve fitting and optimization procedure

3.426311 Introduction

The following procedure was given by Gomez.[14] To optimize the system, the set of data generated into gas-lift performance curves as explained previously can be fitted to polynomial equations, and the set of equations used as input data in the program of optimization. Other methods may be used.

3.426312 Curve fitting

Data is commonly represented in engineering work by functional relationships that are given as x-y pairs. When the computer operates on this data, it is convenient to express $y = f(x)$, when $f(x)$ is a polynomial. Also, when experimental data are gathered, errors of measurement may be involved which need to be smoothed. Do this by fitting an equation in the form of a polynomial to the data. The method of least squares is commonly used for this purpose. A program has been developed which uses this method to fit the second-degree polynomial $Q_o = C_1 + C_2 \times Q_g + C_3 \times Q_g^2$ to the particular set of data.

The least squares method finds an equation so the sum of the squares of the individual residual is minimized. A residual is the difference between the actual Q_o value and the Q_o value computed with the fitted polynomial. Form a set of $M + 1$ equations in $M + 1$ unknowns by taking derivatives of the sum of the squares of the residuals with respect to each of the unknown C_i coefficients. Then solve the set of equations for the C_i's using Gauss Elimination. Use the resulting polynomial to calculate each Q_o value and calculate a percent error for each set of data pairs. Calculate an average percent error and the standard deviation of the percent errors.

3.426313 Optimization procedure

The program starts the optimization by calling a subroutine to calculate the maximum rate where the slope of the gas-lift performance curve is equal to zero for each well. If the summation of all these maximums is less than or equal to the total gas available, then the maximum gas injection rate is allocated to each well to yield the maximum production. Otherwise, if the summation of all these maximums exceed the total gas

available, then an equal amount of gas is injected into each well. Then another subroutine is called to calculate the production for each well. The program scans the wells to determine which well produced the largest amount of oil, and an increment of gas is injected into that well.

If that well still yields the largest increment of oil, it will receive a third increment of gas. If not, whichever of the other wells that yields the second largest increment in the production for the first amount of gas injected will receive a second increment of gas. This process is continued until all the available gas is used. The output will give the oil production and gas injection for each individual well and the total gas injection and total production.

TABLE 3.437
GIVEN WELL DATA

Well D-1	
1. Inclined depth	11,083.0 ft
2. Vertical depth	9,029.0 ft
3. Distance where diameter changes (measured from bottom)	1,843.0 ft
4. Pipe diameter used in the lower portion of the well	3½ in.
5. Pipe diameter used in the upper part of the well	4½ in.
6. Formation gas-liquid ratio	322.0 scf/STB
7. Wellhead pressure	144.7 psia
8. Static bottom hole pressure	3,014.7 psia
9. Productivity index	5.68 psi/ft
10. Operating surface pressure	1,140.00 psig
11. Wellhead temperature	120.00°F.
12. Bottom hole temperature	180.0°F.
13. Water cut	0.20%
14. Gas specific gravity	0.76
15. Oil API° gravity	34.0
16. Specific gravity of the water	1.03
17. Gas injection	3,148.0 Mscf/d
18. Oil production	5,513.0 STB/d

TABLE 3.438
GIVEN WELL DATA

Well D-2	
1. Inclined depth	11,501.0 ft
2. Vertical depth	9,185.0 ft
3. Distance where diameter changes (measured from bottom)	1,261.0 ft
4. Pipe diameter used in the lower portion of the well	3½ in.
5. Pipe diameter used in the upper part of the well	4½ in.
6. Formation gas-liquid ratio	553.0 scf/STB
7. Wellhead pressure	164.7 psia
8. Static bottom hole pressure	3,114.7 psia
9. Productivity index	1.28 psi/ft
10. Operating surface pressure	1,100.0 psig
11. Wellhead temperature	120.0°F.
12. Bottom hole temperature	180.0°F.
13. Water cut	57.5%
14. Gas specific gravity	0.76
15. Oil API° gravity	34.0
16. Specific gravity of the water	1.03
17. Gas injection	1,622.0 Mscf/d
18. Oil production	830.0 STB/d

TABLE 3.439
WELL D-1
CONTINUOUS FLOW GAS-LIFT PERFORMANCE

Liquid rate (STB/D)	Oil rate (STB/D)	GLR injection (scf/STB)	Gas volume injection (Mscf/D)	Injection pressure (psia)	Injection point (ft)
5,346.70	5,336.00	250.00	1,336.67	1,447.10	8,594.42
6,068.41	6,056.28	500.00	3,034.21	1,468.43	9,178.41
6,284.23	6,271.66	750.00	4,713.17	1,474.85	9,353.37
6,307.93	6,295.31	1,000.00	6,307.93	1,475.56	9,372.63
6,254.96	6,242.45	1,250.00	7,818.70	1,473.98	9,329.65

TABLE 3.440
WELL D-2
CONTINUOUS FLOW GAS-LIFT PERFORMANCE

Liquid rate (STB/D)	Oil rate (STB/D)	GLR injection (scf/STB)	Gas volume injection (Mscf/D)	Injection pressure (psia)	Injection point (ft)
2,175.31	924.51	250.00	543.83	1,495.11	11,399.26
2,307.60	980.73	500.00	1,153.80	1,411.89	11,501.00
2,358.04	1,002.17	750.00	1,768.53	1,372.48	11,501.00
2,386.67	1,014.33	1,000.00	2,386.67	1,350.11	11,501.00
2,404.67	1,021.98	1,250.00	3,005.83	1,336.05	11,501.00
2,415.65	1,026.74	1,500.00	3,623.78	1,327.31	11,501.00
2,421.31	1,029.06	1,750.00	4,237.29	1,323.05	11,501.00
2,424.56	1,030.44	2,000.00	4,849.13	1,320.51	11,501.00
2,424.56	1,030.44	2,250.00	5,455.27	1,320.51	11,501.00

3.42632 Field example

Gomez conducted a field study of 13 wells directionally drilled from an isolated offshore platform. Typical data for two of these wells is shown in Tables 3.437 and 3.438. Continuous flow gas lift performance curves were then prepared for all wells. Tables 3.439 and 3.440 show the tabulated results from the well data of Tables 3.437 and 3.438, respectively.

These data were then plotted on Figures 3.442 and 3.443 and are shown as actual data points.

The calculated values of flow rates vs. gas volume requirements are noted in Tables 3.441 and 3.442.

The corresponding equations which give the best curve fits for the data are given at the bottom of Tables 3.437 and 3.438. The solid lines on Figures 3.442 and 3.443 represent these equations and the curve for this particular well. In Tables 3.437 and 3.438 the percent of errors is small; hence, a good solution can be expected by using the equations describing each well's performance. Some "fits" are excellent, as noted in Figures 3.444 and 3.445.

3.42633 Discussion of results

Gomez conducted a field study of 13 wells on an offshore platform. The operating conditions in the field for the group of 13 wells used 30 MMscf/D to produce 32,168 STBO/D.

Table 3.439 shows the natural flowing conditions of the wells. All the wells in the system produce by natural flow except Wells D-17 and D-23, which have lower formation GLR values. The system can deliver 17,200 STB/D by natural flow. But when 30 MMscf/D is

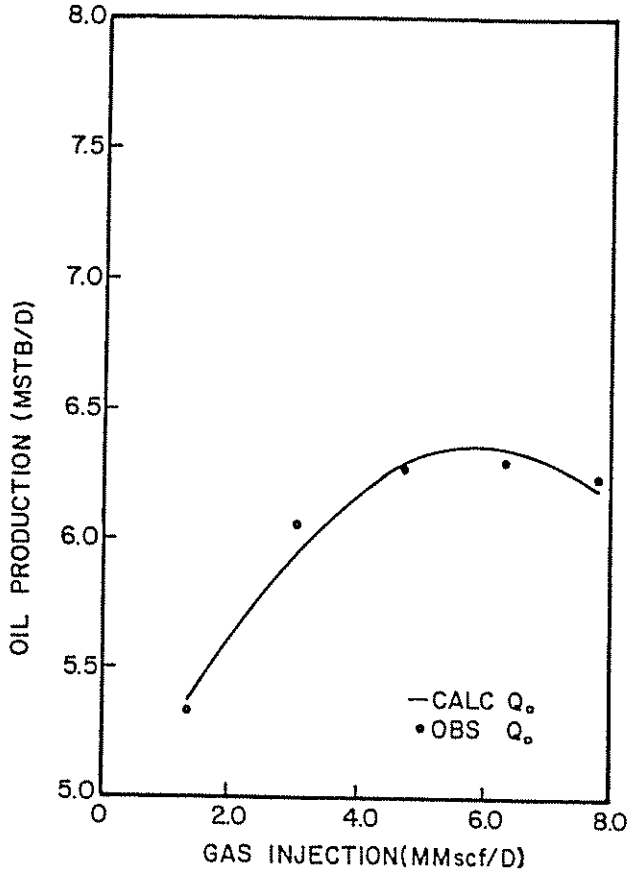


Fig. 3.442 Continuous Flow Gas-Lift Performance Curve (Well D-1)

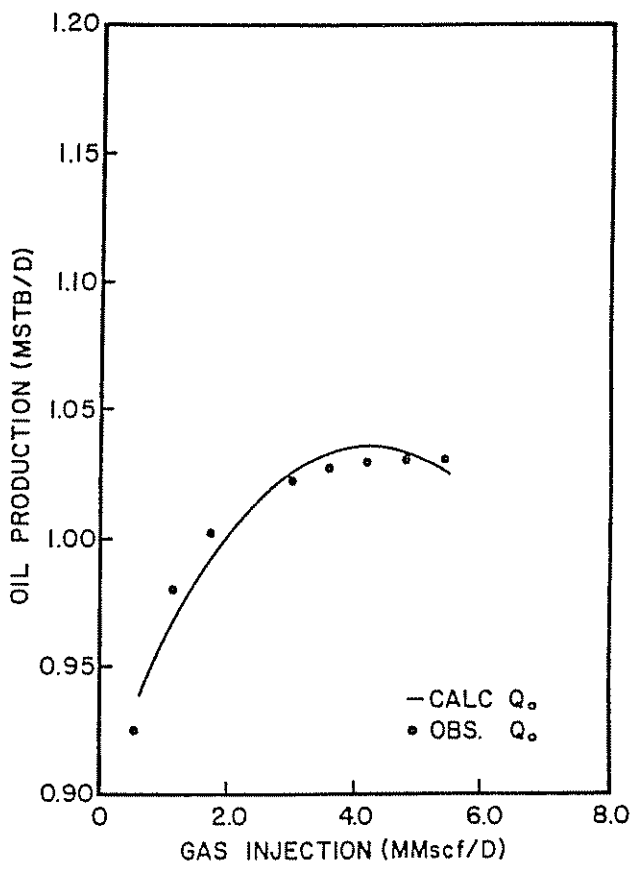


Fig. 3.443 Continuous Flow Gas-Lift Performance Curve (Well D-2)

injected proportionally into the wells, the oil production is increased to 32,168 STB/D as shown in Table 3.444.

In this study, the same group of wells was placed on continuous flow gas-lift and was optimized for oil production rates possible using from 10 MMscf/D to 70 MMscf/D of gas available.

Table 3.445 shows that when the capacity of the compressor is 10 MMscf/D, the system can deliver 30,440 STBO/D; but when 20 MMscf/D is injected into the wells, the total oil production rate will be 33,483 STB/D. This represents an increase of 3,043 STBO/D. Similarly, when 30 MMscf/D of gas is injected, the oil production rate will be 35,360 STB/D. This represents an increase in the oil production rate of 1,877 STB/D above the oil production rate given by the 20 MMscf/D of gas injected. Notice that as the gas injection increases above this point, the oil production rate increases, but at a slower rate. This is because the system is almost producing at the maximum capacity with 30 MMscf/D. In other words, each well is producing almost at the maximum point in the performance curve with 30 MMscf/D. By increasing the capacity of the compressor from 30 MMscf/D to 40 MMscf/D, the increase in oil production rate will be 982 STB/D.

Similarly, when the capacity of the compressor is increased from 40 MMscf/D to 50 MMscf/D, the increase in oil production rate will be only 362 STB/D, and the maximum oil production for the system is reached when 58.098 MMscf/D of gas is injected. Injecting an additional 8.098 MMscf/D represents an increase of only 84 STBO/D.

According to this optimization study, the system which is presently producing 32,168 STBO/D could be increased to 35,360 STBO/D by optimizing the gas volume available of 30 MMscf/D as prorated to each well. However, various mechanical malfunctions in the system could account for this difference. Apparently, these wells are being produced at near their optimum rates for this system. The results of this study agree with the actual performance of the field. However, an oil production rate of 33,483 STB/D can be obtained if 20 MMscf/D of gas is injected, when the gas is redistributed in the optimum manner proposed by this study by increasing

TABLE 3.441
WELL D-1
CURVE FITTING RESULTS

Obs. Q_g (Mscf/D)	Obs. Q_o (STB/D)	Calc. Q_o (STB/D)	Residual (STB/D)	Percent error (%)
1,336.67	5,336.0000	5,375.1211	39.1211	.7332
3,034.20	6,056.2813	5,968.8906	−87.3906	−1.4430
4,713.16	6,271.6602	6,293.3672	21.7070	.3461
6,307.92	6,295.3086	6,359.5547	64.2461	1.0205
7,818.69	6,242.4492	6,204.7500	−37.6992	−.6039

The equation which fits the observed values of Q_o, Q_g is:

$$Q_o = 4719.6 + 0.55241 \times 10^{-3} \times Q_g - 0.46357 \times 10^{-10} \times Q_g^2$$

Avg. % error = 0.0106
Standard deviation = 0.9105

TABLE 3.442
WELL D-2
CURVE FITTING RESULTS

Obs. Q_g (Mscf/D)	Obs. Q_o (STB/D)	Calc. Q_o (STB/D)	Residual (STB/D)	Percent error (%)
543.82	924.5100	938.2566	13.7466	1.4869
1,153.79	980.7300	968.1631	−12.5669	−1.2814
1,768.52	1,002.1699	992.7537	−9.4163	−.9396
2,386.66	1,014.3301	1,011.8633	−2.4668	−.2432
3,005.83	1,021.9800	1,025.3574	3.3774	.3305
3,623.78	1,026.7400	1,033.1899	6.4500	.6282
4,237.29	1,029.0601	1,035.3975	6.3374	.6158
4,849.12	1,030.4399	1,032.0725	1.6326	.1584
5,455.27	1,030.4399	1,023.3367	−7.1033	−.6893

The equation which fits the observed values of Q_o, Q_g is:
$Q_o = 906.97 + 0.615431 \times 10^{-4} \times Q_g - 0.737116 \times 10^{-11} \times Q_g^2$
Avg. % error = 0.0074
Standard deviation = 0.8285

the gas injection rates in wells D-1, D-3, D-7, D-12, and D-17, and decreasing the gas injection rates in wells D-2, D-4, D-5, D-8, D-11, D-16, D-20, and D-23.

On the other hand, since there is 30 MMscf/D of gas available to inject into the wells, the oil production rate of the system will be 35,360 STBO/D. This represents an increase of 3,192 STBO/D over the present production rate of 32,168 STBO/D if the gas is allocated into the wells in the manner shown in Table 3.445 which is the optimum distribution. This will yield the maximum oil production for the availabe gas capacity.

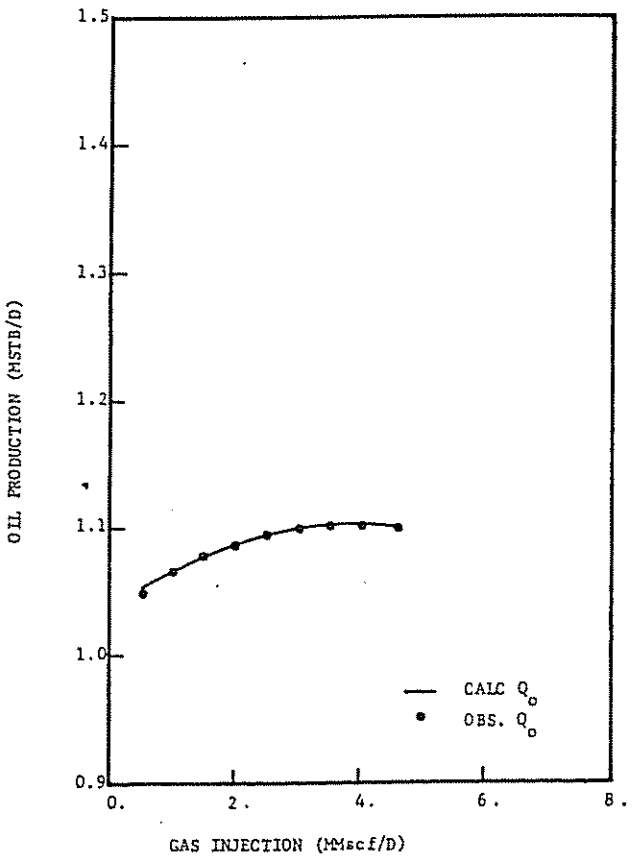


Fig. 3.444 Continuous Flow Gas-Lift Performance Curve (Well D-8)

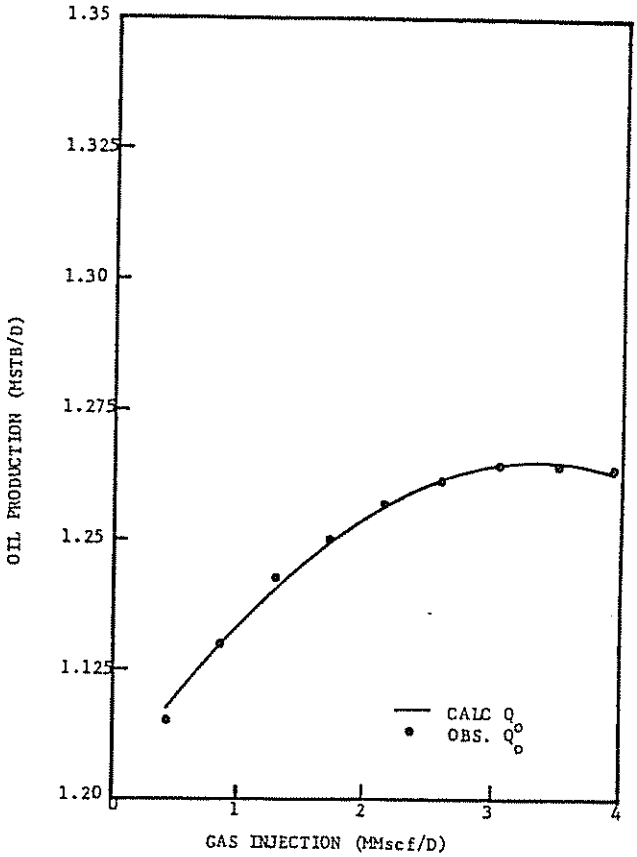


Fig. 3.445 Continuous Flow Gas-Lift Performance Curve (Well D-16)

The study made on this group of wells shows that near optimum conditions now exist on the platform although another 3,192 STBO/D could be obtained if there are no mechanical problems. This represents a sizeable increase in revenue.

From field experience, it is known that many gas lift systems are not in as good a position as this study showed for the thirteen-well system. Many systems are operating far from the optimum. A good gas lift engineer assigned to a field can normally increase the production considerably.

3.4264 Variable wellhead solution

For the variable wellhead solution, the engineer is confronted with surface flowlines and a fixed separator pressure. As either the gas or liquid rate changes, so does the wellhead pressure.

3.42641 Computational procedure

Similar to the constant wellhead solution, the continuous flow gas-lift program for a variable wellhead pressure is designed to allow the determination of the injection gas volume and injection point with an assumed injection pressure sufficient to lift an assumed rate from the well. The following information is required as input data: well depth, flowline length, separator pressure, tubing size, flowline size, static reservoir pressure, productivity index of the well, tubing flowing pressure, "kick-off" and available pressure, wellhead and bottom temperatures, water cut, gas-liquid ratio,

TABLE 3.443
NATURAL FLOWING CONDITIONS

Well no.	Liquid rate (STB/D)	Water rate (STB/D)	Oil rate (STB/D)	Formation GLR (scf/STB)	Flowing bottom hole pressure (psia)	Flowing wellhead pressure (psia)
D-1	3,433.45	6.86	3,426.58	322.0	2,410.22	144.78
D-2	1,829.51	1,051.97	777.54	553.0	1,685.40	164.72
D-3	1,479.27	36.98	1,442.29	288.0	2,425.35	144.66
D-4	4,075.83	652.13	3,423.70	484.0	2,042.11	145.12
D-5	765.65	393.54	372.10	280.0	2,649.05	164.64
D-7	628.55	74.17	554.38	203.0	2,640.56	139.88
D-8	1,826.29	834.61	991.68	779.0	1,407.79	154.69
D-11	1,065.94	10.66	1,055.28	282.0	2,454.39	134.48
D-12	2,654.52	122.11	2,532.42	386.0	2,259.84	159.64
D-16	1,629.42	451.35	1,178.07	874.0	1,282.62	159.77
D-17	0.00	0.00	0.00	131.0	—	—
D-20	1,449.03	2.90	1,446.13	401.0	2,177.18	184.74
D-23	0.00	0.00	0.00	101.0	—	—
Total	20,837.46	3,637.28	17,200.17			

TABLE 3.444
COMPARISON OF RESULTS

	Natural flowing conditions		Observed values injecting gas		Calculated values injecting gas	
Well no.	Gas injection (Mscf/D)	Oil production (STB/D)	Gas injection (Mscf/D)	Oil production (STB/D)	Gas injection (Mscf/D)	Oil production (STB/D)
D-1	0.00	3,426.58	3,148.0	5,513.0	3,148.0	5,999.14
D-2		777.54	1,622.0	830.0	1,622.0	987.40
D-3		1,442.29	2,394.0	3,835.0	2,394.0	3,672.40
D-4		3,423.70	3,240.0	4,021.0	3,240.0	4,492.11
D-5		372.10	2,489.0	1,135.0	2,489.0	1,042.30
D-7		554.38	2,148.0	2,156.0	2,148.0	2,564.90
D-8		991.68	1,471.0	775.0	1,471.0	1,077.70
D-11		1,055.28	2,106.0	2,462.0	2,106.0	2,422.52
D-12		2,532.42	1,799.0	3,599.0	1,799.0	4,011.24
D-16		1,178.07	1,956.0	1,303.0	1,956.0	1,253.94
D-17		0.00	2,698.0	2,487.0	2,698.0	2,691.10
D-20		1,446.13	2,783.0	3,186.0	2,783.0	3,117.79
D-23	0.00	0.00	2,437.0	866.0	2,437.0	888.67
Total	0.00	17,200.17	30,291.0	32,168.0	30,291.0	34,221.50

oil API gravity, and specific gravity of gas and water.

Hong studied 6 wells and the following discussion on the variable wellhead solution is essentially the same as his presentation.[16] He used the vertical flow correlation of Hagedorn and Brown[4] and the horizontal flow correlation of Dukler.[17]

The following procedure for a variable wellhead pressure is used in these computations:

(1) Determine whether or not the well will flow naturally and at what rate.
(2) Assume a value of flow rate greater than the natural flow rate.
(3) Calculate the flowing bottom hole pressure and compare this with casing gas pressure at depth. If the casing pressure is greater than the flowing bottom pressure, set the point of injection at the well depth.
(4) If the casing pressure is less than the flowing bottom hole pressure, increase the GLR by some increment (50-200 scf/B).
(5) Knowing the separator pressure and using the GLR of Step 4, calculate the wellhead pressure using the combination of Dukler's[16] friction factor and Eaton's[7] hold-up correlation.
(6) Starting with the wellhead pressure, calculate the tubing pressure in increments down the well using the Hagedorn and Brown[4] correlation. Compare the tubing pressure with the casing gas pressure at each incremental depth. The differential between these two pressures is determined for intervals of 200 feet down the hole. A linear interpolation using a pressure differential of 100 psi gives the point of gas injection.
(7) Starting from the point of gas injection, calculate the pressure traverse below the point of gas injection using the formation GLR.
(8) Extend this traverse downwards until reaching the total depth and note the flowing bottom hole pressure.
(9) Compare the flowing bottom hole pressure from Step 8 with the flowing bottom pressure obtained from the linear PI equation. If the difference between these two values is greater than $\pm$1 psi, calculate a new flow rate by the Eqn. $q_L = (P_R - P_{wf})$ (PI) where P_{wf} is the value calculated by using the correlation. This case assumes a constant PI, but an inflow curve may be used.
(10) Repeat the procedure from Step 3 until the differential pressure is less than 1 psi.
(11) If the differential pressure is less than or equal to 1 psi, then the assumed liquid flow rate is the correct flow rate for the assumed GLR above the point of injection.
(12) Increase the total GLR by some increment (Hong used 150 scf/B).
(13) Repeat procedure from Step 2 to Step 9 until the flow rate will not increase with an increase in the GLR.

TABLE 3.445
GAS ALLOCATION AND OIL PRODUCTION FOR DIFFERENT GAS VOLUMES AVAILABLE

	Gas volume = 10 MMscf/D		Gas volume = 20 MMscf/D		Gas volume = 30 MMscf/D		Gas volume = 40 MMscf/D	
Well no.	Gas injection (Mscf/D)	Oil production (STB/D)	Gas injection (Mscf/D)	Oil production (STB/D)	Gas injection (Mscf/D)	Oil production (STB/D)	Gas injection (Mscf/D)	Oil production (STB/D)
D-1	1,890.0	5,598.02	3,360.0	6,052.29	4,450.0	6,259.76	5,330.0	6,346.93
D-2	10.0	907.58	10.0	907.58	10.0	907.58	220.0	920.15
D-3	2,100.0	3,567.80	3,090.0	3,873.90	3,830.0	4,014.88	4,420.0	4,073.46
D-4	10.0	3,827.31	1,140.0	4,145.93	2,530.0	4,410.85	3,660.0	4,522.96
D-5	10.0	735.13	10.0	715.13	790.0	854.54	3,160.0	1,089.15
D-7	1,400.0	2,323.24	2,330.0	2,611.29	3,020.0	2,743.10	3,580.0	2,798.77
D-8	10.0	1,037.07	10.0	1,037.07	10.0	1,037.07	10.0	1,037.07
D-11	200.0	1,877.83	1,610.0	2,314.45	2,670.0	2,516.64	3,520.0	2,600.61
D-12	1,450.0	3,886.26	2,650.0	4,257.71	3,540.0	4,427.07	4,250.0	4,497.34
D-16	10.0	1,203.37	10.0	1,203.37	10.0	1,203.37	10.0	1,203.37
D-17	1,940.0	2,439.86	3,060.0	2,786.32	3,890.0	2,944.21	4,560.0	3,010.28
D-20	960.0	2,591.17	2,360.0	3,024.51	3,400.0	3,222.52	4,230.0	3,304.77
D-23	10.0	445.15	360.0	533.92	1,850.0	818.26	3,050.0	937.09
Total	10,000.0	30,439.77	20,000.0	33,483.45	30,000.0	35,359.84	40,000.0	36,341.94

TABLE 3.445 (CONTINUED)

	Gas volume = 50 MMscf/D		Gas volume = 60 MMscf/D		Gas volume = 70 MMscf/D	
Well no.	Gas injection (Mscf/D)	Oil production (STB/D)	Gas injection (Mscf/D)	Oil production (STB/D)	Gas injection (Mscf/D)	Oil production (STB/D)
D-1	5,730.0	6,362.81	5,958.15	6,365.23	5,958.15	6,365.23
D-2	2,760.0	1,020.68	4,174.59	1,035.43	4,174.59	1,035.43
D-3	4,700.0	4,084.55	4,848.70	4,086.07	4,848.70	4,086.07
D-4	4,170.0	4,543.23	4,463.06	4,546.35	4,463.06	4,546.35
D-5	4,250.0	1,132.29	4,857.38	1,138.63	4,857.38	1,138.63
D-7	3,840.0	2,809.00	3,978.40	2,810.40	3,978.40	2,810.40
D-8	1,510.0	1,078.52	3,783.15	1,102.14	3,783.15	1,102.14
D-11	3,910.0	2,615.85	4,121.00	2,618.00	4,121.00	2,618.00
D-12	4,580.0	4,510.42	4,762.30	4,512.32	4,762.30	4,512.32
D-16	1,470.0	1,245.14	3,384.24	1,265.02	3,384.24	1,265.02
D-17	4,860.0	3,022.10	5,032.31	3,023.91	5,032.31	3,023.91
D-20	4,620.0	3,320.19	4,829.82	3,322.34	4,829.82	3,322.34
D-23	3,600.0	958.70	3,900.70	961.78	3,900.70	961.78
Total	50,000.0	36,703.46	58,097.70	36,787.59	58,097.70	36,787.59

3.42642 Description of the optimization procedure

3.426421 Introduction

The basis of gas lift optimization is the same as for a constant wellhead pressure—as the gas injection rate into a well is increased in equal increments of lift gas, an incremental increase of oil production from the well occurs. The oil production rate eventually reaches a maximum as the injection gas rate is increased and then starts to decrease with further increases in gas. When the availability of gas is limited, the optimum gas injection rate of a well is usually less than the rate that yields maximum oil production. For a variable wellhead pressure, the gas volume for maximum rate can be considerably less than that gas required to create the minimum gradient (lowest pressure) in the tubing string.

To optimize the system, the continuous flow gas lift program previously described is used to generate the oil production rates and gas injection rates. The oil production rate as a function of gas injection rate can be determined and is plotted as the relationship between oil production rate and gas injection rate as shown in Figure 3.446. The basis of optimization, as stated in the first paragraph of this chapter, is readily apparent. For example, Figure 3.446 shows three typical gas lift performance curves for a gas lift well with inside tubing diameter equal to 3.958 inches and three different flowline diameters. Notice that as the gas input is increased, the production rate increases, flattens, then passes through a maximum. Beyond the maximum, any further increase in gas input causes a decrease in production. The optimum gas input is a point to the left of the maximum when the availability of gas is limited.

The oil production rate is a function of gas injection

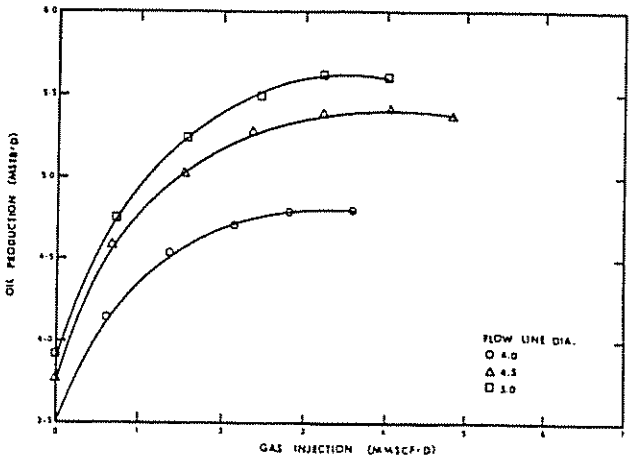


Fig. 3.446 Gas-Lift Performance Curve With 3.958-Inch Tubing Diameter (Well 1)

rate, and Hong used a cubic spline interpolating scheme to interpolate the data pair, namely, oil production rate and gas injection rate.

3.426422 Cubic spline interpolating scheme

A cubic spline interpolating scheme is a function which approximates a continuous and differentiable function on the interval (a,b) in a piecewise fashion, using third degree interpolating polynomials over a nonoverlapping subinterval of (a,b).

Following is the Newton's divided difference interpolation formula for the cubic spline interpolating scheme:

$$S(X) = S(X_i) + (X - X_i)S(X_i, X_{i+1}) + (X - X_i)(X - X_{i+1})S(X, X_i, X_{i+1}) \quad (3.413)$$

Where:

$i = 1, 2, 3, \ldots\ldots\ldots n.$

X is in (X_i, X_{i+1})

$$S(X, X_i) = 5(X_i) + \tfrac{1}{2}(X - X_i)5''(X_i) + \frac{1}{6}(X - X_i)^2 3''(X_i, X_{i+1}) \quad (3.414)$$

$$S(X, X_i, X_{i+1}) = \tfrac{1}{2}\,3''(X_i) + \frac{1}{6}(X - X_i) + (X_{i+1} - X_i)\,5''(X_i, X_{i+1}) \quad (3.415)$$

Using the equations described above, the computer program was developed.

3.426423 Optimization procedure

Based on the logic that incremental gas injection rates result in an increased incremental oil production, a computer program can be developed. The program starts the optimization by calculating the maximum gas injection rate for each well, that is, where the slope of the gas lift performance curve is equal to zero for each well.

If the summation of all these maximum gas injection rates for all wells is less than or equal to the total gas available, then the maximum gas injection rate is allocated to each well to yield the maximum oil production. If the summation of all these maximums exceeds the total gas available, then an equal amount of gas is injected into each well. A low value of gas is initially assumed for each well. The cubic spline interpolating scheme is then used to calculate the oil production rate for each increment of gas injected.

The program scans the wells to determine which well produces the largest amount of oil with equal amounts of gas injection to each well. An increment of gas is injected into the well which yields the maximum oil production rate. The wells are scanned again to determine the well showing the most increase in oil production per increment of gas increase. It may or may not be the same well, since eventually an incremental gas increase will show very little increase in production for a particular well. Gas is then injected into the well having shown the greatest increase.

This process is continued until all of the available lift gas is utilized. The output of this program will give the oil production rate and gas injection rate for each well and the total gas injection and total oil production.

3.42643 Effect of variables in optimization of variable wellhead gas lift systems

3.426431 Introduction

Hong conducted a study on six wells to show the effect of several variables.[16] In a gas lift well some of the variables which affect production rate can be controlled and some cannot be controlled. The first category includes tubing sizes, flowline sizes, casing available pressure, and separator pressure. The second category includes the reservoir static pressure, the fluid properties, water cut and the productivity. Given a set of well conditions, any change in one of the variables will change the production rate and/or gas injection rate.

Hong selected six typical gas lift wells to show how variables such as tubing size, flowline size, separator pressure, available casing pressure and water cut, affect the gas injection rates and oil production rates.[15] The well properties chosen for one of these wells are shown in Table 3.446, Table 3.447 shows the allocation of gas for the six wells for a tubing I.D. of 3.958" and flowline I.D. of 4.0".

3.426432 Effect of tubing size and flowline size

Figure 3.446 shows gas lift performance curves for 3.958" tubing for one well. Figure 3.447 shows total gas allocation vs. production rates for the six-well system. As noted, the oil production increased rapidly with increasing tubing diameter. For example, a gas injection rate of 5 MMcf/D for the system of six wells produces 13,515.328 STB/D for a tubing of inside diameter equal to 3.5" and 14,621.355 STB/D for a tubing of inside diameter equal to 3.958"

Figure 3.448 shows the effect of flowline size on the oil production rate. For a given tubing diameter of 3.958" I.D., any increase in flowline diameter will increase the oil production rate. For example, a gas injection of 5 MMcf/D in a 4.0" diameter flowline will produce 14,621 STB/D while the 4.5" diameter flowline will produce 15,652 STB/D.

Although larger tubing sizes show an advantage in increasing the oil production, they may be prohibitive due to the casing size. Also, the income from the increased production rate may not offset the increased cost of larger tubing and flowline sizes.

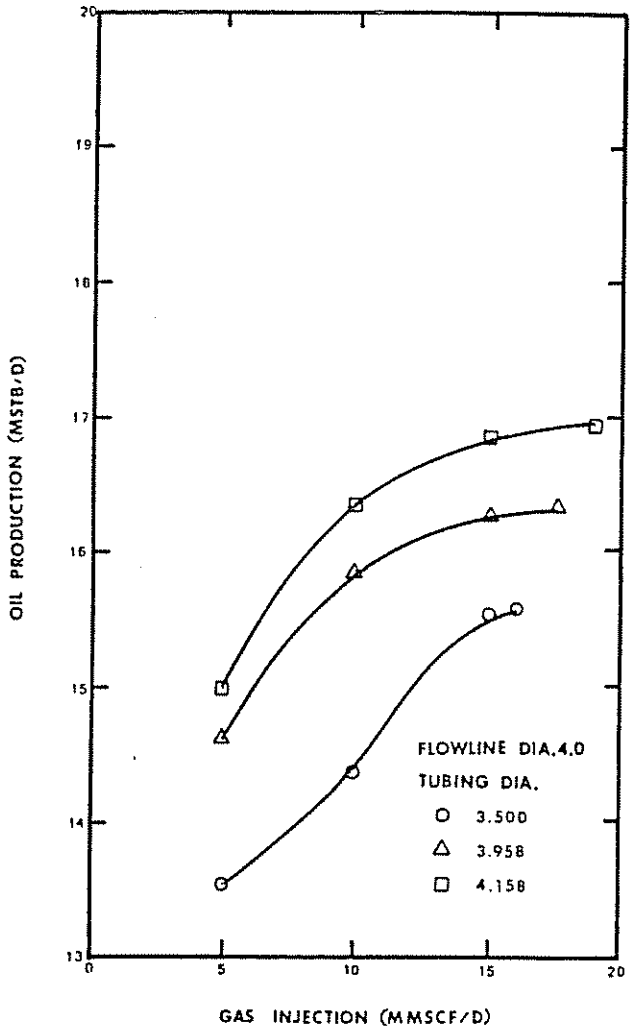


Fig. 3.447 Total Gas Allocation and Oil Production Curve Showing Effect of Tubing Diameter

3.426433 Effect of separator pressure

The same well data were used to show the effect of separator pressure on the production rate for a given tubing size of 3.958" and flowline size of 4.5" diameter. An increase in separator pressure causes an increase in the flowing wellhead pressure which, in turn, increases the flowing bottom hole pressure. A typical performance curve for Well #1 (data of Table 3.446) is shown in Figure 3.449.

Obviously production decreases with any increase of separator pressure; other conditions remaining constant. Figure 3.450 shows the effect of separator pressure on the six-well gas lift system. For example, for 5 MMcf/D of gas injected, the production varies from 15,675 STB/D for 80 psi separator pressure to 14,515 STB/D for 155 psi separator pressure.

Although the lower separator pressure can increase the oil production rate, the separator pressure may be predetermined due to gas sales line pressure required or due to the composition of the fluid with reference to a required back pressure to give the most economical separation.

TABLE 3.446
INPUT DATA FOR WELL NO. 1

1	Assume fluid production (water + oil), B/D	3,500.000
2	Total formation gas-liquid ratio, SCF/STB	288.600
3	Inside diameter of the tubing, in.	3.958
4	True vertical depth of the well, ft.	9,029.000
5	Bottom hole static pressure of the well, PSIA	3,150.000
6	Productivity index of the well, BBL/D/PSIA	5.310
7	Casing available pressure, PSIA	1,130.000
8	Surface temperature of the well, °F.	120.000
9	Bottom hole temperature of the well, °F.	180.000
10	Percentage of water cut in the well, %	.100
11	Oil gravity, API	34.000
12	Specific gravity of water	1.000
13	Separator pressure, psia	80.000
14	Total length of the horizontal pipe, ft.	6,427.000
15	Temperature of the seperator, %F.	101.000
16	Specific gravity of gas	.760
17	Diameter of the flow line, in.	4.000

3.426434 Effect of water cut

Figure 3.451 shows gas lift performance curves for Well #1 (data of Table 3.447). Figure 3.452 shows how water cut affects the rate of liquid production for the six-well system. With the gas injection rate set at 5 MMcf/D, the oil production decreases from 15,053.168 STB/D to 13,125.328 STB/D with an increase of 10%

TABLE 3.447
TUBING DIA. 3.958 FLOWLINE DIA. 4.0
GAS ALLOCATION AND OIL PRODUCTION FOR DIFFERENT GAS VOLUMES

	Gas volume 5 MMSCF/D		Gas volume 10 MMSCF/D		Gas volume 15 MMSCF/D		Gas volume 20 MMSCF/D	
Well no.	Gas injection (MSCF/D)	Oil production (STB/D)	Gas injection (MSCF/D)	Oil production (STB/D)	Gas injection (MSCF/D)	Oil production STB/D)	Gas injection (MSCF/D)	Oil production (STB/D)
1	1,200.00	4,489.387	2,100.00	4,726.637	3,200.00	4,834.047	3,200.00	4,834.047
2	1,200.00	3,049.416	2,100.00	3,276.118	3,200.00	3,360.208	3,200.00	3,360.208
3	900.00	3,029.220	1,900.00	3,284.798	3,100.00	3,380.258	3,300.00	3,389.021
4	600.00	1,265.466	2,000.00	1,567.971	2,300.00	1,597.517	3,700.00	1,622.256
5	1,000.00	1,968.668	1,700.00	2,148.093	2,400.00	2,217.095	2,700.00	2,225.470
6	100.00	819.207	200.00	836.864	800.00	886.886	1,500.00	900.00
Total	5,000.00	14,621.355	10,000.00	15,842.473	15,000.00	16,276.004	17,600.00	16,331.359

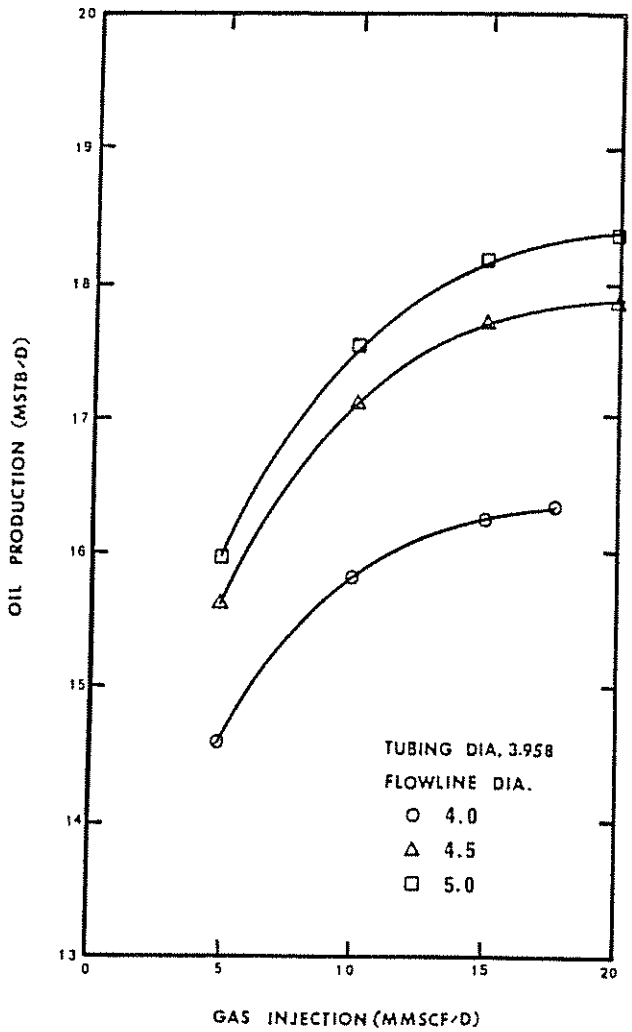


Fig. 3.448 Total Gas Allocation and Oil Production Curve Showing Effect of Flowline Size

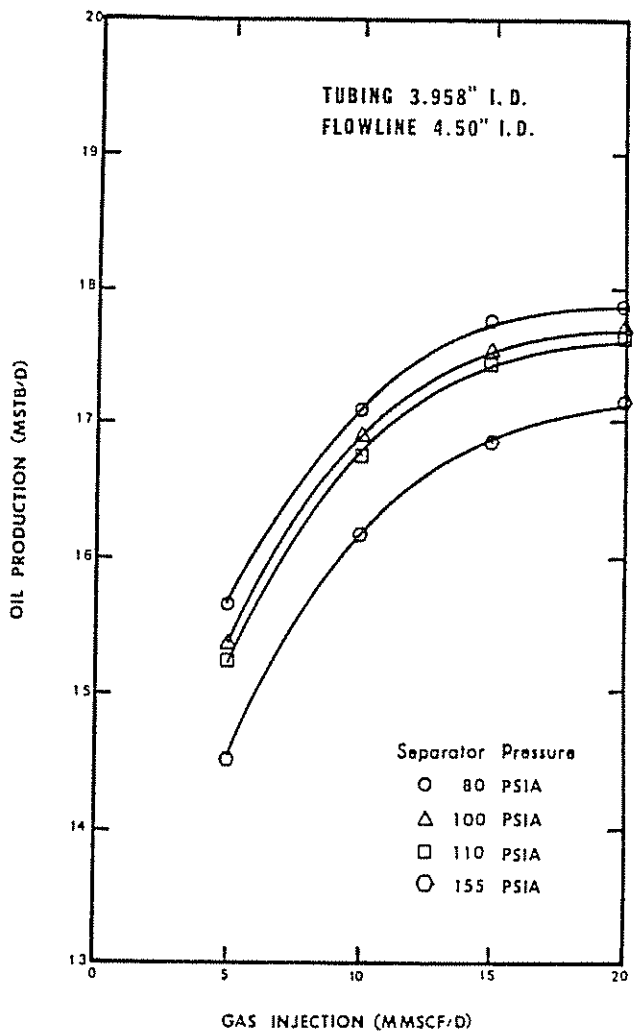


Fig. 3.450 Total Gas Allocation and Oil Production Curve Showing Effect of Separator Pressure

water cut. This is due to the influence on the density of the liquid and to the change in gas-liquid ratio below the point of injection.

3.426435 Effect of available injection pressure

In a continuous gas lift system, two factors which greatly control the production rate are injection gas pressure and injection gas volume. Figure 3.453 sho performance curves for Well #1 for different injecti pressures. Figure 3.454 shows the effect of injecti pressure and gas volume for the six-well system. As the gas injection pressure increases, the point of injection will be lower. This causes a lightening of the flowing gradient from a deeper and deeper point, thereby reducing the gas volume required and increasing the

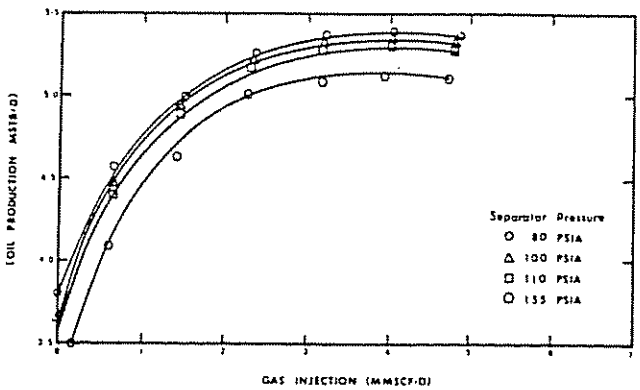


Fig. 3.449 Gas-Lift Performance Curve With 3.958-Inch Tubing Diameter and 4.5-Inch Flowline Diameter (Well 1)

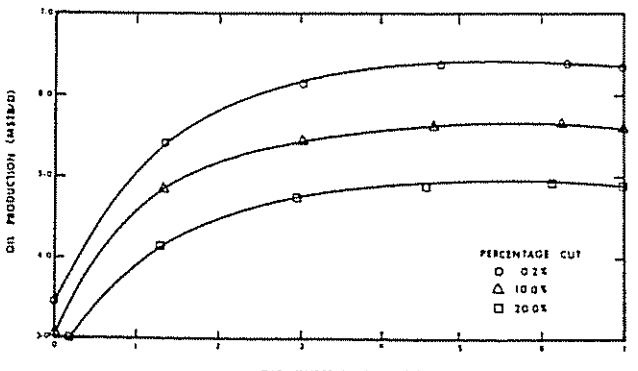


Fig. 3.451 Gas-Lift Performance Curve With 3.958-Inch Tubing Diameter Showing Effect of Water-Cut (Well 1)

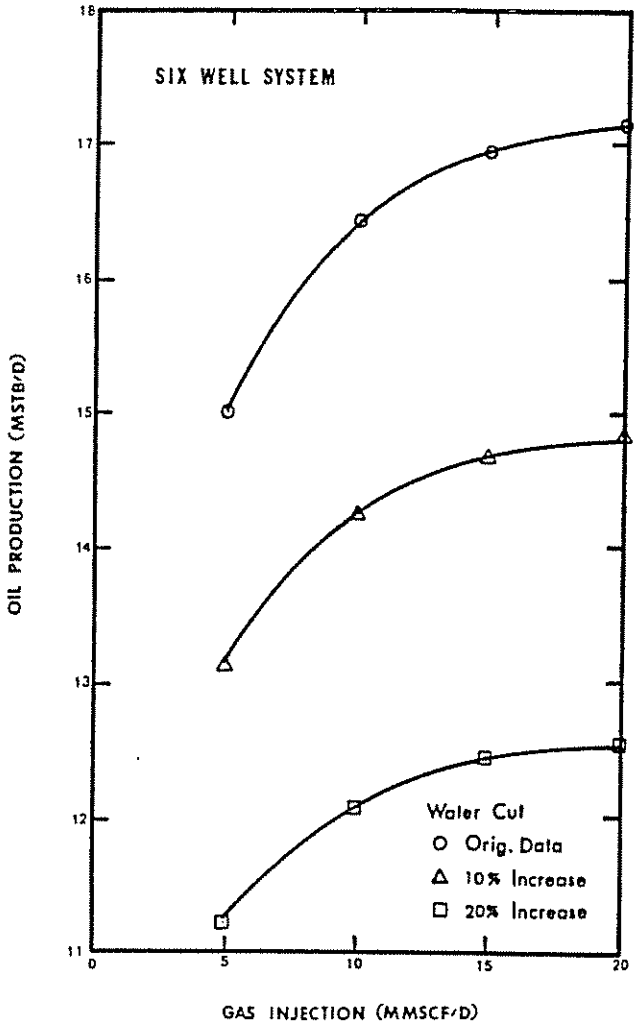


Fig. 3.452 Total Gas Allocation and Oil Production Curve Showing Effect of Water-Cut

production rate. For example, Table 3.448 shows the effect of injection pressure for Well #1. As the injection pressure increases from 830 psi to 1230 psi, the point of injection lowers from 4363.76 feet to 6290.8 feet, and the oil production increases from 4296.120 STB/D to 4668.86 STB/D.

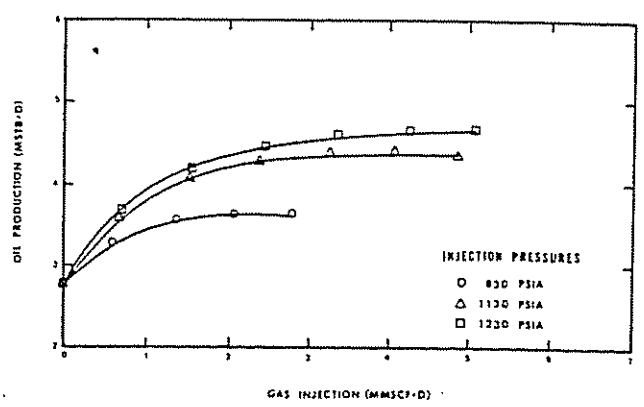


Fig. 3.453 Gas Lift Performance for Different Injection Pressures (Well 1)

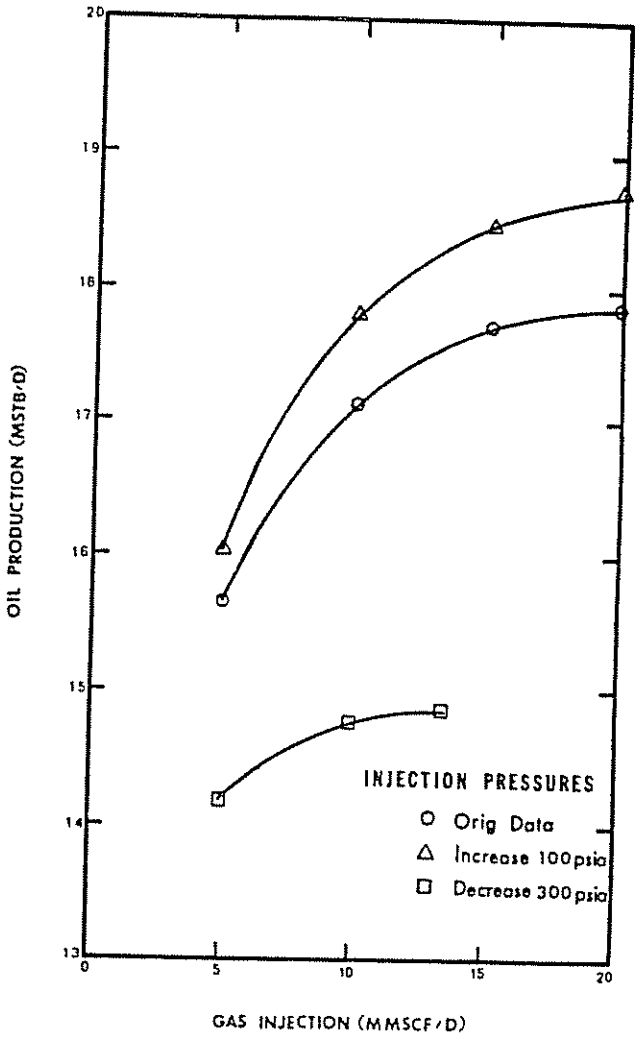


Fig. 3.454 Total Gas Allocation and Oil Production Curve Showing Effect of Injection Pressures

If the availability of the gas supply is limited, the minimum flowing gradient can be attained with a higher gas injection pressure. For example, in Figure 3.454 a production rate of 1700 STB/D requires 7 MMcf/D of injection gas volume compared to 9 MMcf/D of injection gas for a 100 psi difference in injection pressure.

TABLE 3.448
WELL NO. 1
CONTINUOUS FLOW GAS LIFT PERFORMANCE WITH DIFFERENT CASING PRESSURE

Casing pressure 830 psia		Casing pressure 1230 psia		Casing pressure 1130 psia	
Oil production (STB/D)	Point injection (ft)	Oil production (STB/D)	Point injection (ft)	Oil production (STB/D)	Point injection (ft)
4,296.120	4,363.760	4,668.860	6,290.800	4,590.380	5,835.070
4,550.270	4,526.780	5,189.000	6,632.410	5,063.030	6,127.800
4,628.860	4,583.480	5,473.633	6,813.080	5,292.900	6,287.830
4,612.550	4,574.360	5,609.130	6,908.150	5,396.500	6,359.110
4,612.550	4,574.360	5,609.130	6,908.150	5,396.500	6,359.110
		5,659.310	6,942.540	5,419.680	6,375.980
		5,659.310	6,933.410	5,393.360	6,358.980

As expected, higher available injection pressures will reduce the gas injection volume and increase the oil production rate.

3.426436 Effect of productivity index

Figure 3.455 shows gas lift performance curves for Well #1 for three values of PI. Figure 3.456 shows a weaker well. Figure 3.457 shows how the productivity index affects the oil production rate of the six-well system. For example: with 10 MMcf/D of injection gas into a system of six wells, the oil production rate drops from 17,736.660 STB/D to 15,692.70 STB/D, as the productivity index drops from a high PI to a low PI. The inflow performance is a variable over which we have no control. However, if it is possible to accurately predict the future productivity, it is possible to predict the effect of other variables on the oil production rate and gas injection rate as the productivity changes. This permits an optimum design of the variables for the life of the well.

3.426437 Summary

The effect of variables in the optimization of gas lift systems has been shown for a six-well system. Any changes in the variables will definitely change the oil production rate and gas injection rate. In general, predicting the future changes in water cut, inflow performance, and static pressure makes it possible to obtain the optimum well parameters such as tubing size, flowline size, separator pressure and compressor capacity. Then it is possible to obtain the optimum oil production rate through the study of the effect of the variables by using multiphase flow correlations and optimization of the gas available.

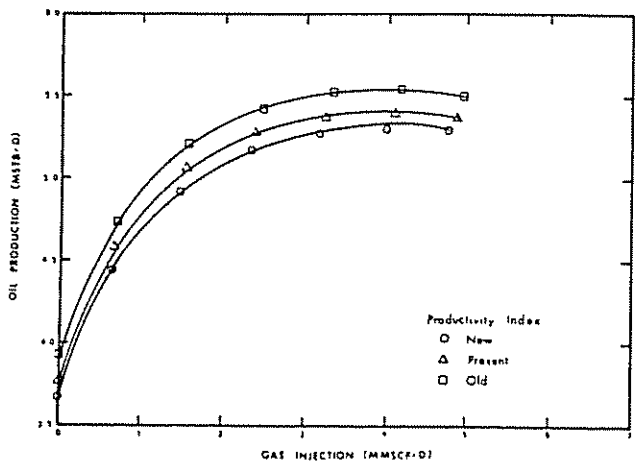


Fig. 3.455 Gas-Lift Performance Curve with 3.958-Inch Tubing Diameter Showing Effect of Productivity Index

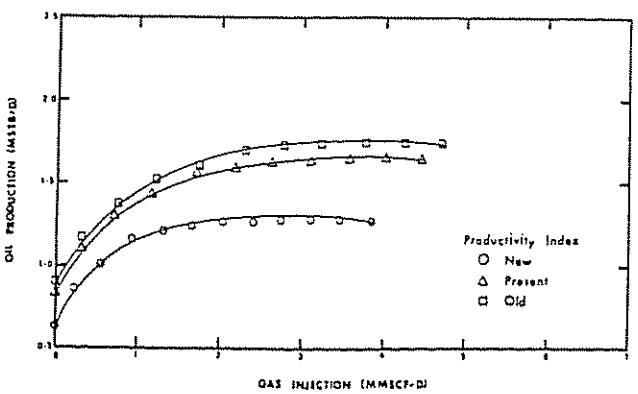


Fig. 3.456 Gas-Lift Performance Curve With 3.958-Inch Tubing Diameter

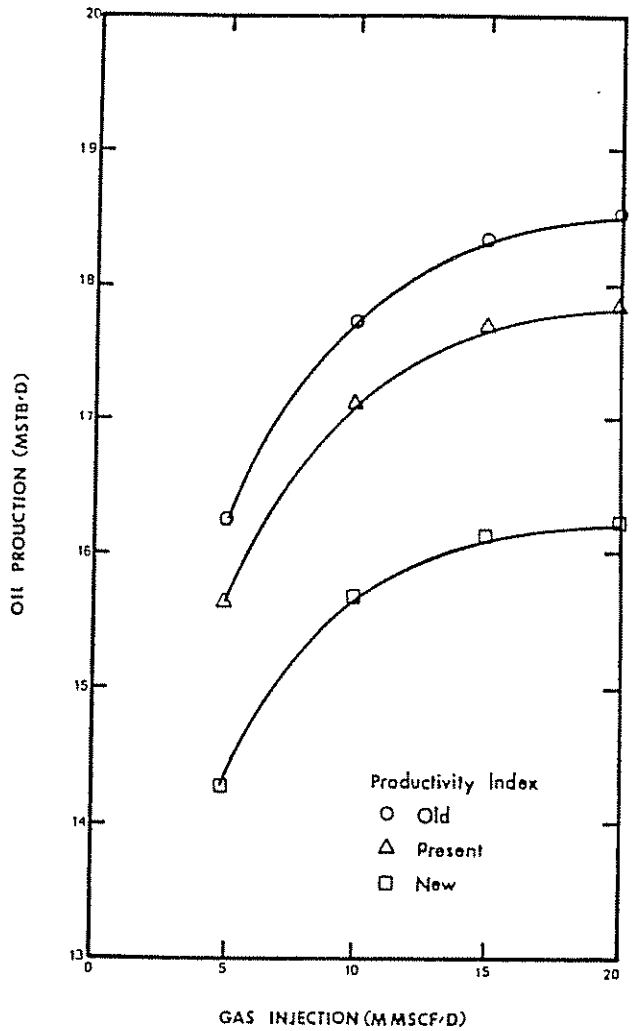


Fig. 3.457 Total Gas Allocation and Oil Production Curve Showing Effect of Productivity Index

3.4265 Conclusions and recommendations

Conclusions:

Based on this study, the following conclusions are made:

(1) When accurate well data are available, the optimum design of the gas lift system can be achieved by observing the effect of the variables from the multiphase flow correlations.

(2) The optimum allocation of gas in a continuous flow gas lift system can be determined analytically.

(3) The continuous flow gas lift performance curve can detect and diagnose inefficient gas lift wells as well as design the system.

(4) Predicting the injection gas volume for the later

life of gas lift wells can be obtained by using the future inflow performance of the well and the optimization program.

(5) A computerized gas lift optimization program can be a valuable tool for design and daily operation of gas lift systems. The program can determine the effect of changing wellhead pressure, injection operating pressure, water cut, pipe size, and changing well characteristic.

Recommendations:

(1) For the solution of this problem, that is, to obtain convergence for a correct GLR, calculations should be made from top to bottom instead of from bottom to top. This is due to the relationship of the flow rate in the multiphase flow correlation and the flow rate as determined by an equation of inflow performance.

(2) For optimization, the cubic spline interpolating scheme can be used instead of the curve fitting technique to save computer time and to avoid unnecessary steps.

(3) For a larger system, divide the system into several small subsystems, and then combine the subsystems.

(4) An economic study should be made to analyze the profitability and the limitation of the variables.

3.4266 Field case of economic optimization

3.42661 Introduction

Redden et al. presented an economic optimization procedure that was used in a Venezuelan oil field and applied to some 1500 gas lift wells.[18] Optimum gas distribution was extremely important in this field because of the need to use some of the available gas for gas pressure maintenance projects. Also, they found a need to reallocate the gas on a daily or weekly basis depending on whether or not a compressor station was down for repairs or for regular maintenance.

3.42662 Optimization logic

Redden et al. gave the following logic for the optimization process:[18]

Simplified Outline of Gas-Lift Optimization Logic

(A) Retrieve well test data, well descriptions, and field producing system description from data cards and computer data storage files.

(B) For each continuous-flow gas-lift well, calculate its productivity index.

(1) Use last test production data and well completion description.

(2) Calculate the flowing bottom-hole pressure using a two-phase vertical flowing pressure correlation.

(3) Productivity index.

$$J = \frac{q}{\overline{P}_R - p_{wf}} \qquad (1)$$

(C) For each continuous-flow gas-lift well, calculate the economic optimum producing rate without system limitations.

(1) Increase the production rate of the well by some small amount (e.g., 5%) above the last well test.

(2) Calculate the producing bottom-hole pressure at the new rate using the wells productivity index.

$$p_{wf} = \overline{P}_R - \frac{q}{J} \qquad (2)$$

(3) Calculate the gas injection rate required at the new rate using a two-phase vertical flow correlation.

(4) Convert the new production rate to a dollar revenue value and the gas injection rate to a dollar expense value.

(5) Compare the calculated revenue and expense values to the similar revenue and expense values for the well at the last test rate.

(a) Let change in revenue be DY.

(b) Let change in expense be DX.

(c) The slope of the line (rate of change) between the two points is DY/DX.

(6) Compare the calculated slope of the performance curve (DY/DX) to the desired slope at the economic optimum rate without system limitations.

(a) Required slope at optimum point is 1 + ROR/100, where ROR is the required rate of return (%) on the incremental expense.

(b) If DY/DX is greater than the required slope, increase the production rate by another small increment and repeat the calculation for the slope between the new points.

(c) If DY/DX is equal to or less than the required slope, calculate a production rate midway between the two production rates used to determine the slope. This is the optimum production rate.

(d) Calculate the gas injection requirements at the optimum production rate. This is the optimum gas injection rate.

(D) For all the continuous-flow gas-lift wells supplied by a compressor plant (source of high pressure gas), determine if the gas available is sufficient to produce the wells at their unlimited optimum rate.

(1) Sum the optimum gas injection requirements for all continuous-flow gas-lift wells supplied by a compressor plant.

(2) Calculate the gas injection rate available for continuous-flow gas lift.

(a) From the total compressor plant capacity, subtract the amount for fuel and system losses, sales, gas injection for reservoir pressure maintenance, and other demands. This leaves an amount available for gas lift.

(b) From the amount available for gas lift, subtract the volume required for intermittent lift and continuous flow wells which must be deleted from the optimization calculation due to missing data or other reasons. This leaves an amount available for continuous-flow gas-lift optimization.

(3) If the compression capacity available for continuous-flow gas-lift optimization is greater than

the sum of the individual well requirements, go to the next compressor plant in the system and repeat the procedure.

(4) If any compressor plant has insufficient gas available for the continuous-flow gas-lift wells to operate at their unlimited optimum rate, calculate a reduced gas injection requirement in the following manner.

(a) For each gas-lift well supplied by the compressor plant with insufficient capacity, subtract some small amount (e.g., 10 B/D) from the unlimited optimum fluid production rate and calculate the gas injection requirement.

(b) Convert the changes in production rate and gas injection to a change in dollars revenue (DY) and a change in dollars expense (DX).

(c) Calculate the rate of change (DY/DX) between the two points for each well.

(d) Rank the well according to their DY/DX value (slope) with the well having the smallest numerical slope at the top of the list.

(e) For the well at the top of the list, subtract the change in gas injection rate between the unlimited optimum rate and the lower calculated rate from the total required gas injection rate.

(f) Subtract another small amount from the new production rate for the top well in the list and calculate a new slope between the two points.

(g) Insert the well into the list of wells depending on its slope.

(h) Repeat the process until the gas requirement is equal to or less than the compressor plant gas available.

(E) Check to see if there are any production capacity limitations on the gas lift wells at the separator station.

(1) Determine the continuous-flow gas-lift well fluid production rates (oil, water, total liquid, and gas) to each separator station.

(2) Calculate the available capacity of each separator station for the continuous-flow gas-lift fluid production.

(a) Summate the fluid protection rates for all wells other than continuous-flow gas-lift wells which are being optimized.

(b) Subtract the production rates (oil, water, total liquid, and gas, respectively) for the non-gas-lift wells from the capacity of each separator station. This gives the capacity of the station for continuous-flow gas-lift wells.

(3) Compare the available separator station capacity for continuous-flow gas-lift fluid production to the rates being produced to the station.

(4) If the station capacity is less than the production rate, continue the procedure described above for reducing the production rate until the capacity limits are met.

(F) Check to see if there are any production limitations on the gas-lift wells due to compressor plant gas inlet capacity.

(1) For each compressor plant, calculate the amount of gas being produced to it from continuous-flow gas-lift wells (gas-lift gas plus formation gas).

(2) Calculate the available capacity of each compressor plant for the gas from continuous-flow gas-lift wells.

(a) Summate the gas produced to the plant from wells other than continuous-flow gas-lift wells (intermittent gas-lift well gas, formation gas from flowing and pumping wells, etc.)

(b) Subtract the sum calculated above from the sum of the compressor plant capacity plus the allowable flare limit. The remainder is the capacity available for gas-lift wells.

(3) Compare the capacity available for continuous-flow gas-lift wells to the amount of gas being produced to the compressor plant.

(4) If the capacity is less than the rate being produced to the plant, reduce the production rate of the continuous-flow gas-lift wells according to the procedure described.

3.42663 Example problem

Redden et al. presented an example problem for a 10-well field.[18] Figure 3.458 shows a model of this system which contains one compressor plant, one separator station, and 10 gas lift wells. The compressor plant had a total capacity of 5600 Mcf/D; however, 3600 Mcf/D were required for other purposes, leaving only 2000 Mcf/D for gas-lifting the ten wells. Table 3.449 shows the necessary data for a typical well. Table 3.450 shows the pertinent data for each well.

First, they calculated the maximum economic production rate assuming no system capacity limitations. (See Table 3.451.) By increasing the compressor capacity available for gas lift from 2000 Mcf/D to 2496 Mcf/D (496 Mcf/D increase), a production increase of 189 BO/D

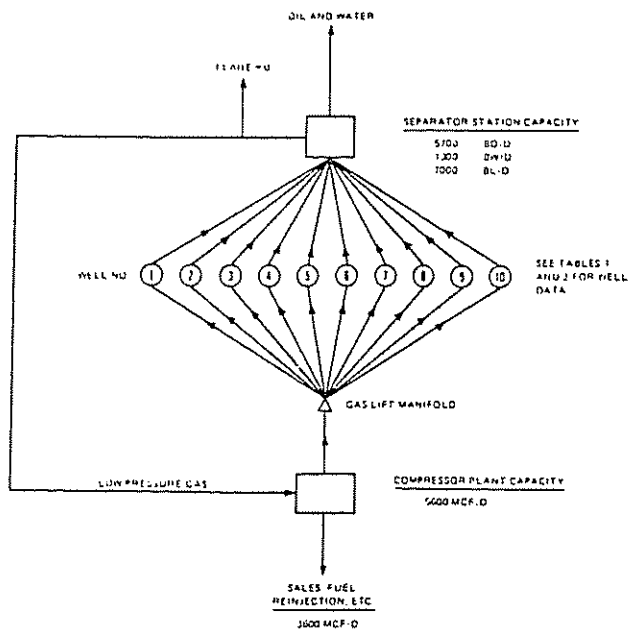


Fig. 3.458 Example Gas Lift System Model (after Redden, et al)

TABLE 3.449
GENERAL WELL DATA

Oil density	35° API
Tubing inside diameter	2.992 in.
Depth of producing interval	5500 ft (subsurface)
Depth of operating gas-lift valve	5000 ft (subsurface)
Static bottom-hole pressure	1400 psig
Producing wellhead pressure	100 psig
Gas specific gravity (air = 1.0)	0.8
Value of oil	$1.00/bbl
Cost of gas-lift gas	$0.25/Mcf

TABLE 3.450
EXAMPLE WELL TEST DATA

Well no.	Total liquid B/D	Oil B/D	Water B/D	Formation gas-oil ratio cu ft/bbl	Gas-lift gas Mcf/D	Calculated productivity index B/D/psi
1	1000	1000	0	500	200	1.12
2	1000	750	250	500	200	1.28
3	1000	500	500	500	200	1.64
4	700	700	0	500	200	0.72
5	600	600	0	500	200	0.61
6	500	500	0	500	200	0.51
7	500	300	200	500	200	0.66
8	500	250	250	500	200	0.73
9	500	500	0	550	200	0.50
10	500	500	0	600	200	0.49
	6800	5600	1200		2000	

TABLE 3.451
OPTIMIZATION WITHOUT SYSTEM LIMITATIONS

	Calculated Optimum Rate			Change from Last Well Test		
Well no.	Total liquid B/D	Oil B/D	Gas-lift gas Mcf/D	Total liquid B/D	Oil B/D	Gas-lift gas Mcf/D
1	1013	1013	258	13	13	58
2	1049	787	312	49	37	112
3	1098	549	316	98	49	116
4	695	695	176	−5	−5	−24
5	608	608	216	8	8	16
6	530	530	262	30	30	62
7	527	316	248	27	16	48
8	532	266	245	32	16	45
9	519	519	247	19	19	47
10	506	506	216	6	6	16
	7077	5789	2496	277	189	496

was noted. Even in this case, Well #4 was reduced in both oil production rate and gas injection rate. Their next step was to determine the most economically desirable disribution of the gas for the system limitation of 2000 Mcf/D. (Refer to Table 3.452). An increase of 13 BO/D is obtained over the last well tests. Gas was decreased to some wells and increased to others.

They also showed how the gas would need to be redistributed for a drop in available lift gas to 1000 Mcf/D. This is shown in Table 3.453.

They illustrated another problem whereby the oil handling capacity at a station was reduced from 5700

TABLE 3.452
OPTIMIZATION WITH COMPRESSOR CAPACITY LIMITATIONS
(2000 Mcf/D AVAILABLE)

	Calculated Optimum Rate			Change from Last Well Test		
Well no.	Total liquid B/D	Oil B/D	Gas-lift gas Mcf/D	Total liquid B/D	Oil B/D	Gas-lift gas Mcf/D
1	983	983	160	−17	−17	−40
2	978	734	160	−22	−16	−40
3	1048	524	240	48	24	40
4	695	695	176	−5	−5	−24
5	608	608	216	8	8	16
6	520	520	237	20	20	37
7	497	298	197	−3	−2	−3
8	512	256	214	12	6	14
9	509	509	222	9	9	22
10	486	486	168	−14	−14	−32
	6836	5613	1990	36	13	−10

TABLE 3.453
OPTIMIZATION WITH COMPRESSOR CAPACITY LIMITATIONS
(1000 Mcf/D AVAILABLE)

	Calculated Optimum Rate			Change from Last Well Test		
Well no.	Total liquid B/D	Oil B/D	Gas-lift gas Mcf/D	Total liquid B/D	Oil B/D	Gas-lift gas Mcf/D
1	913	913	7	−87	−87	−193
2	959	719	127	−41	−31	−73
3	968	484	181	−32	−16	−19
4	655	655	112	−45	−45	−88
5	498	498	61	−102	−102	−139
6	430	430	81	−70	−70	−119
7	447	268	148	−53	−32	−52
8	442	221	157	−58	−29	−43
9	419	419	62	−81	−81	−138
10	426	426	58	−74	−74	−142
	6157	5033	994	0643	−567	−1006

TABLE 3.454
OPTIMIZATION WITH SEPARATOR CAPACITY LIMITATIONS
(4000 BO/D MAXIMUM)

	Calculated Optimum Rate			Change from Last Well Test		
Well no.	Total liquid B/D	Oil B/D	Gas-lift gas Mcf/D	Total liquid B/D	Oil B/D	Gas-lift gas Mcf/D
1	903	903	0	−97	−97	−200
2	679	509	21	−321	−241	−179
3	648	324	87	−352	−176	−113
4	515	515	11	−185	−185	−189
5	418	418	30	−182	−182	−170
6	340	340	44	−160	−160	−156
7	307	184	103	−193	−116	−97
8	252	126	111	−248	−124	−89
9	329	329	30	−171	−171	−170
10	346	346	19	−154	−154	−181
	4737	3994	456	−2063	−1606	−1544

to 4000 BO/D. Table 3.454 shows how the gas can be re-distributed to obtain 4000 BO/D maximum.

3.42664 Summary

Proper gas allocation to each well is important. To accomplish this properly and on short notice, automatic control devices for gas input are needed. If manual adjustments by the man in the field are required, the problem becomes difficult and engineering supervision is probably necessary. Proper gas lift valves (variable orifice type) for continuous flow must be used. Sometimes a change in the injection gas rate will cause certain types of valves to begin an intermitting action (opening and closing). This can cause more headaches than benefits. In other cases, additional valves may open, causing excess gas use.

However, with properly designed installations the valves should respond properly to injected gas, and optimization procedures can be beneficial.

Valve port sizes and chokes should be designed for optimum conditions with the normal operating volume of gas. Retrievable equipment will allow changes in order to reach this condition. Most of these valves will then permit changes in gas injection volumes without serious consequences. The gas lift valve for each well must permit this flexibility or the re-allocation of gas to each well will fail. For example, some valves will intermit if sufficient gas is not maintained to keep the valve open. Therefore, if gas allocated to this well is less than sufficient to keep the valve open, this valve must be retrieved and replaced with the properly sized valve.

3.427 Selection of gas lift parameters

3.4271 Introduction

Before final design of the gas lift system, the gas lift designer should have plots of the gas lift well showing the relation of the parameters of flow rate, PI, total gas-liquid ratio, surface operating pressure, wellhead pressure, etc. This enables the designer to simulate changing conditions expected in the future and to select gas pressures, gas volumes, wellhead pressures, etc. He then can modify the final design to accommodate the most optimum present conditions and expected changes.

For example, there may exist the possibility that one of the compressors may shut down in which case the surface operating pressure may drop by 100 psi. The compressors may also shut down completely, and gas may be available from a nearby gas well at a high pressure and relatively high cost. The designer may want to incorporate in the design the capability to switch to and operate at these emergency conditions as efficiently as possible. This may involve the installation of extra valves on the tubing string or extra mandrels for quick wireline installation when necessary and may mean a re-allocation of the gas to different wells.

3.4272 Mitchell's procedure

An attempt was made to provide a simple method for preparing plots to evaluate various parameters and a general cross-plotting method was devised by Mitchell.[10]

This procedure is for a constant wellhead pressure and does not consider the most economical TGLR, but allows one to change any design parameter and note the effect on the others. A procedure for construction of a general cross plot of the main gas lift parameters is described. The method of gas lift design using the pressure vs. TGLR curve is used so economic considerations described previously by Mitchell can be considered.

Procedure:

(1) Prepare the multiphase flow plots for various assumed flow rates:
   (a) Assume various flow rates
   (b) Plot the vertical multiphase flow data with pressure on the ordinate, TGLR on the abscissa for various depths

   Determine the maximum TGLR curve by connecting the points of zero slope for each depth plotted. Do this for each flow rate assumed. These plots are shown in Figs. 3.459A, B, and C.

(2) (a) Prepare pressure vs. depth plots on transparent graph paper so the pressure scale is similar to the pressure scale of the pressure vs. TGLR plot in Step 1.
   (b) Assume various values of surface operating

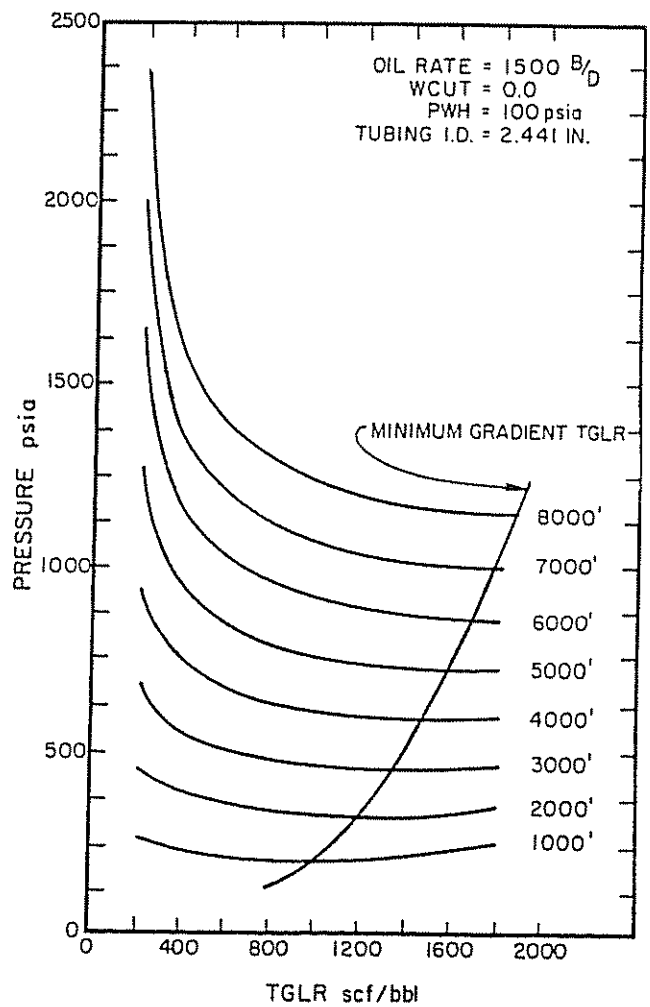


Fig. 3.459-A Pressure vs. TGLR Curve for A Flow Rate of 1500 BBL/D

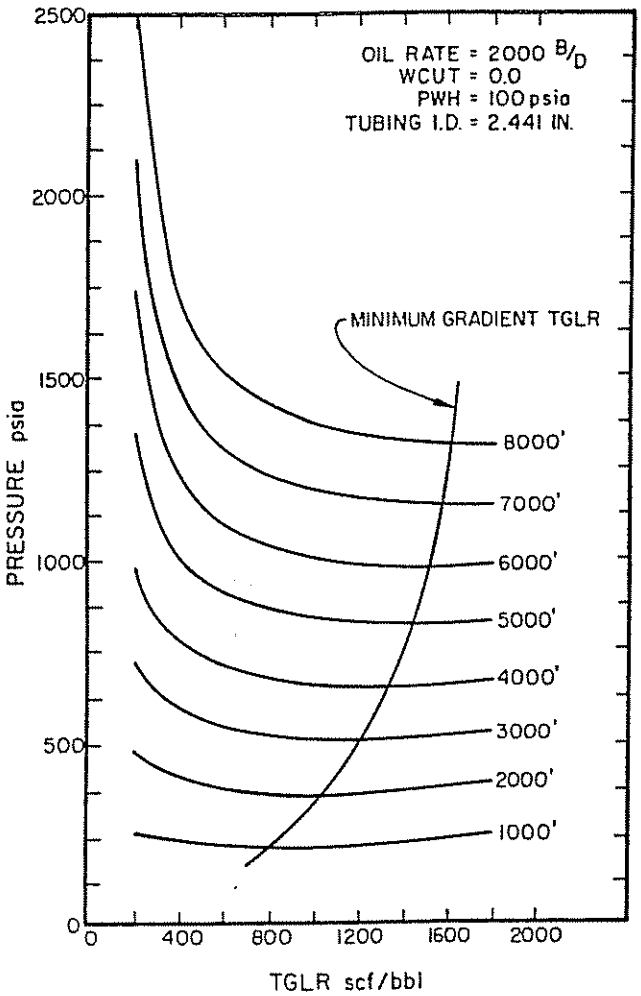


Fig. 3.459-B Pressure vs. TGLR Curve for A Flow Rate of 2000 BBL/D

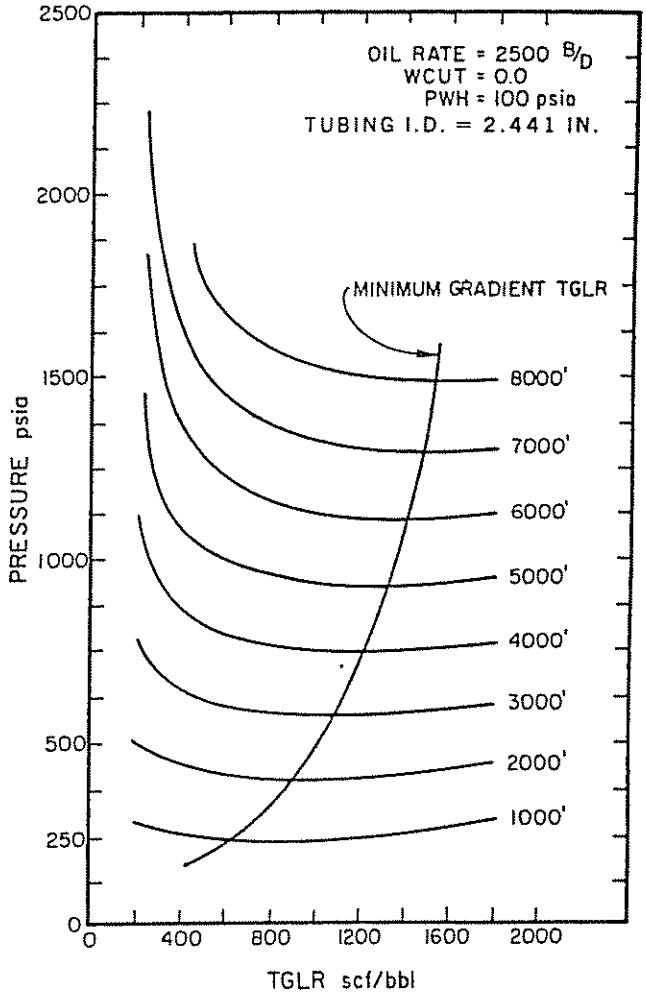


Fig. 3.459-C Pressure vs. TGLR Curve for A Flow Rate of 2500 BBL/D

pressures (P_{so}) and plot these values at zero depth. Determine the gas gradients and plot the gas pressures at well depth. Draw dashed-straight lines through these points for each value of P_{so}. These lines represent the gas injection pressure. Subtract 100 psi from each value of P_{so} and plot these values at zero depth. Draw solid lines parallel to the dashed lines drawn above—one line for each value of P_{so}. These solid lines are called the gas operating lines and represent the gas pressure available to the tubing at the injection points. The 100 psi represents the pressure drop occurring across the gas lift valve. These plots are shown in Figs. 3.460A, B and C.

(3) (a) Overlay Fig. 3.459 for a particular flow rate with Fig. 3.460 for a particular surface operating pressure so that the pressure scale matches, i.e. 500 psia on Fig. 3.459 corresponds with 500 psia on Fig. 3.460.

(b) Choose a value of TGLR

(c) Slide Fig. 3.460 horizontally over Fig. 3.459 until the depth read on the abscissa of Fig. 3.460 is equal to the depth interpolated on Fig. 3.458 for the value of TGLR assumed. Mark

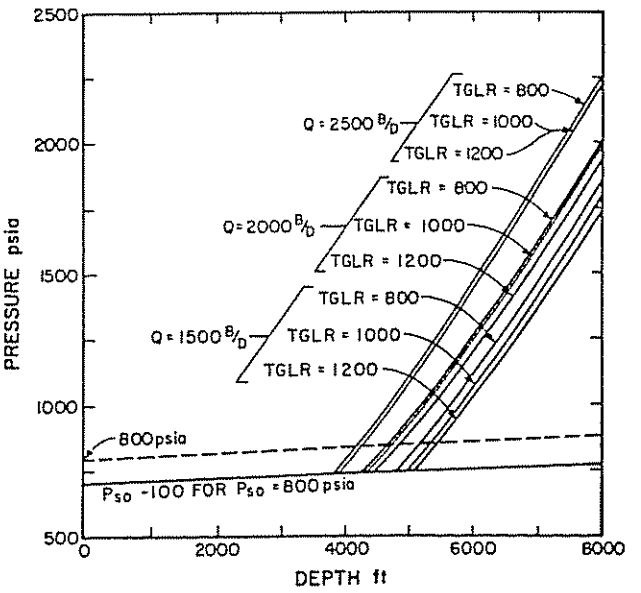


Fig. 3.460-A Pressure vs. Depth Curve for A Surface Operating Pressure of 800 PSIA

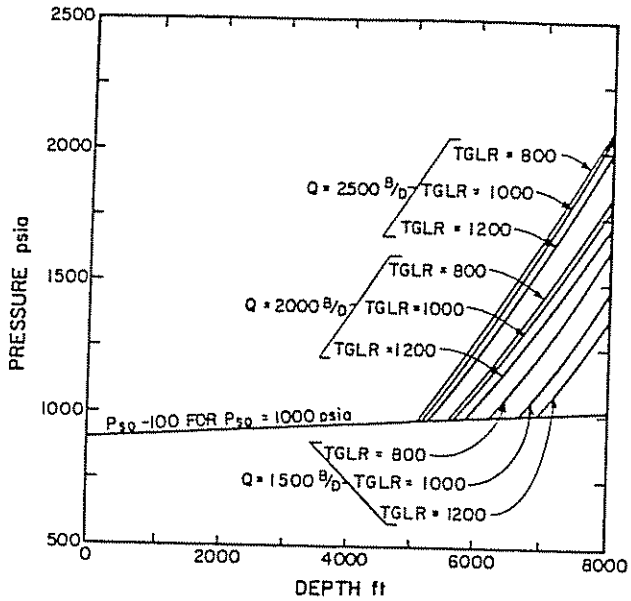


Fig. 3.460-B Pressure vs. Depth Curve for A Surface Operating Pressure of 1000 PSIA

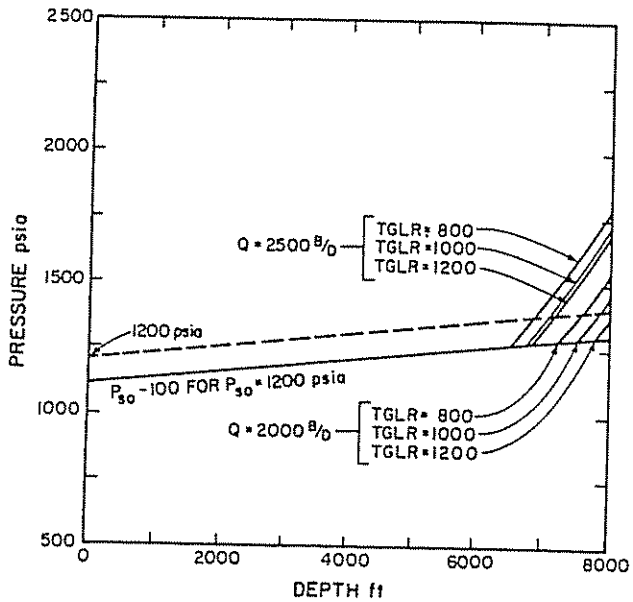


Fig. 3.460-C Pressure vs. TGLR Curve for A Surface Operating Pressure of 1200 PSIA

this point on Fig. 3.460 as the injection point.

(d) Slide Fig. 3.460 horizontally over Fig. 3.459 until the point of injection determined in Step (c) lines up on the vertical line of Fig. 3.459 corresponding to the formation GLR. Starting from the injection point estimate 1000 ft. below, read off the pressure on Fig. 3.459. Plot this point on Fig. 3.460 as a depth of injection point plus 1000 ft. Continue estimating depths and plotting the pressures on Fig. 3.460 until the well depth is reached. This traces out the traverse below the point of injection. Read the pressure at the well depth. This is the flowing bottom hole pressure required for the well to flow at the assumed rate and TGLR. As an alternative the pressure traverse at the formation GLR can be traced from existing pressure vs. depth gradient curves or can be calculated by computer.

(4) (a) Prepare Fig. 3.461 as a plot of pressure vs. flow rate. For ease of plotting, make the plot on transparent paper with the pressure scale similar to Figs. 3.459 and 3.460.

(b) Plot the inflow performance of the well, i.e. plot bottom hole pressure vs. flow rate. Also plot the points obtained in Step 3(a).

(c) Repeat from Step 3(a) to Step 4(b) for all the values of TGLR to be considered.

(d) Repeat from Step 3(a) to Step 4(c) for all values of surface operating pressures assumed. Tabulate data as noted in Table 3.455.

Repeat Steps 1 through 4(d) for all flow rates. On Fig. 3.461 connect points obtained using the same value of surface operating pressure and TGLR for different flow rates. If separate plots for each value of surface operating pressure are used, connect points of equal TGLR to form a curve. This is the tubing multiphase flow performance curve. The point where this curve intersects the inflow performance curve is the producing rate of the well.

(5) (a) Prepare Fig. 3.462 which is a plot of surface operating pressure vs. flow rate similar to Fig. 3.461.

(b) Plot the flow rate obtained in Step 5 for each value of TGLR and P_{so}.

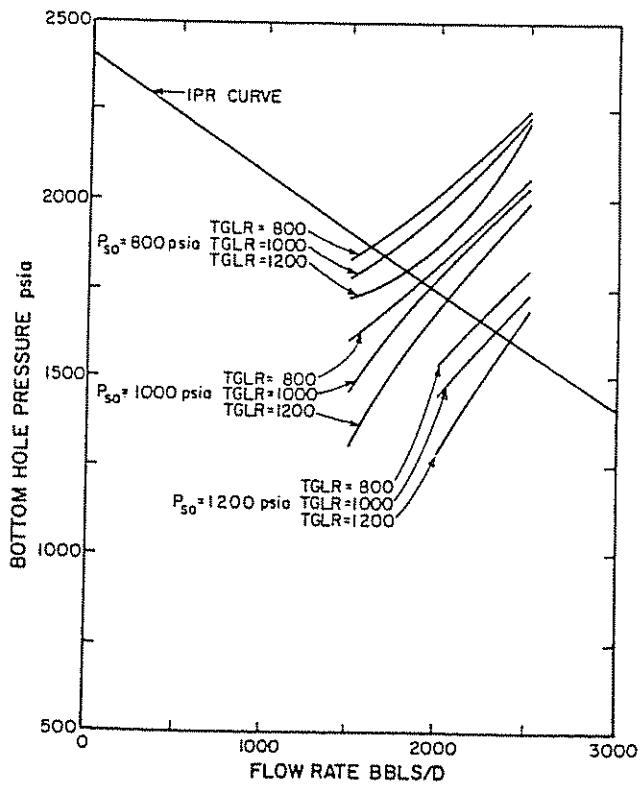


Fig. 3.461 Determining Flow Rates from Gas Lift Well (after Mitchell)

TABLE 3.455
RESULTS FROM GENERAL CROSS PLOT

TGLR	P_{so}	Rate	Depth	P_{inj}	P_{wf}
800	800	2500	3800	725	2250
1000	800	2500	3900	727	2225
1200	800	2500	3900	727	2225
800	800	2000	4250	730	2012
1000	800	2000	4300	730	1975
1200	800	2000	4400	735	2125
800	800	1500	4800	737	1837
1000	800	1500	5000	740	1785
1200	800	1500	5100	742	1725
800	1000	2500	5100	970	2062
1000	1000	2500	5150	973	2040
1200	1000	2500	5250	975	1990
800	1000	2000	5550	978	1812
1000	1000	2000	5650	980	1775
1200	1000	2000	5850	985	1700
800	1000	1500	6200	987	1600
1000	1000	1500	6600	990	1463
1200	1000	1500	6900	995	1312
800	1200	2500	6600	1268	1800
1000	1200	2500	6800	1275	1740
1200	1200	2500	6850	1277	1700
800	1200	2000	7250	1285	1525
1000	1200	2000	7550	1300	1440
1200	1200	2000	7800	1310	1440

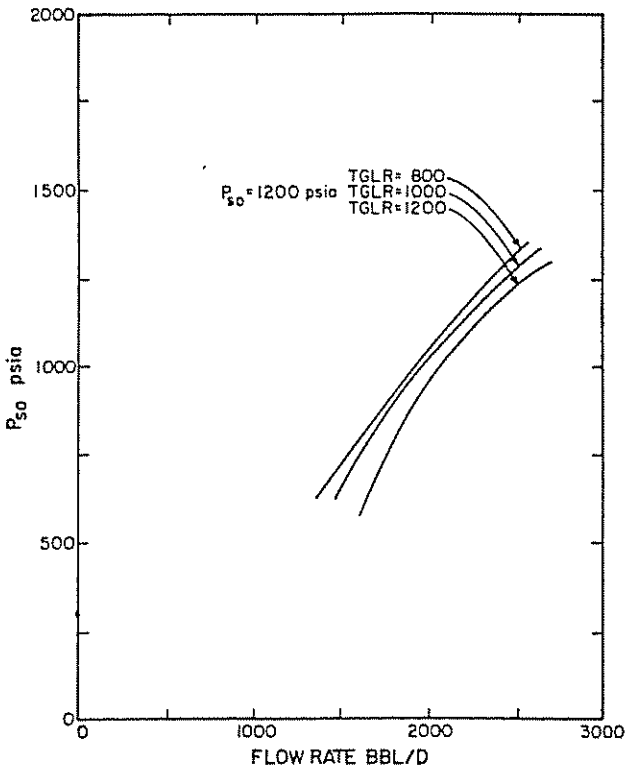


Fig. 3.462 Surface Operating Pressure vs. Flow Rate (after Mitchell)

(c) Connect the points of the corresponding values of P_{so} for varying TGLR's.

(6) (a) Prepare Fig. 3.463 (a plot of flow rate vs. depth) with depth of injection on the ordinate and flow rate on the abscissa.

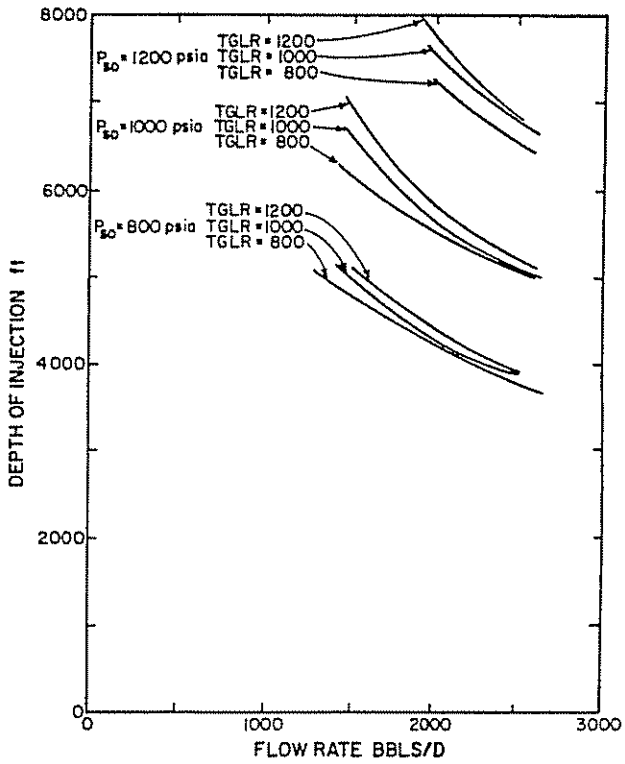


Fig. 3.463 Depth of Injection vs. Flow Rate (after Mitchell)

(b) Plot the values of flow rate vs. depth of injection for the various values of surface operating pressure and TGLR.

(c) Connect the points of corresponding values of surface operating pressure and TGLR by a smooth curve. This method is further illustrated by the following example problem:

EXAMPLE PROBLEM

Tubing size = 2½"
Depth = 8000'
API gravity = 30°
$\bar{P}_R$ = 2400 psia
P_{wh} = 100 psia
PI = 3 bbl/D/psi (constant)
γ_g = 0.65
R_s = 200 scf/bbl
P_{so} = 800, 900, 1000 psia
TGLR = 800, 1000, 1200 scf/bbl

(1) (a) Assume flow rates of 1500, 2000 and 2500 B/D. The well is estimated to produce somewhere within this range. If the well does not flow within the range of flow rates chosen, alternative flow rates must be assumed. This is determined in Step 4.

(b) Prepare Fig. 3.459 (vertical multiphase flow pressure vs. TGLR plots for each flow rate). Plot pressure on the ordinate (1" to 500 psi) with the TGLR on the abscissa (1" to 400 scf/bbl). The pressure range extends from 0 psi to 2500 psi. Mark off the GLR scale which extends from 0 - 2000 scf/bbl in increments of 200 scf/bbl.

(2) Prepare Fig. 3.460 (a pressure vs. depth scale on transparent paper) so the pressure scale is identical to that in Step 1. The depth scale is made 1" to 1000 ft. Mark off the depths of 0 - 8000 ft in increments of 1000 ft. Determine the gas gradients to be 19, 23.6, and 26.1 psi/1000 ft for P_{so}'s of 800, 1000 and 1200 psia, respectively. Calculate the gas pressure at 8000 ft for each value of P_{so} to be 952, 1189, and 1409 psi, respectively. Mark off the gas pressures at 8000 ft. A dashed line may be drawn through these points. Subtract 100 psia from each P_{so} value and draw the gas operating lines parallel to the corresponding dashed lines beginning at each value of (P_{so} - 100 psia) at zero depth.

P_{so}	Pressure at 0 ft	Pressure at 8000 ft
800	800	952
1000	1000	1189
1200	1200	1409

(3) (a) Overlay Fig. 3.459 with Fig. 3.460 so the pressure scale matches. Choose one value of TGLR, 800 scf/bbl. Choose a value of P_{so}, 800 psia. Slide Fig. 3.460 horizontally across Fig. 3.459 keeping the pressure scale the same until the depth interpolated on Fig. 3.459 is equal to the depth read along the gas operating line. Mark off this point on Fig. 3.460A as the point of injection (4800 ft) and note the corresponding pressure to be 757 psia. With the use of a proportional divider, an accuracy up to 10 ft or so can be achieved.

(b) Slide Fig. 3.460 horizontally over Fig. 3.459 until the point of injection determined in Step 3(a) lines up to the vertical line in Fig. 3.459 corresponding to the formation GLR of 200 scf/bbl. From the injection point, move up 1000 ft on Fig. 3.459, i.e. at 5800 ft, and read the pressure of 1062 psia. This corresponds to the pressure 1000 ft below the point of injection. Mark off this point on Fig. 3.460, i.e. 1162 psia and 5800 ft. Move up another 1000 ft on Fig. 3.459, i.e. to 6800 ft, and read the pressure (1375 psi). Mark this point on Fig. 3.460. Repeat this procedure until 8000 ft or well depth is reached. Draw a smooth curve through these points. This curve represents the pressure traverse below the point of injection. Read from this curve the pressure at 8000 ft to be 1837 psia. Alternatively, as described previously, the pressure traverse below the point of injection can be traced from existing gradient curves or calculated.

(c) Repeat Steps 3 (a) and (b) for values of TGLR 1000 and 1200 scf/bbl, and for P_{so} values of 1000 and 1200 psia.

(d) Repeat Step 3 (a), (b) and (c) for the other values of flow rates assumed, i.e. 2000, 2500 bbl/D.

(e) Record all values of TGLR, P_{so}, flow rate, injection pressure, injection depth, and flowing bottom hole pressure. See Table 3.455.

(4) (a) Prepare Fig. 3.461 which is a pressure vs. flow rate plot. The pressure scale may be the same as before on the ordinate, the flow scale may be drawn 1" per 1000 B/D to 3000 B/D. Draw in the linear inflow performance curve by plotting the point 2400 psia at zero B/D flow rate and the point 2400 psia – (3000/3.0) = 1400 psia at 3000 B/D. Join the points with a straight line. Plot values of flowing bottom hole pressure and flow rate for the various values of P_{so} and TGLR assumed in Step 3, recorded in Table 3.455.

Connect points of different flow rates so there is a curve for each combination P_{so} and TGLR. These curves represent the multiphase flow performance. Record the intersection of the above curves and the inflow performance curve as noted in Table 3.456. These values of flow rate at the intersection points represent the flow rate possible from the well for the particular combination of TGLR and P_{so}. If the multiphase flow performance curve does not intersect the inflow performance curve, then the well will not produce at the rates assumed. New rates should be chosen and the procedure repeated from Step 1.

TABLE 3.456
RESULTS FROM GENERAL CROSS PLOT

TGLR (scf/bbl)	P_{so} (psia)	Flow rate (B/D)
800	800	1600
1000	800	1675
1200	800	1800
800	1000	1906
1000	1000	1955
1200	1000	2045
800	1200	2230
1000	1200	2313
1200	1200	2380

Obtain a good estimate of the new flow rates by extrapolation of the multiphase flow performance curves to intersect the inflow performance curves. The points of intersection give the values of the new assumed rates.

(5) (a) Prepare Fig. 3.462 similarly to Fig. 3.461. Plot values of flow rate for each combination of P_{so} and TGLR determined in Step 4 and recorded in Table 3.456. Connect points of equal TGLR.

(6) Prepare Fig. 3.463 which is a plot of depth on the ordinate vs. flow rate on the abscissa. For this problem the depth scale is 1" = 1000 ft and the pressure scale is 1" = 500 bbl/D. Using data from Table 3.456, plot depth of injection vs. flow rate for the different values of surface operating pressure and TGLR. From this plot, injection depth may be determined for any value of surface operating pressure and TGLR.

Analysis of design:

The designer now has the following three plots with which to analyze the gas lift system:

Fig. 3.461—P_{wf} vs. flow rate for different values of P_{so} and TGLR
Fig. 3.462—P_{so} vs. flow rate
Fig. 3.463—Depth of injection vs. flow rate
These plots can be used as follows:

Figure 3.461:
This plot is useful only when there is a change in PI or bottom hole pressure. This usually occurs with time or if some well stimulation operation is performed. The new PI relation is then plotted on Figure 3.461 and new Figures 3.462 and 3.463 are plotted.

Figures 3.462 and 3.463:
These are used together to study the gas lift well performance. For example, if the surface operating pressure drops from 1000 psia to 900 psia with the well producing 2000 B/D, Figure 3.463 indicates that the depth of injection will change from 5700 feet to 5000 feet; therefore, a valve should be located at around 5000 feet. On the other hand, Figure 3.463 indicates that 2000 B/D can be maintained by increasing the TGLR from 1100 scf/bbl to 1400 scf/bbl. This is observed by interpolating for the TGLR at the point where the surface operating pressure equals 900 psi and the flow rate is 2000 B/D.

The economical limit must also be considered and this is easily accomplished by using the minimum gradient TGLR obtained from the pressure vs. TGLR curves. Mitchell explains this procedure in Section 3.425.

For example, a well producing 2000 B/D from an injection depth of 5700 feet, a TGLR of 1100 scf/bbl, and a surface operating pressure of 1000 psi has a minimum gradient TGLR of 1500 scf/bbl. Table 3.431 indicates that the economical TGLR at a slope of 0.01 occurs at 1450 scf/bbl. If our surface operating pressure drops to 900 psia, the TGLR must be increased to 1400 scf/bbl to operate at a depth of 5700 feet and to maintain the present flow rate. The economical limit TGLR is 1450 scf/bbl; hence, economical operation is still being maintained.

For land operations with long flow lines, the effect of the flow line and changing wellhead pressure must be included. Accomplish this by preparing the pressure vs. TGLR curves for a constant separator pressure, hence including the flow line in the multiphase flow calculations. This means different multiphase performance curves (pressure vs. TGLR) must be prepared for each well.

3.4273 Special design to make selection of parameters in continuous-flow gas lift

3.42731 Introduction

To apply gas lift to a reservoir or group of wells, an average condition of one well is sometimes considered in making the design. The information thus obtained determines the specification of surface facilities, e.g. compressors, high pressure separator capacity, sizing of gas distribution lines, gas injection pressures and volumes.

To decide whether or not gas lift is feasible, determine the natural flow conditions. The original IPR and the variation in the static bottom hole pressure, along with future inflow performance curves, give an idea of the well performance.

After determining that a satisfactory production rate cannot be maintained under natural flow conditions, determine gas quantities, injection pressure, and injection depth required for favorable gas lift conditions. Select these parameters from the continuous flow gas lift equilibrium curve system, which consists of four different diagrams:

(1) pressure-rate diagram
(2) pressure-depth diagram
(3) gas-liquid ratio—rate diagram
(4) injection pressure—rate diagram

The principal diagram is the pressure rate diagram which is constructed using the equilibrium curve concept for a number of total (natural + injection) gas-liquid ratios, considering the multiphase flow performance and the inflow performance of the wells. Afterwards, the additional parameters can be studied by constructing the other three diagrams.

3.42732 Design procedure

A general procedure to construct the continuous flow gas lift equilibrium curve system was presented by Nijveld,[19] and further elaborated on by Marin.[6]

(1) Assume various flow rates
(2) Assume various depths (injection points)
(3) Assume various total gas-liquid ratios near the available or required amount of gas.
(4) Select the proper vertical flowing gradient curve for each flow rate assumed or prepare for computer calculations.
(5) For each flow rate assumed, determine the required flowing bottom-hole pressure from the inflow ability of the well, that is,

$$P_{wf} = \bar{P}_R - (q_L/PI) \qquad \text{(for constant PI)}$$

where:

P_{wf} = flowing bottom-hole pressure,
$\bar{P}_R$ = static bottom-hole pressure, psi,
q_L = production flow rate, bpd, and
PI = productivity index, bpd per psi drawdown (assume linear).

(6) Determine the flowing bottom-hole pressure for each of the assumed depths (injection points).
  (a) Using a sheet of rectangular coordinate paper, prepare a graph of depth vs. pressure. The graph should have the same scale as the flowing gradient curves if they are to be used.
  (b) Mark the flowing bottom-hole pressures obtained in Step 5 at the total depth on graph part (a).
  (c) Overlay and align the depth-pressure graph on the chosen flowing gradient curve (flow rate). From the flowing bottom pressure marked, slide upwards or downwards along the baseline (depth in feet) until reaching the formation gas-liquid ratio. From this point, trace the corresponding flowing gradient in an upward manner until it reaches the assumed depths.
  (d) Note the flowing bottom-hole pressures for the various depths.
  At this point, the procedure is the same for each flow rate.

(7) The results of Step 6 are presented in tabulated form as IPR (P) + VFP to depth.
(8) For each flow rate assumed, the flowing bottom-hole pressure at each assumed depth for the different total gas-liquid ratio assumed is determined as follows:
  (a) Prepare a plot of depth vs. pressure on rectangular coordinate paper. The graph should have the same scale as the flowing gradient curves if they are to be used.
  (b) Mark the value of P_{wh} at the surface on the depth vs. pressure plot.
  (c) Overlay and align the depth-pressure plot on the flowing gradient curve chosen and slide downward along the baseline (depth in feet) until the flowing wellhead pressure intercepts the total gas-liquid ratio assumed. From this point, trace the flowing gradient curve downward until it reaches the total depth.
  (d) From the curve as arrived at in Step (c), the flowing bottom hole pressure is determined for the various depths assumed.
  (e) Repeat Steps (c) and (d) for each total gas-liquid ratio assumed.
(9) The results of the vertical performance in Step 8 are presented in tabulated form as "VFP (TGL), 0' - to desired depth."
(10) Prepare a flowing bottom hole pressure-rate diagram by plotting the data obtained in steps 7 and 9. The intersection of the inflow performance set of curves with the vertical performance curves for a determined total gas-liquid ratio at the same depths gives the equilibrium points.
(11) Draw the line through the intersection points of the two sets of curves. This resulting curve is the equilibrium curve for the total gas-liquid ratio assumed.
(12) The same procedure of Steps 10 and 11 are followed for the other total gas-liquid ratios, resulting in a family of equilibrium curves at different total gas-liquid ratios.
(13) Construct a pressure-depth diagram using the equilibrium curve for each total gas-liquid ratio (Step 12) in a crossplotting manner, defining the equilibrium curves.
(14) Construct a gas-liquid ratio-flow rate diagram using the equilibrium curve for each total gas-liquid ratio (Step 12) in a crossplotting manner, defining the equilibrium curves for different depths.
(15) Construct an injection pressure-rate diagram assuming various injection pressures at the surface.
  (a) For each injection pressure, determine the gas injection line in the pressure-depth diagram.
  (b) The intersection of the injection line, with the different equilibrium curves (different total gas-liquid ratios), determines the injection points and the flowing bottom hole pressures.
  (c) For each pressure determined in Step 15 (b), the flow rate is determined for each total gas-liquid ratio in the pressure-flow rate diagram in a crossplotting manner.
  (d) Repeat Steps 15 (b) and 15 (c) for each injection pressure assumed.
  (e) Plot data obtained in Step 15 (d) to construct the injection-pressure-flow rate diagram.

3.42733 Selecting parameters

To select the various parameters in a continuous gas lift design, the gas lift equilibrium curve system is constructed. This system constitutes the basis for selecting the four gas lift parameters necessary for any flow configuration.

They include the following:
(1) Production rate
(2) Total gas-liquid ratio
(3) Injection depth
(4) Injection pressure

This design procedure can be used in many ways; but if a desired flow rate is assumed, the following procedure is used in determining other parameters. Rates possible for different injection pressures can also be determined.

(1) In the pressure rate diagram, with the selected flow rate, the minimum total gas-liquid ratio is chosen as follows: from the selected flow rate, move to the right on the pressure scale until the minimum equilibrium curve is reached. This equilibrium curve represents the minimum total gas-liquid ratio.
(2) From the point in Step 1, proceed upward on the pressure-depth diagram until intercepting the equilibrium curve of the corresponding total gas-liquid ratio. The injection depth is thus determined.
(3) At the point of injection depth, determine the flowing bottom-hole pressure existing inside the flow string. This pressure establishes the minimum injection pressure required at the surface. (Disregard the friction losses when gas is injected in the annulus.)
(4) Determine the minimum injection pressure as follows:

$$P_{so} \text{ at surface} = P_{wf} - \text{gas pressure gradient} \times \text{injection depth}$$

Knowing the selected parameters, determine the injection gas volume requirements for the well using the following equation:

$$\text{injection gas volume} = q \times (TGLR - R_s)$$

where:

q = production flow rate, B/D
$TGLR$ = total gas-liquid ratio, Scf/B
R_s = solution gas-liquid ratio, Scf/B

(5) The injection depth is also determined by moving to the right from the point in Step 1 on the gas-liquid ratio-rate diagram until reaching the corresponding total gas-liquid ratio determined in Step 1.

3.42734 Example problem

To select the parameters for a continuous-flow gas lift system, a correlation in the form of flowing pressure gradient curves is used to work the following example problem.

The fluid physical properties of the well are assumed to be the same as in the correlation to predict the flowing pressure gradients. The Hagedorn and Brown vertical correlation is used.

The conditions of the well are:

Well depth = 9,650 ft
Tubing size = 3 in.
$\gamma_g = 0.65$
$\overline{P}_R = 3{,}850$ psi
$P_{wh} = 150$ psi (constant)
PI = 7 bpd/psi (assume constant)
$R_s = 50$ scf/bbl
Production = 100% oil
Desired rate = 4,000 B/D

Required:

(a) Select the parameters (injection pressure, injection depth, and the total gas-liquid ratio) for a gas lift continuous design for an assumed flow rate of 4,000 B/D.
(b) For given injection pressures of 800, 1,000, and 1,200 psi, determine the flow rates.

Solution procedure. For this solution, it is necessary to construct the continuous gas lift equilibrium curve system graphs as noted in Fig. 3.464. These are constructed individually and are shown in more detail on Figs. 3.464A, 3.464B, 3.464C and 3.464D.

The previously described procedure is used:

(1) Production rates assumed are 1,000, 2,000, 3,000, 4,000, and 5,000 B/D.
(2) Depths (injection points) assumed include 500, 1,000, 2,000, 3,500, 5,000, 6,500, and 8,000 ft.
(3) Total gas-liquid ratios assumed are 100, 200, 300, 400, 600, and 800 scf/bbl.
(4) The proper vertical flowing gradient curves are selected or a computer program may be used.
(5) For each production rate, determine the flowing pressure from the inflow ability of the well.

$$P_{wf} = (\overline{P}_R - q_L/PI). \text{ (assume Constant PI)}$$

For 1,000 B/D,

$$P_{wf} = (3{,}850 - 1{,}000/7) = 3{,}707 \text{ psi.}$$

The other values are found in the same manner.

Tabulate each P_{wf} required from the well's inflow ability, as noted in Table 3.457.

TABLE 3.457
RATES VS. FLOWING PRESSURES

Assumed rate	P_{wf}
1,000	3,707
2,000	3,560
3,000	3,421
4,000	3,280

(6) For each production rate, the flowing bottom hole pressure is determined at the assumed depths, using a formation gas-liquid ratio of 50 Scf/B, as Step 6 in the general procedure. The results are presented in Table 3.457.
(7) For each production rate, the flowing bottom-hole pressure at each assumed depth for the different total gas-liquid ratios is determined as described in Step 8 from the given procedure. The results are presented in Table 3.458.

TABLE 3.458
FLOWING BOTTOM-HOLE PRESSURES VS. RATES — AT VARIOUS DEPTHS AND DIFFERENT GAS-LIQUID RATIOS

Assumed rate	9,650'	8,000'	6,500'	5,000'	3,500'	2,000'
For TGLR = 100 Scf/B						
1,000	3,560	2,920	2,365	1,810	1,230	700
2,000	3,580	2,960	2,410	1,850	1,290	730
3,000	3,600	2,990	2,420	1,850	1,290	730
For TGLR = 200 Scf/B						
1,000	3,110	2,490	1,935	1,380	840	400
2,000	3,160	2,540	1,980	1,420	890	465
3,000	3,420	2,780	2,200	1,620	1,070	610
For TGLR = 300 Scf/B						
1,000	2,620	2,010	1,450	980	620	370
2,000	2,770	2,140	1,580	1,080	720	400
4,000	3,170	2,520	1,940	1,360	950	530
For TGLR = 400 Scf/B						
1,000	2,060	1,570	1,140	795	525	300
2,000	2,370	1,740	1,280	890	630	380
4,000	2,960	2,320	1,740	1,260	880	550
For TGLR = 600 Scf/B						
2,000	1,880	1,390	1,060	840	560	370
4,000	2,560	1,980	1,530	1,180	840	550
5,000	2,900	2,270	1,770	1,360	970	600
For TGLR = 800 Scf/B						
2,000	1,600	1,260	1,000	770	560	290
4,000	2,350	1,870	1,490	1,140	840	350
5,000	2,700	2,160	1,720	1,330	970	380

(8) A pressure-flow rate diagram is prepared as noted in Fig. 3.464A with the values shown in tables 3.457 and 3.458 and showing the intersection of the set of inflow performance curves with the set of curves for vertical performance, determining the equilibrium point at each depth. In Figure 3.464A only the intersection points correspond to the total gas-liquid ratio of 200 Scf/B is shown.
(9) The connection of the equilibrium points of Step 8 defines the equilibrium curve for a total gas-liquid ratio of 200 Scf/B.
(10) Repeat the procedures in Steps 8 and 9 for the other total gas-liquid ratios assumed. Then, a family of equilibrium curves is defined (see Fig. 3.464A).
(11) Construct the pressure-depth diagram (see Fig. 3.464B) as Step 13 in the general procedure.
(12) Construct the gas-liquid ratio-flow rate diagram (see Fig. 3.464C) as Step 14 in the general procedure.

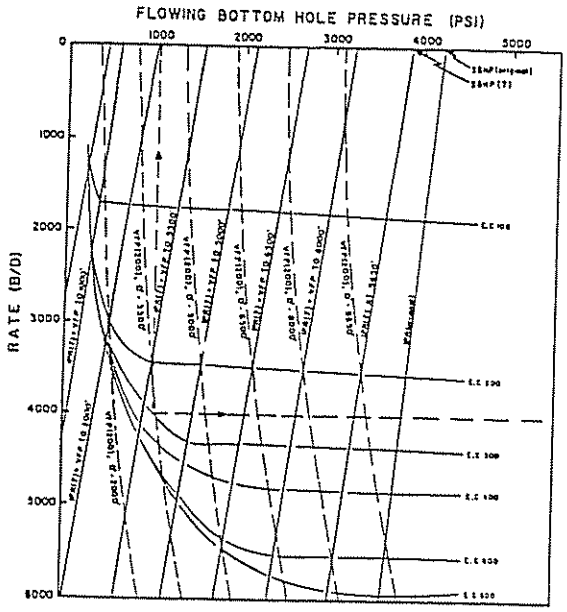


Fig. 3.464-A Flowing Bottom Hole Pressure-Rate Diagram

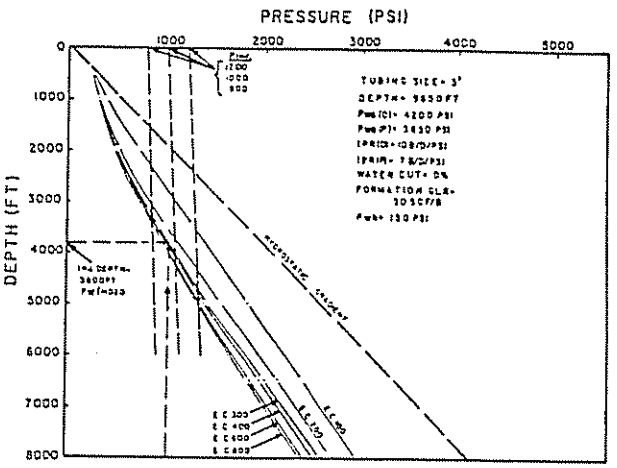


Fig. 3.464-B Pressure-Depth Diagram

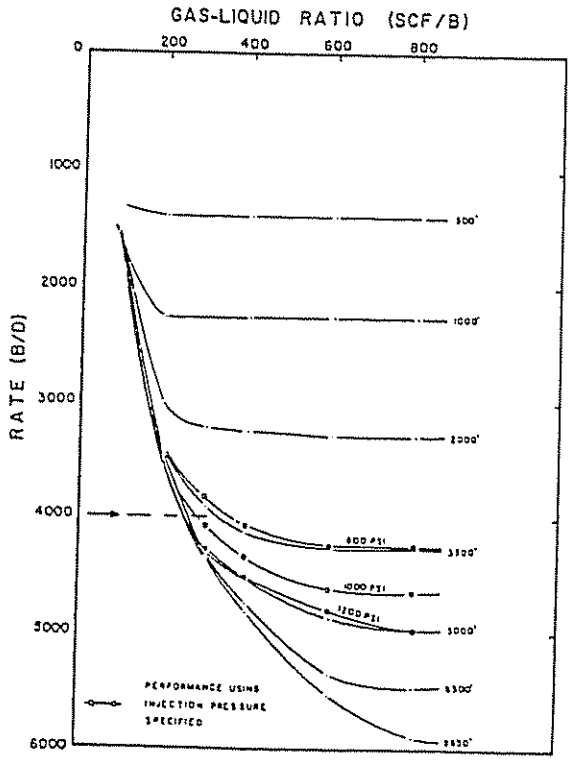


Fig. 3.464-C Gas-Liquid Ratio-Rate Diagram

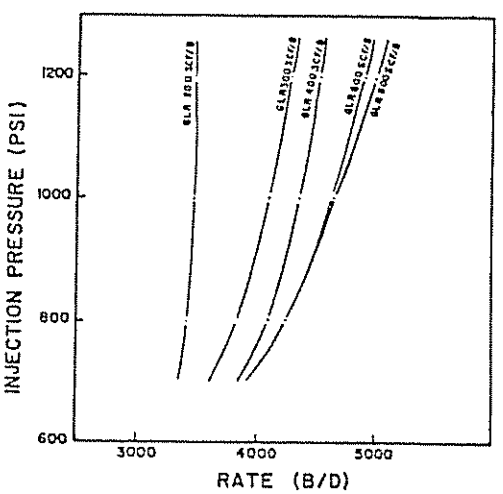


Fig. 3.464-D Injection Pressure-Rate Diagram

Fig. 3.464 Continuous Gas Lift Equilibrium Curve System

(13) For a given injection pressure construct an injection pressure gas-liquid ratio as (see Fig. 3.464D) Step 15 in the general procedure.

TABLE 3.459
FLOWING BOTTOM-HOLE PRESSURE - IPR (7) + VFP

Assumed rate	Flowing Bottom-Hole Pressures					
	9,650'	8,000'	6,500'	5,000'	3,500'	2,000'
1,000	3,707	2,800	2,340	1,700	1,100	460
2,000	3,560	2,880	2,220	1,620	990	370
3,000	3,421	2,720	2,060	1,420	810	240
4,000	3,280	2,575	1,930	1,280	630	100

Having constructed the four diagrams for the continuous gas lift equilibrium curve system (Fig. 3.464), the following selection of parameters was made. The results are as follows:

Total gas-liquid ratio = 300 Scf/B
Injection depth = 3,800 ft
Flowing pressure = 1,020 psi

The surface injection pressure can be selected based on the flowing tubing pressure of 1020 psi at depth and a proper differential between tubing and casing pressure of 100 psi.

$$\text{Injection gas volume} = q \times (TGLR - R_s)$$
$$= 4000 \text{ bpd } (300 - 50) \text{ Scf/B}$$
$$= 1 \text{ MMScf/D}$$

The determination of the production flow rate for the various assumed injection pressures at a total gas-liquid ratio of 300 Scf/B is obtained from the injection pressure-flow diagram (see Fig. 3.464D). From the depth-pressure diagram (Fig. 3.464B), the respective injection points are determined.

TABLE 3.460
INJECTION DEPTHS AND GAS VOLUMES

Injection pressure assumed	Flow rate	Injection depth	Injection gas volume M Scf/D
800	3,820	3,250	955
1,000	4,080	4,020	1,020
1,200	4,280	4,750	1,070

This system gives another procedure for evaluating the various gas lift parameters and can be used in the design of new systems.

3.428 A new gas lift concept—"two-step gas lift installation," by Juan Faustinelli

3.4281 Introduction

This new gas lift concept by Faustinelli proposes a method to increase liquid production, preferably in those wells which are producing at fairly low rates.[20] Wells with very low productivity indexes and low static bottom hole pressures are not candidates for the proposed new gas lift concept. The following discussion is essentially the same as given by Faustinelli.[20]

For a given standard gas lift installation, a minimum P_{wf} can be obtained with the correct gas lift design. For set conditions it may not be possible to reduce the flowing BHP further and obtain a greater drawdown. This work proposes a method to produce the liquid to a certain height by one tubing string, and to use another tubing string to produce the fluid to the surface. The purpose is to create a lower flowing bottom-hole pressure than is possible by one string. Flow in the annular space can serve as one of the strings.

A gas lift well with the correct tubing size and gas lift valves will produce a certain maximum rate with a constant surface operating pressure and a constant gas volume. There is no way to increase this rate since the production of the well depends on the FBHP and it must balance with the hydraulic column pressure of the flowing two-phase fluids.

However, if the fluids are produced in one tubing string to a certain height and then on to the surface in another string, the flowing BHP can be reduced and a larger liquid rate obtained. Two tubing strings are required to lift the liquid to the surface. One disadvantage of this method is its complicated completion equipment; another is that only one production zone can be lifted. This method can be better utilized for a single completion with casing sizes of 7 or 9-⅝ inches.

3.4282 Description of the two-step gas lift method

Figure 3.465 shows the principle of the two-step continuous gas lift method. The formation fluid—liquid and gas—is lifted by gas lift injected through tubing #1 to a certain height where the pressure traverse is a little higher than the surface flowing pressure (P_{wh}).

At this level, a window communicates tubing #1 with the casing annular space. The free gas is produced

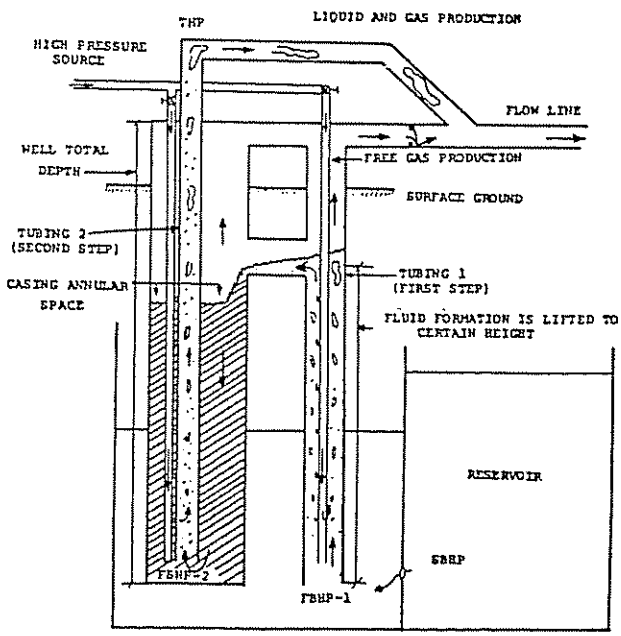


Fig. 3.465 Two-Step Gas Lift Principle

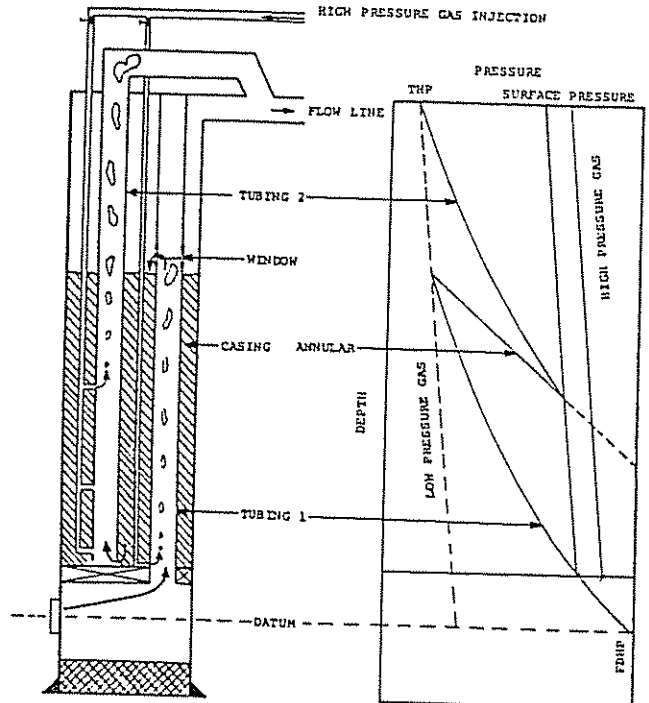


Fig. 3.466 Theoretical Two-Step Gas Lift Completion with The Pressure Gradient Curves

at a low pressure upward and the liquid flows downward in the casing annular space. The gas-free liquid has a high pressure gradient and builds a higher flowing pressure (P_{wf_2}) than the original flowing pressure(P_{wf_1}). With this higher P_{wf_2}, the liquid is better able to produce to the surface through tubing #2.

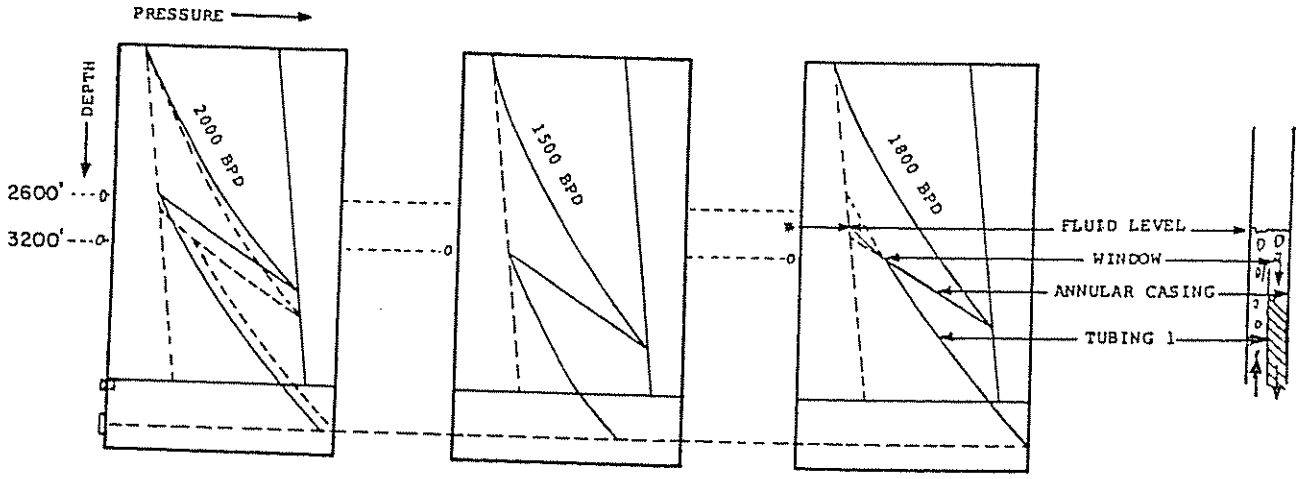
A theoretical wellbore completion with the two-step gas lift installation and its vertical flowing pressure gradient curves is shown in Figure 3.466.

A graphical two-step gas lift design is shown in Figure 3.467. Observe that the depth of the window—2600 feet—calculated with the present reservoir data (Figure 3.467a) would be too high if the shutin BHP and PI decrease during the life of the well. In Figure 3.467b another design is made for future reservoir data and the window is proposed at 3200 feet. Figure 3.467c represents the initial flowing pressure curves—using present reservoir data—for the window proposed at 3200 feet instead of 2600 feet. The depth of this window is not the ideal depth for the present condition and a loss of production is obtained. If the variation of the shutin BHP and PI during the life of the well is small, the production loss also will be very small and the well will be able to produce efficiently over a long period of time.

In cases where the window is set lower than the ideal design for initial reservoir data, the fluids erupt throughout the window with a considerable pressure but decrease rapidly in the casing annular space due to its large surface cross section. The casing above the window, due to its large cross-sectional area and its vertical position, works as an excellent gas-liquid vertical separator.

Figure 3.468 compares the standard and two-step gas lift installation for different reservoir data and well characteristics. In this figure, observe that the flowing BHP for the two-step procedure is lower than the flowing BHP for a standard design. This represents a larger drawdown and therefore a larger production rate.

Theoretically, the two-step gas lift installation is always better than the conventional gas lift installation because the fluids are lifted from a lower depth; but due to its complicated subsurface equipment, only those wells with large production casings and low or



(a) Ideal Design for Actual Reservoir Data. The window (0) appears at 2600 ft. Slotted curve is the same curve of Fig. 7c.

(b) Ideal Design for a Future Reservoir Datum. Lower SBHP and PI. The window is lowered at 3200 ft.

(c) Initial Pressure Traverse Curves if the Window is set at 3200 ft instead of 2600 ft. As we can see the fluid level (*) in the casing space is well

Fig. 3.467 Graphical Two-Step Gas Lift Design for A Given Well Datum

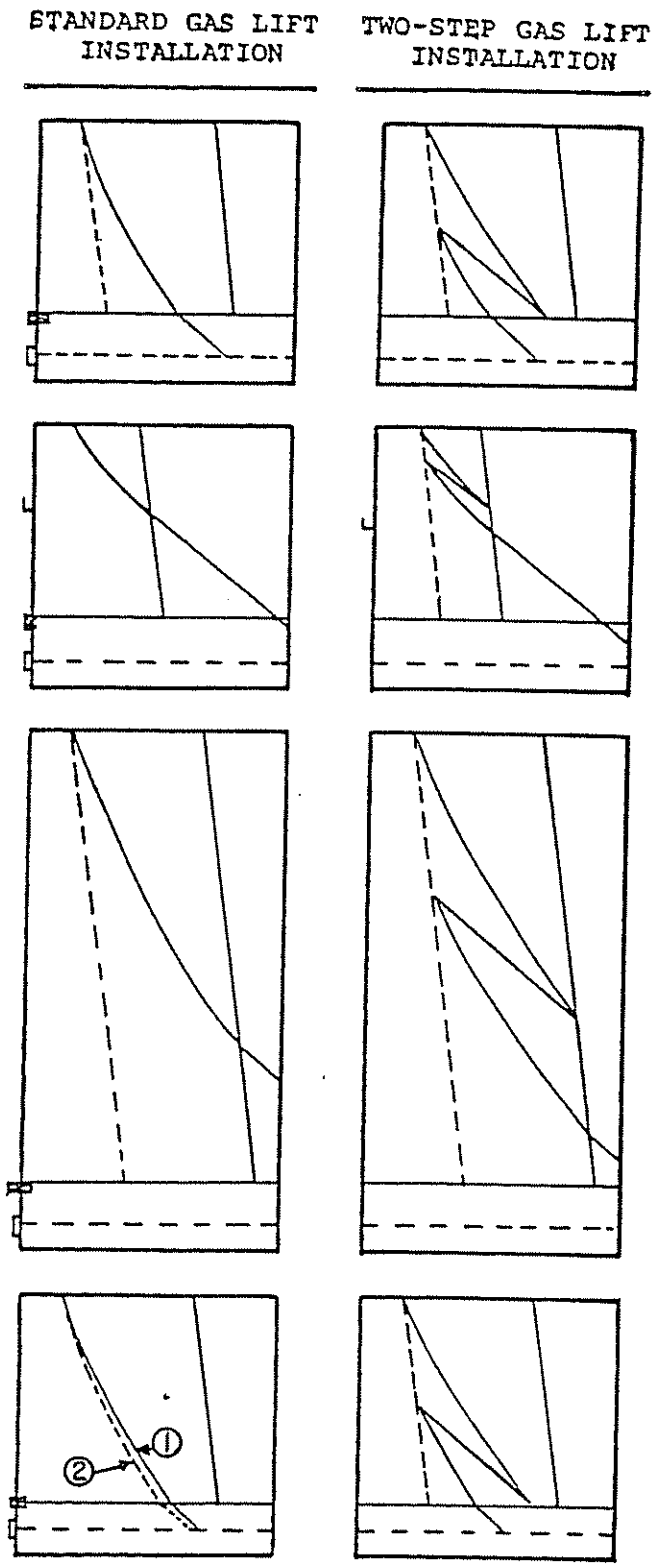


Fig. 3.468 Comparison of Pressure Traverse Curves Between Standard And Two-Step Gas Lift Installation for Different Reservoir And Well Characteristics

moderate rates must be considered for this new gas lift installation.

3.4283 Proposed well bore completion for a two-step gas lift installation

Faustinelli proposed the following completion procedures for the two-step gas lift installation.

3.42831 Parallel two-step completion for wells of Lake Maracaibo, Venezuela

A large number of wells in Lake Maracaibo are dually completed with two strings of 3½-in. tubing and casing of 9⅝ inches. Due to normal reservoir depletion, a high percentage of these wells were successively recompleted in higher productive zones until many of them reached the economical production limit. The oil production is from the Eocene formation which has an approximate thickness of 2000 feet, and the top of this formation is limited by an unconformity at ±7000 feet in the center of the lake. The lower reservoirs of the Eocene formation are thick with a high water-drive mechanism and can be easily produced to a 98% water cut with standard gas lift completions. The upper reservoir consists of thin layers with a partial water drive and segregation drive mechanism. Many wells completed in the uppermost layers or those completed in the Graben zone are limited up dip by a large fault and secondary faults. They produce with a relatively clean production and a GOR of 1000 Scf/B. When one of these wells dies due to a lower shutin BHP but a normal GLR, and if the well is not producing too much water, a standard gas lift installation will not permit a large drawdown on the formation; therefore, the well will produce at a low rate.

Those wells dually completed and with 9⅝-in. casing, which have both strings idle or one idle and the other zone producing a low fluid rate by standard gas lift, are candidates for a parallel two-step gas lift installation (see Fig. 3.469). The sizes of the slim gas injection pipes, gas injection levels, and additional depth specifications must be determined according to the two-step gas lift design.

In some cases, a large sucker-rod pump will produce at the same rate as the two-step gas lift method. However, this method needs periodical rig service, and oil leaks from the stuffing boxes cause pollution problems in the lake.

3.42832 Concentric two-step completion

Before analyzing the two-step installation, review the dual gas lift installation. A dual installation is one whereby two zones are to be produced to the surface through two separate flow paths. This may be accomplished by two parallel strings of tubing or by concentric strings.

The main trouble with dual gas lift installations occurs when there are large differences between the reservoir data of the two zones to be lifted. This problem will not exist in a two-step gas lift installation because only one zone will be lifted through the two strings.

Figure 3.470 is a multi-concentric two-step gas lift installation. The fluid from the formation passes through the tubing at the packer depth, then it is crossed over into the liner annulus. The direction and type of flow can be visualized in Figure 3.470.

The gas lift valves proposed are of the concentric type. Also, it is proposed in this installation to run a special gas lift mandrel with a recoverable gas lift valve. This valve will be important when the reservoir data changes, because this valve controls the gas injection and the flowing BHP of the reservoir. Figure 3.471 is a cross-sectional and longitudinal view for a 9⅝-in. and 7-in. casing size. Cross-sectional views are noted in Figures 3.472 and 3.473.

Table 3.461 shows special tubing and casing specifications, and Table 3.462 gives the annular area for the possibility of running concentric pipes.

3.4284 Two-step gas lift examples

The correlation used in this study to find the pressure traverse values is the modified Hagedorn and Brown correlation, which uses the Griffith correlation for the bubble flow region. For annular flow, the hydraulic diameter was used in determining the relative roughness and Reynolds number. The equivalent diameter was used for all the remaining equations.

Faustinelli worked the following examples which have this physical completion equipment:

(1) 9⅝ in. casing
(2) Tubing of 2.992 in. I.D. from the formation depth to the surface for a natural flow well or for a standard gas lift well
(3) The concentric two-step completion is the same as Fig. 3.470
  (a) Tubing #1 = 2.992 in. I.D. from formation depth to packer depth
  (b) Cross-over N:1: connects Tubing #1 with Annulus #1
  (c) Annulus #1: 6.276 in. × 5.000 in. from packer depth to the window
  (d) Window: connects annulus #1 with casing annulus
  (e) Annulus #2: 8.681 in. × 7.000 in. from window to Cross-over N:2 for liquid downstream flow
  (f) Cross-over N:2: connects Annulus #2 to Tubing #2
  (g) Tubing #2: from Cross-over N:2 to surface
(4) The parallel two-step completion consists of two strings of 2.992 in. I.D. tubing. A slim pipe is run in each string for gas injection (see Fig. 3.469).

Table 3.463 is a summary of the production obtained by natural flow, standard gas lift, and two-step gas lift for different reservoir data. In the two-step gas lift installation, a higher rate is obtained by the concentric rather than the parallel method. For example, run no. 1-b (plotted in Fig. 3.474 and noted in Table 3.459) gives a rate of 3421 bopd with a concentric installation, while a rate of 2195 bopd is obtained for the parallel installation, and a rate of 1079 bopd by the standard gas lift method. This is because the annular cross-sectional area for the multiphase flow is larger for the concentric than for the parallel installation. The concen-

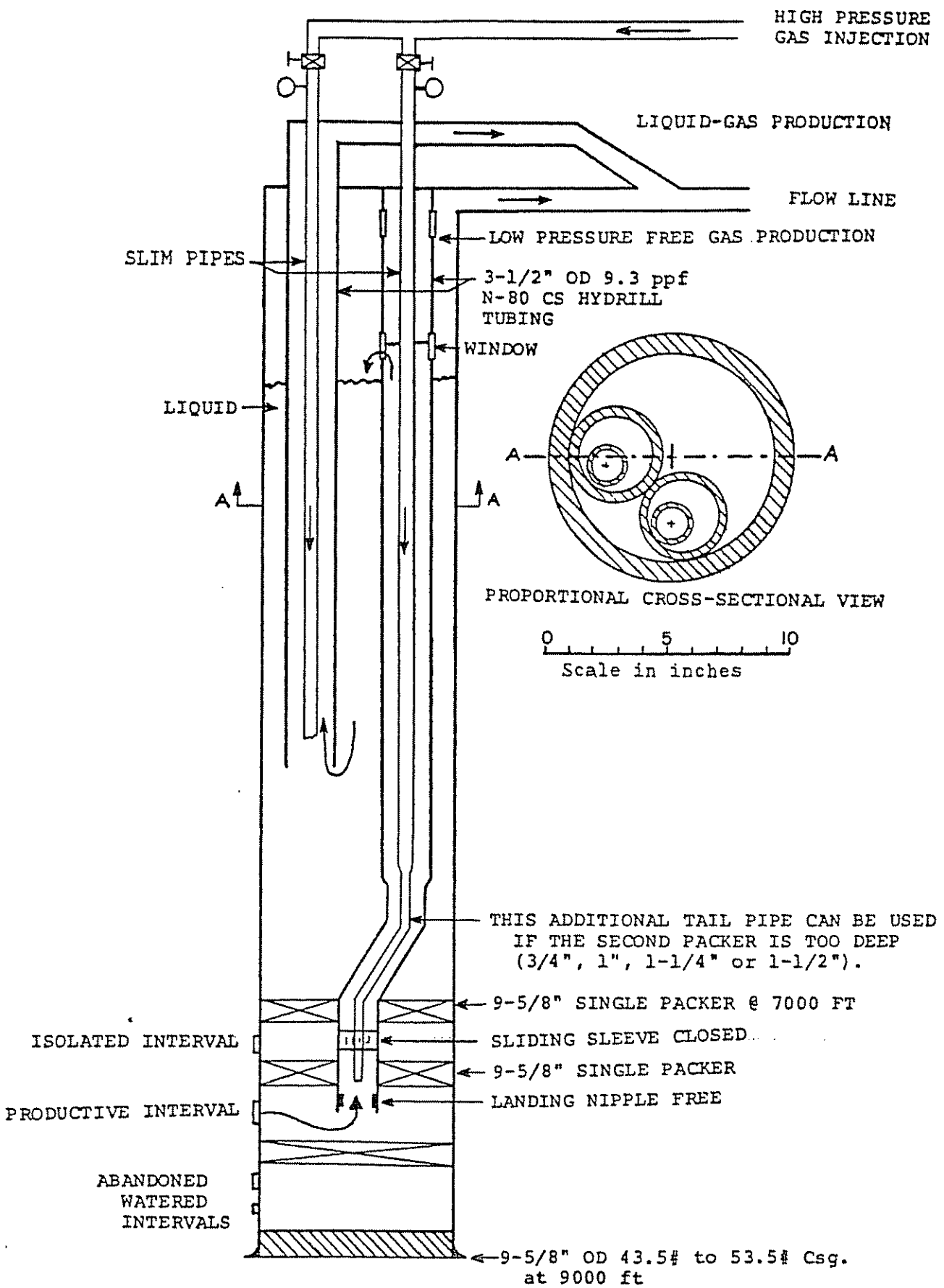


Fig. 3.469 Parallel Two-Step Gas Lift Installation for A Casing of 9 ⅝ in. in Lake Maracaibo, Venezuela

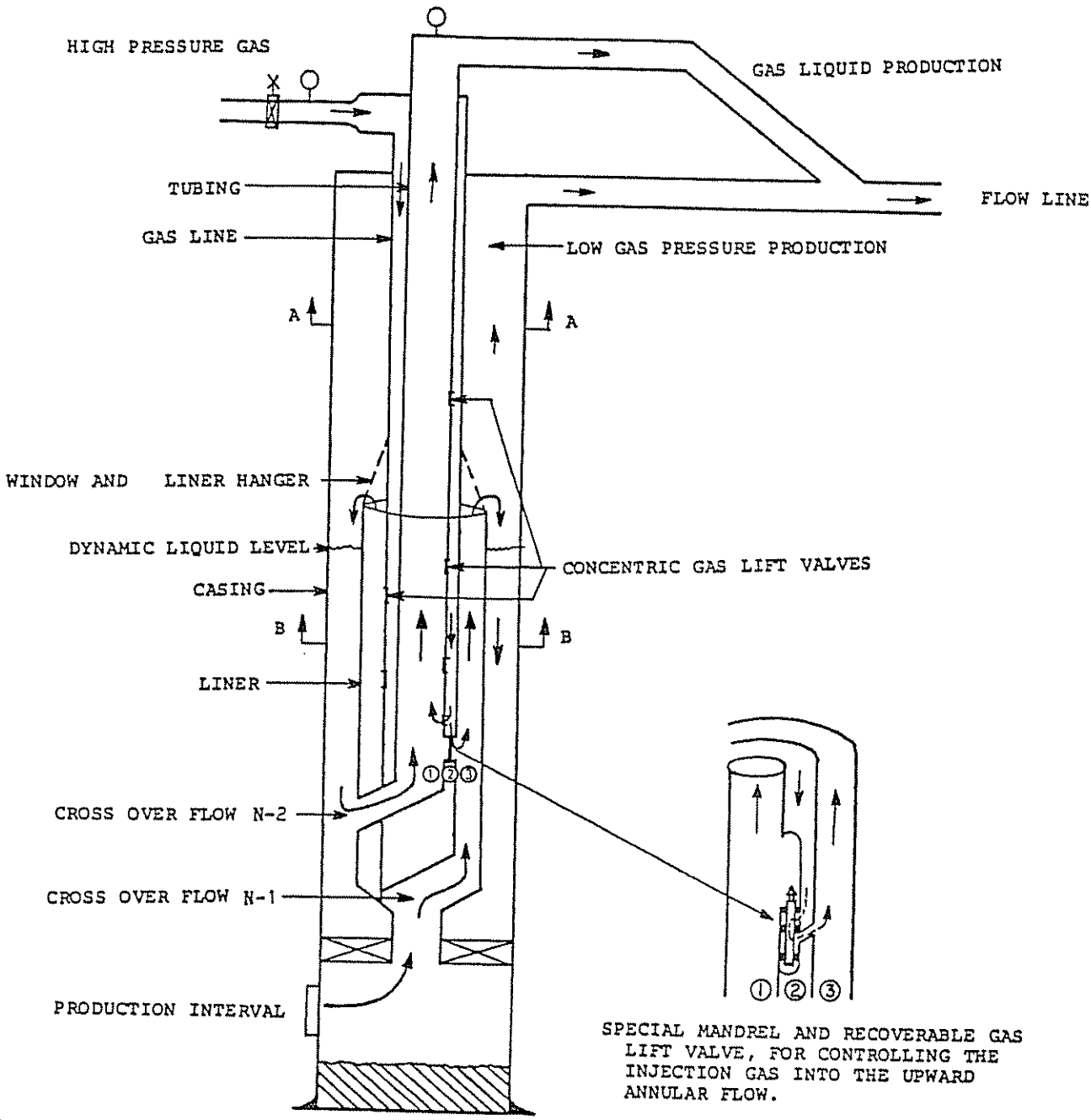


Fig. 3.470 Scheme of A Multi-Concentric Two-Step Gas Lift Installation

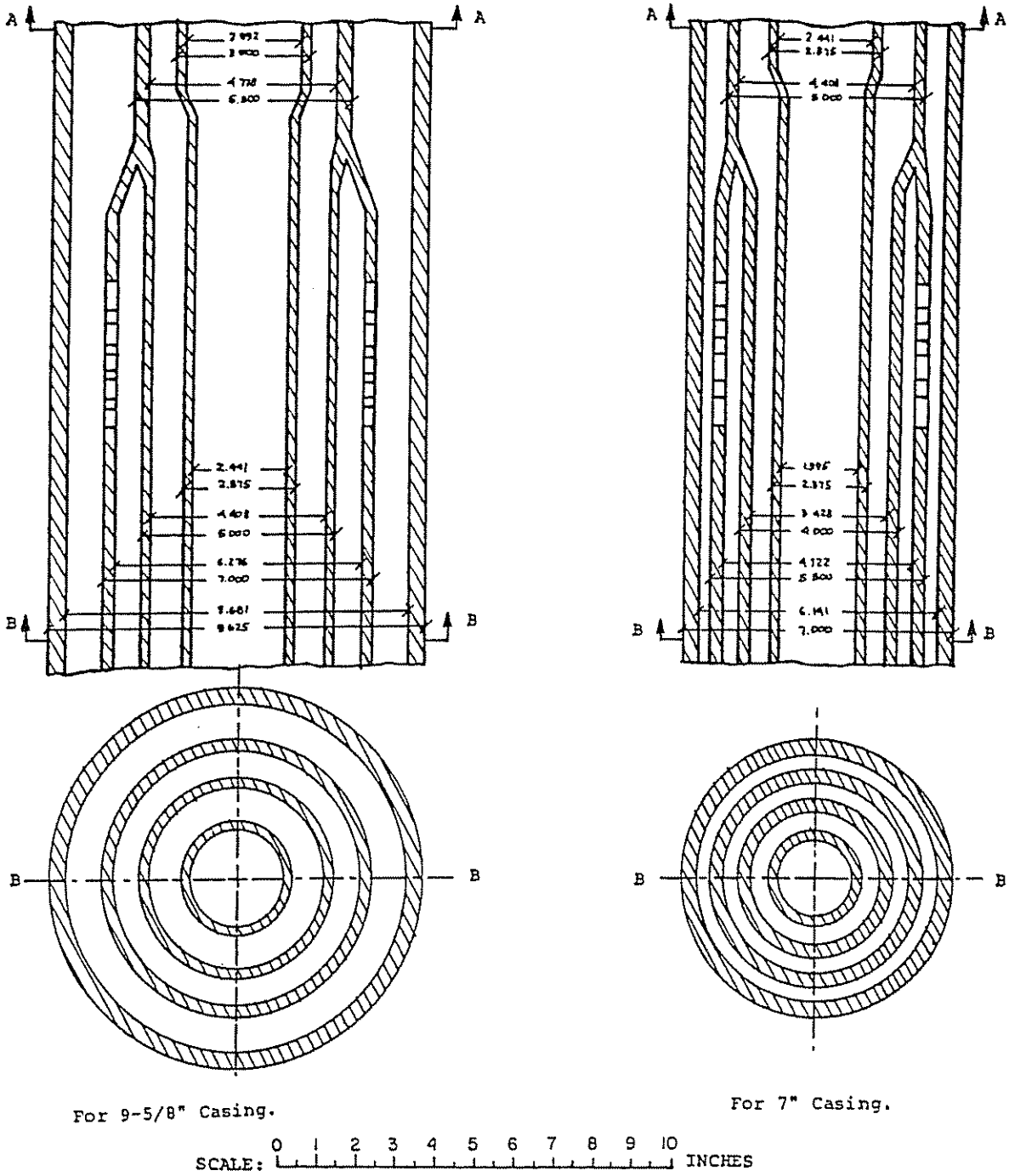


*Fig. 3.471 * Cross-Sectional and Longitudinal Views of Pipes in Concentric Two-Step Gas Lift Installation*

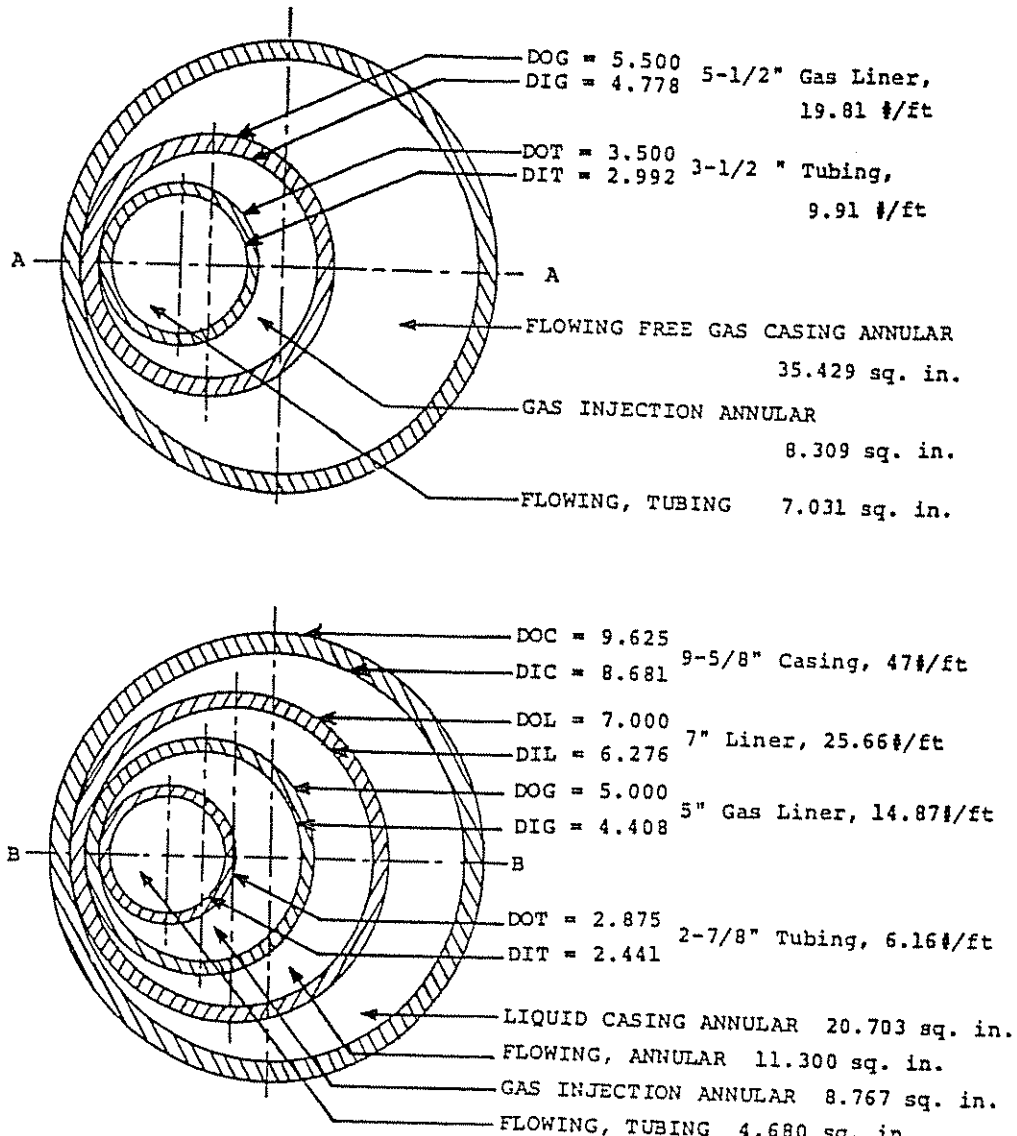


Fig. 3.472 Actual Eccentric Cross-Sectional Views for 9⅝-in. Casing (ref. to Fig. 3.471)

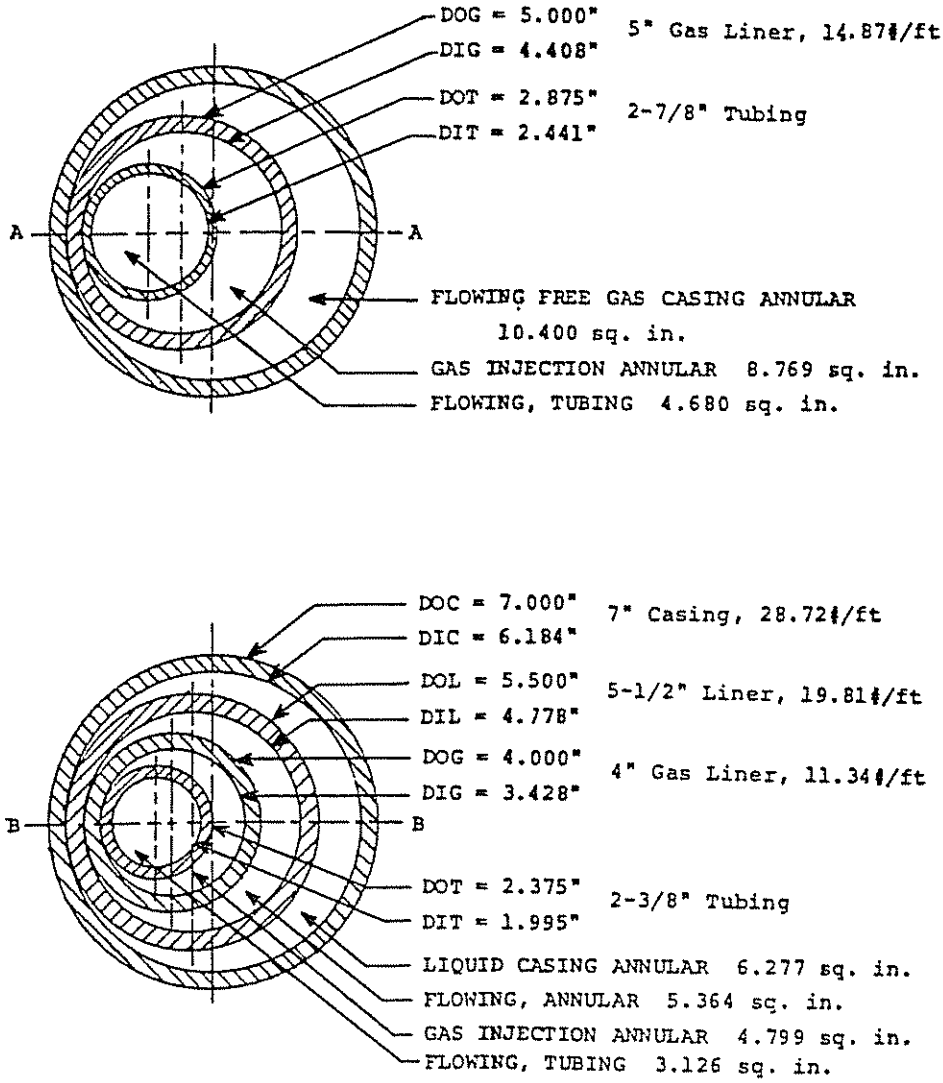


Fig. 3.473 Actual Eccentric Cross-Sectional Views for 7-in. Casing (ref. to Fig. 3.471)

TABLE 3.461
SPECIAL TUBING AND CASING SPECIFICATIONS

Nominal Size	Tubing O.D.	"IJ" Integral Joint				External Upset Tubing				Non-Upset Tubing				Improved Buttress-Tread Tubing				Streamline Teflon Seal Integral Joint			
		W.	I.D.	D.D.	O.D.	W.	I.D.	D.D.	O.D.	W.	I.D.	D.D.	O.D.	W.	I.D.	D.D.	O.D.	W.	I.D.	D.D.	O.D.
¾	1.050					1.20	.824	.730	1.554	1.14	.824	.730	1.165								
1	1.315	1.72	1.049	.955	1.555	1.80	1.049	.955	1.677	1.70	1.049	.955	1.554					1.20	.824	.730	1.310
1¼	1.660	2.33	1.380	1.286	1.885	2.40	1.380	1.286	2.040	2.30	1.380	1.286	1.845					1.80	1.049	.555	1.550
1½	1.900	2.76	1.610	1.516	2.115	2.90	1.610	1.516	2.375	2.75	1.610	1.516	2.104					2.40	1.380	1.286	1.880
	2.00																	2.90	1.610	1.516	2.110
2 1/16	2.063	3.25	1.751	1.657	2.330									3.40	1.670	1.576	2.300				
2	2.375					4.70	1.995	1.901	2.910	4.60	1.995	1.901	2.670	4.60	1.995	1.901	2.700	3.25	1.750	1.656	2.320
														5.80	1.867	1.773	2.700	4.70	1.995	1.901	2.700
2½	2.875					6.50	2.441	2.347	3.460	6.40	2.441	2.347	3.235	6.40	2.441	2.347	3.220				
														7.90	2.323	2.229	3.220				
3½	3.500													8.60	2.259	2.165	3.220				
														9.20	2.992	2.867	3.865				
														10.20	2.992	2.797	3.865				
4	4.000													14.50	2.602	2.477	3.865				
4½	4.500													10.90	3.476	3.351	4.400				
														12.60	3.958	3.833	4.920				

where:

w = weight, pound per foot
I.D. = average internal diameter
D.D. = drift diameter
O.D. = outside diameter of the joint

TABLE 3.461 (CONT'D)
SPECIAL TUBING AND CASING SPECIFICATIONS

Nominal Size	Tubing O.D.	Hydril "95" Joint				Hydril "CS" Joint				Hydril "PH-6" Joint				Atlas Bradford "DS-HT" Joint				Atlas Bradford "DSS-HT" Joint			
		W.	I.D.	D.D.	O.D.	W.	I.D.	D.D.	O.D.	W.	I.D.	D.D.	O.D.	W.	I.D.	D.D.	O.D.	W.	I.D.	D.D.	O.D.
¾	1.050					1.20	.824	.730	1.327												
1	1.315					1.80	1.049	.955	1.552					1.80	1.049	.955	1.520	1.80	1.049	.955	1.522
						2.25	.957	.848	1.600												
1¼	1.660	2.40	1.380	1.286	1.898	2.40	1.380	1.286	1.883					2.40	1.380	1.286	1.858	2.40	1.380	1.286	1.883
1½	1.900	2.90	1.610	1.516	2.134	2.90	1.610	1.516	2.113					2.90	1.610	1.516	2.094	2.90	1.610	1.516	2.113
	2.000																				
2 1/16	2.063	3.40	1.750	1.656	2.325	3.40	1.750	1.656	2.330												
2	2.375	4.70	1.995	1.901	2.656	4.70	1.995	1.901	2.700	5.95	1.867	1.773	2.9063	3.40	1.750	1.656	2.260	3.40	1.750	1.656	2.330
						5.30	1.939	1.845	2.700	6.20	1.653	1.759	2.9375	4.70	1.995	1.901	2.594	4.70	1.995	1.901	2.700
										7.70	1.703	1.609	3.125								
2½	2.875	6.50	2.441	2.347	3.177	6.50	2.441	2.347	3.220	7.90	2.323	2.229	3.375	6.50	2.441	2.347	3.156	6.50	2.441	2.347	3.200
										8.70	2.259	2.165	3.500								
										9.50	2.195	2.101	3.625								
										11.00	2.065	1.971	3.750								
3½	3.500					9.30	2.992	2.867	3.865	10.30	2.992	2.797	4.1875								
						10.30	2.992	2.797	3.865	13.30	2.764	2.639	4.3125								
										12.95	2.750	2.625	4.3125								
4	4.000									14.00	3.340	3.215	4.625								
4½	4.500									15.10	3.826	3.701	5.125								
										20.00	3.640	3.515	5.3125								

TABLE 3.461 (CONTD.)
SPECIAL TUBING AND CASING SPECIFICATIONS

	Hydril "FJ" Tubing & Casing Joint				Hydril Tripleseal Casing Joint				Extremeline Casing			
	W.	I.D.	D.D.	O.D.	W.	I.D.	D.D.	O.D.	W.	I.D.	D.D.	O.D.
2⅜	4.43	1.995	1.901	2.375								
2⅞	6.16	2.441	2.347	2.875								
3½	8.81	2.992	2.867	3.500								
	9.91	2.992	2.797	3.500								
4	11.34	3.428	3.303	4.000								
4½					10.79	4.026	3.901	4.6875				
					11.35	4.000	3.875	4.6875				
					12.24	3.958	3.833	4.6875				
	13.04	3.920	3.795	4.500	13.04	3.920	3.795	4.6875				
	14.98	3.826	3.701	4.500	14.98	3.826	3.701	4.750				
5					12.83	4.494	4.369	5.21875				
	14.87	4.408	4.283	5.000	14.87	4.408	4.283	5.21875	15.00	4.276	4.183	5.360
	17.93	4.276	4.151	5.000	17.93	4.276	4.151	5.250	18.00	4.276	4.151	5.360
5½					13.70	5.012	4.887	5.750				
					15.35	4.950	4.825	5.750	15.50	4.950	4.721	5.78
	16.87	4.892	4.767	5.500	16.87	4.892	4.767	5.750	17.00	4.892	4.686	5.78
	19.81	4.778	4.653	5.500	19.81	4.778	4.653	5.750	20.00	4.778	4.653	5.78
	22.54	4.670	4.545	5.500	22.54	4.670	4.545	5.78125	23.00	4.670	4.545	5.78
	25.54	4.548	4.423	5.500								
6⅝									24.00	5.921	5.766	6.930
									28.00	5.791	5.666	6.930
									32.00	5.675	5.550	6.930
7					19.54	6.456	6.331	7.3125				
	22.63	6.366	6.241	7.000	22.63	6.366	6.241	7.3125	23.00	6.366	6.156	7.310
	25.66	6.276	6.151	7.000	25.66	6.276	6.151	7.3125	26.00	6.276	6.151	7.310
	28.72	6.184	6.059	7.000	28.72	6.184	6.059	7.3125	29.00	6.184	6.059	7.310
	31.68	6.094	5.969	7.000	31.68	6.094	5.969	7.34375	32.00	6.054	5.969	7.310
	34.58	6.004	5.879	7.000	34.58	6.004	5.879	7.34375	35.00	6.004	5.879	7.390
	37.26	5.920	5.795	7.000	37.26	5.920	5.795	7.34375	38.00	5.920	5.795	7.390
7⅝	25.56	6.969	6.844	7.625	25.56	6.969	6.844	7.9375	26.40	6.969	6.755	7.920
	29.04	6.875	6.750	7.625	29.04	6.875	6.750	7.9375	29.70	6.875	6.750	7.920
	33.04	6.765	6.640	7.625	33.04	6.765	6.640	8.000	33.90	6.765	6.640	7.920
	38.05	6.625	6.500	7.625	38.05	6.625	6.500	8.000	39.00	6.625	6.500	7.920
8⅝	31.10				31.10	7.921	7.796	8.9375	32.00	7.921	7.710	9.030
	35.14				35.14	7.825	7.700	8.9375	36.00	7.825	7.700	9.030
	39.29				39.29	7.725	7.600	8.9375	40.00	7.725	7.600	9.030
	43.39				43.39	7.625	7.500	9.03125	44.00	7.625	7.500	9.030
	48.00				48.00	7.511	7.386	9.03125	49.00	7.511	7.386	9.030
9⅝	34.86				34.86	8.921	8.765	10.000				
	38.94				38.94	8.835	8.679	10.000	40.00	8.835	8.650	10.020
	42.70				42.70	8.755	8.599	10.000	43.50	8.755	8.599	10.020
	46.16				46.14	8.681	8.525	10.000	47.00	8.681	8.525	10.020
	52.85				52.85	8.535	8.379	10.0625	53.50	8.535	8.379	10.020
10¾	38.88				38.88	10.050	9.804	11.1875				
	44.22				44.22	9.950	9.794	11.1875	45.50	9.950	9.794	11.460
	49.50				49.50	9.850	9.694	11.1875	51.00	9.850	9.694	11.460
	54.21				54.21	9.760	9.604	11.1875	55.50	9.760	9.604	11.460
	59.40				59.40	9.660	9.504	11.250	60.90	9.666	9.504	11.460
	64.53				64.53	9.560	9.404	11.250				

TABLE 3.462
SECTIONAL AREA FOR VARIOUS ANNULI

Nominal Diam.		¾"	1"	1¼"	1½"	—	2"	2½"	3½"	4"	4½"	5"	5½"	7"	7⅝"	8⅝"	9⅝"
Drift Diam.		.730	.955	1.286	1.516	1.576	1.901	2.347	2.797	3.303	3.701	4.151	4.548	5.920	6.623	7.511	8.535
Average Int. Diam.		.824	1.049	1.380	1.610	1.670	1.995	2.441	2.992	3.428	3.873	4.342	4.722	6.141	6.809	7.721	8.701
Nominal Diam.	Outside Diam.																
¾"	1.050			.6298	1.1699	1.3245	2.2600	3.8139	6.1650	8.3634	10.915	13.941	16.646	28.753	35.547	45.955	58.595
1"	1.315				.678	.832	1.768	3.322	5.673	7.871	10.423	13.449	16.154	28.261	35.055	45.463	58.103
1¼"	1.660						.962	2.516	4.867	7.065	9.617	12.643	15.348	27.455	34.245	44.656	57.296
1½"	1.900							1.844	4.196	6.394	8.946	11.972	14.677	26.783	33.578	43.985	56.625
*	2.00							1.538	3.889	6.088	8.640	11.666	14.371	26.477	33.271	43.679	56.319
2"	2.375								2.601	4.799	7.351	10.378	13.0825	25.1885	31.983	42.391	55.031
2½"	2.875									2.738	5.289	8.315	11.020	23.127	29.921	40.329	52.969
3½"	3.500										2.160	5.186	7.891	19.998	26.792	37.200	49.839
4"	4.000											2.241	4.946	17.053	23.847	34.254	46.894
4½"	4.500												1.608	13.715	20.509	30.916	43.556
5"	5.000													9.984	16.778	27.186	39.826
5½"	5.500													5.861	12.655	23.062	35.702
7"	7.000															8.336	20.976
7⅝"	7.625																13.797
8⅝"	9.030																
9⅝"	10.000																
	0.000	.533	.864	1.496	2.036	2.190	3.126	4.680	7.031	9.229	11.781	14.807	15.512	29.619	36.413	46.821	59.461

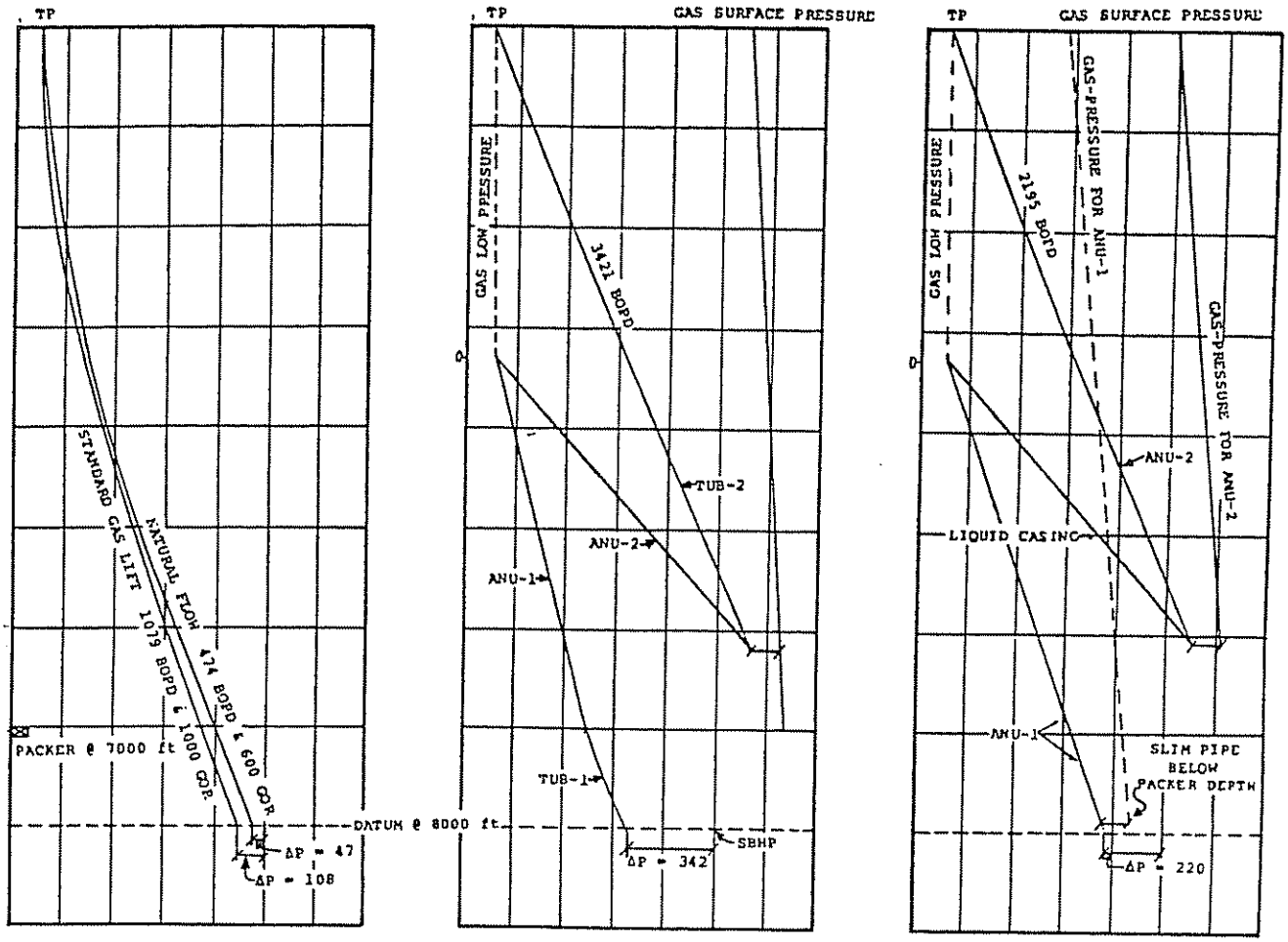
*This size is not common and is not nominal size. This tubing exists on Improved Buttress-thread tubing from Vinson Supply Company.

TABLE 3.463
NATURAL FLOW, STANDARD, AND TWO-STEP GAS LIFT COMPARISONS

	Reservoir data					High pressure gas source	
Run	SP	SBHP	PI	%W	FGOR	Gas Work Pressure	Total GLR
1-a-b-c	100	1000	10.0	0.0	600	1100	1000
2	100	1000	10.0	40.0	600	1100	1000
3	100	1750	1.0	0.0	600	1100	1000

	Natural flow		Standard gas lift		
Run	Q_L	Q_o	Q_L	Q_o	Gas injection MMSCFPD
1-a-b-c	474	474	1079	1079	.435
2	(cannot	flow)	842	505	.539
3	738	738	885	885	.354

	Two-step gas lift						
	Concentric			Parallel			
Run	Q_L	Q_o	Gas injection MMSCFPD	Q_L	Q_o	Gas injection MMSCFPD	Slim gas inj. pipes
1-a	3421	3421	4.790	1380	1380	1.932	1¼"
1-b	3421	3421	4.790	2195	2195	3.073	1"
1-c	3421	3421	4.790	(Slim pipes of ¾" are too small for gas injection, see Appendix A)			¾"
2	3055	1833	5.009	1272	763	2.087	1¼"
3	1388	1388	1.944	(none calculated)			



(a) Natural Flow and Standard Gas Lift Method.

(b) Concentric Two-Step Gas Lift Method.

(c) Parallel Two-Step Gas Lift Method.

Fig. 3.474 Pressure Traverse Curves for (a) Natural Flow and Standard Gas Lift Method, (b) Concentric Two-Step Gas Lift Method, (c)Parallel Two-Step Gas Lift Method

tric two-step installation takes advantage of the larger circular cross-sectional area of the casing string.

3.4285 Summary and conclusions

Due to the complexity of the equipment involved, the "two-step" gas lift installation will be tried on only a few wells. The well must have a large casing and convenient different tubing diameters. These pipes must be flush joint to avoid reducing the annular cross-sectional area in the tubing joint. If the multi-concentric type is used, two special cross-overs are needed. No rig problems are expected in running the completion equipment because one string can be run independent of the other.

Faustinelli offered the following conclusions:

(1) The two-step gas lift method is recommended only for those wells with relatively low shutin BHP and high PI, assuming that the cost is not prohibitive and the installation is not too complex.

(2) This new method often can convert an idle standard gas lift well into an economical one; if the well is still producing at a low rate by standard gas lift, it can double or triple its rate.

(3) The two-step gas lift method is recommended in those wells which are producing—by natural flow or conventional gas lift—with low wellhead pressures and a GLR of 900 Scf/B or higher.

Faustinelli recommended that the two-step gas lift installation be tried in a large casing well which is already producing a low rate by a continuous standard gas lift method with a GLR of approximately 1000 Scf/B. The increased production will serve as an incentive for further two-step gas lift installations. To the author's knowledge, this method has not been tried, but is presented here to show the increase in production that can be obtained if the installation does not become unwieldy.

3.429 Spacing of continuous flow gas lift valves

3.4291 Introduction

The spacing of continuous flow gas lift valves depends on many factors. In the literature are several

spacing procedures, and most of them normally will do the job. The principal purpose of spacing is to reach the operating gas lift valve and maintain single point injection of gas at that point. The method for determining the point of gas injection has been described in Section 3.4231. As mentioned previously, the single point injection of gas by continuous flow is more efficient than the injection of gas through more than one valve. Universal spacing for wells of similar characteristics will suffice in certain areas, but each well should normally be considered separately. Remember, the valves cannot be spaced too closely near the point of gas injection. The author has run installations with valves spaced three joints of tubing apart and those with valves spaced 150-200 feet apart. For high productivity wells, the uncovering of another 100 feet may mean 100-500 B/D more production. The following factors have an influence on the spacing of gas lift valves for continuous flow.

(1) *Type of gas lift valve.* The type of gas lift valve has an effect on the spacing of the valves. For example, fluid operated valves are normally spaced closer than pressure operated valves. Also, certain types of valves require a drop in surface operating pressure between valves. This is an absolute requirement for valves nearing a balanced condition (no tubing pressure effect) such as the valve incorporating a flexible element. If an unbalanced gas lift valve is used, it is possible to set valves with the same surface operating pressure down the hole. However, I recommend at least a 10 psi drop down the well per valve in either surface opening or closing pressure for single completions and when sufficient pressure is available. By necessity, in duals, we set valves at either the same surface opening or surface closing pressures. By design, many continuous flow valves do not have constant closing pressures and therefore must be set on opening pressures.

The various combination valves using both a pressure-operated and fluid-operated section must be spaced based on the operation of the fluid operated section.

(2) *Whether or not the well is to be unloaded to a lower pressure.* In many cases the well is unloaded to a lower wellhead pressure for the first valve only and then turned into the normal surface production facilities. In some cases the well can be unloaded to bottom before turning the well into its surface production facilities. This allows deeper spacing for each valve. However, the lower pressure facilities must conform to regulations concerning environment.

(3) *The static fluid level in the well and whether or not the well was loaded with a "kill" fluid prior to pulling tubing:* If the static fluid level is lower than $(P_{so}-P_{wh})/G_s$ the first valve may be placed at the static fluid level. It should be placed no lower than the static fluid level.

(4) *Whether or not a special kick-off pressure is available to unload the well, or if the well is to be unloaded with normal field operating pressure.*

(5) *Whether or not an unlimited gas supply is available for unloading purposes:* This influences the spacing by determining whether or not the minimum unloading gradient can be attained in the tubing string.

The techniques used in continuous flow valve design vary widely throughout the industry. However, the basic objectives should be the same:

(1) To inject gas as deep as possible.
(2) To conserve as much casing operating pressure as possible.
(3) To ensure all the upper unloading valves are closed after the final point of injection is established.
(4) To be able to open a valve with kickoff pressure or less, regardless of the liquid level in the tubing.

3.4292 Universal design and spacing for all types of continuous flow gas lift valves

3.42921 Introduction

A spacing procedure that will work for all types of valves is necessary in many cases. For example, mandrels for retrievable gas lift valves may be installed in wells upon initial completion. The decision as to the type of valve (pressure operated, fluid operated, or combination fluid-pressure operated valve) may not yet be decided.

In the course of changing supervisors in an area and/or changing well conditions, a preference of one type of valve over another may be made. Thus, if the mandrels are properly spaced, any type of gas lift valve may be installed. If a strict pressure-operated valve spacing has been used, the installation of fluid operated valves may be impossible.

The following example problem will be worked for all types of valves utilizing the same spacing procedure for several cases and slightly different spacing for other cases:

Given:
Tubing size = 2-⅞" O.D.
γ_g = 0.70
Well loaded with kill fluid of gradient = 0.40 psi/ft
GOR = 400 scf/B
Well produces 50% water
GLR = 200 scf/b
$\bar{P}_R$ = 1920 psi
°API = 35
Bubble point pressure = 1500 psi
PI = 5 = linear until reaching P_b
P_{wh} = 120 psig (constant)
P_{so} = 900 psi (950 available)
P_{ko} = 1000 (1050 available)
Depth = 8000 ft
Desired rate = maximum
Temp bottom = 170°F.
Geothermal temperature gradient = 1.2°F./1000 ft
(109°F surface)

3.42922 Standard pressure operated valves—constant surface opening pressure

The following step-wise procedure serves as a guide in spacing and setting the gas lift valves for pressure operated unbalanced valves. Refer to Figure 3.477 for the complete spacing procedure.

(1) Find the production rate possible. This has been described earlier and is found to be 1000 B/D as noted in Figure 3.475. This may vary slightly, depending upon the multiphase flow correlation utilized.

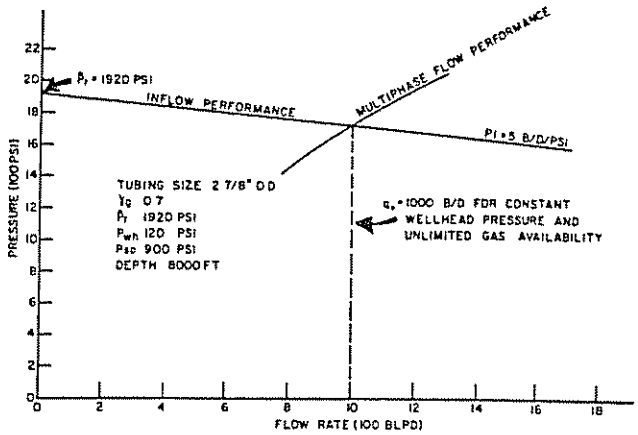


Fig. 3.475 Pressure Flow Rate Diagram to Determine Rate

As described earlier there are many systems, parameters, or conditions considered when constructing a pressure-flow rate diagram, which determines the rate possible from a gas lift well. All system parameters present in the field operation must be accounted for in the pressure-flow rate analysis, or the predictions based on that analysis may not be valid. The gas lift designer must then design a gas lift system which will create the conditions shown to be best or desirable by the pressure-flow rate analysis.

For example, several GLR's were evaluated with a pressure-flow rate diagram. Figure 3.475 shows that for the example, the maximum production rate of 1000 BLPD occurs at a GLR of about 1000 Scf/B. The goal is to design a gas lift system which will create a 1000 Scf/B GLR when producing 1000 BLPD and, given the proper design, the operator should adjust the injection gas until maximum production is obtained which should be near to the GLR predicted by the pressure-flow rate diagram. The designer controls this parameter, gas flow, with valve type and port size. Thus, pressure-flow rate analysis can be used to help design as well as aid in the operation of a gas lift system.

Another parameter which must be assumed to construct a pressure-flow rate diagram is the so called "operating differential" (see Fig. 3.476A). In other words, it is the difference in the operating gas injection pressure at depth and the established flowing tubing pressure at depth. The results of a pressure-flow rate analysis are usually very sensitive to the selection of this value which must be selected prior to the analysis. Figure 3.476-B shows how the estimated production rate can vary significantly with change in the assumed ΔP. Notice that as well productivity increases, this parameter becomes more and more important. Figure 3.476C shows a typical system graph that must be prepared for each ΔP selected.

The gas lift designer must control this operating differential with the gas lift valve spacing. Figure 3.476-D depicts the relationship between valve spacing and operating differential. The solid line shows that if only Valves 1 and 3 were present, the allowable operating differential could be as high as 200 psi before transfer to the next lower valve would occur. Note that the differential may be smaller, but the designer is only ensuring a ΔP of 200 psi.

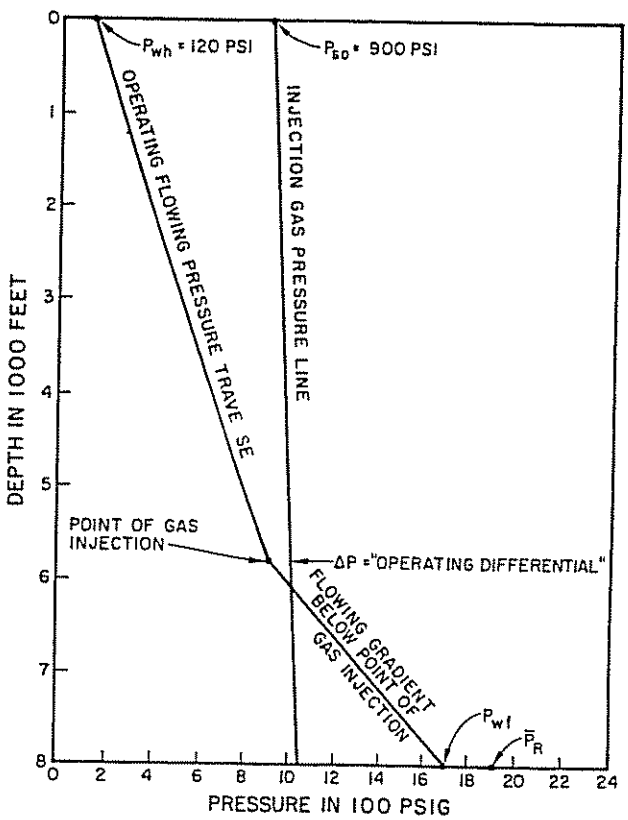


Fig. 3.476-A Illustration of Operating Differential "Δ P"

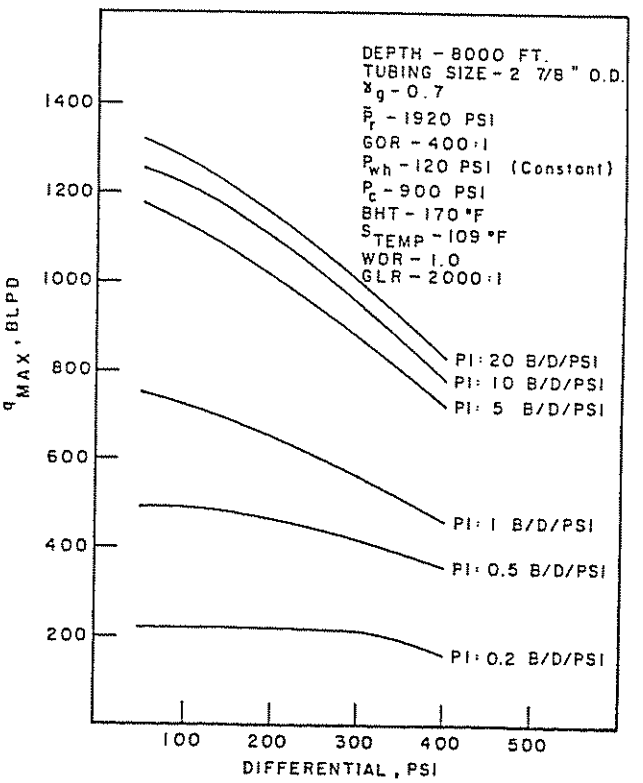


Fig. 3.476-B Operating Differential vs. Flow Rate

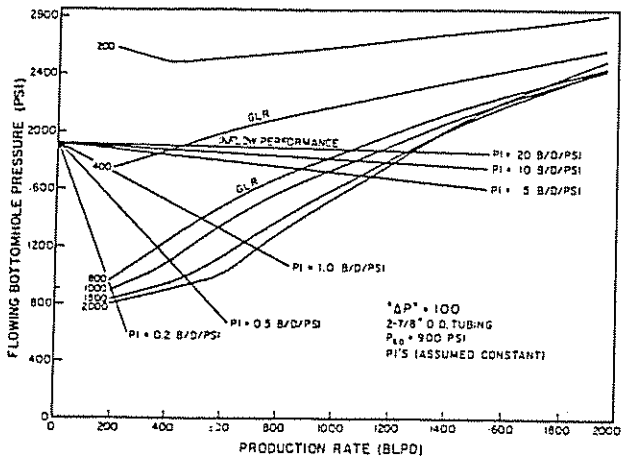


Fig. 3.476-C Systems Graph for Δ P - 100 PSI

The dashed gradient lines show that if a gas lift valve were located at position 2, halfway between Valves 1 and 2, the maximum allowable operating differential would be reduced to 100 psi. The maximum differential created by the 1000-ft spacing is twice as large as that created by the 500-ft spacing. This is due to the linear relationship between valve spacing and the flowing gradient below the point of gas injection. Of course, this assumes the flowing gradient below the point of injection will be the same in both cases. This relationship is:

$$\Delta P_{diff} = G_f \Delta D_v$$

G_f = Flowing gradient below the injection, psi/ft

ΔD_v = Distance between valves, ft

For example, in the problem with an operating differential of 100 psi, the gas lift valve spacing in the vicinity of the point of injection should be close enough to ensure a maximum allowable differential of 100 psi. The maximum valve spacing in the vicinity of the point of injection should be:

$$\Delta D_v = \frac{P_{diff}}{G_S} = \frac{100}{.40} = 250 \text{ ft}$$

This operating differential is often confused with the pressure drop across a certain port size to pass the necessary gas. However, it is an established operating differential controlled by the valve spacing. Once a desireable operating differential is selected, the gas lift designer must select a port size sufficient to pass the necessary amount of gas with the selected casing pressure to tubing pressure differential. If by reason of some physical constraint a gas lift valve with a large enough port cannot be used, then a larger operating differential must be selected.

(2) The point of gas injection is 5800 ft and is found by the previously described procedure for 1000 B/D. (Refer to Figure 3.477 for a completed design).

(3) Draw a design tubing pressure line from the surface to the point of gas injection. This line should start from a surface pressure of P_{wh} + .20 (P_{so}) or P_{wh} + 200, whichever is greatest. For this problem it is P_{wh} + 200 = 320 psi. Some designers may use P_{wh} + 0.25 (P_{so}). A similar procedure was originally suggested by U.S. Industries.[21]

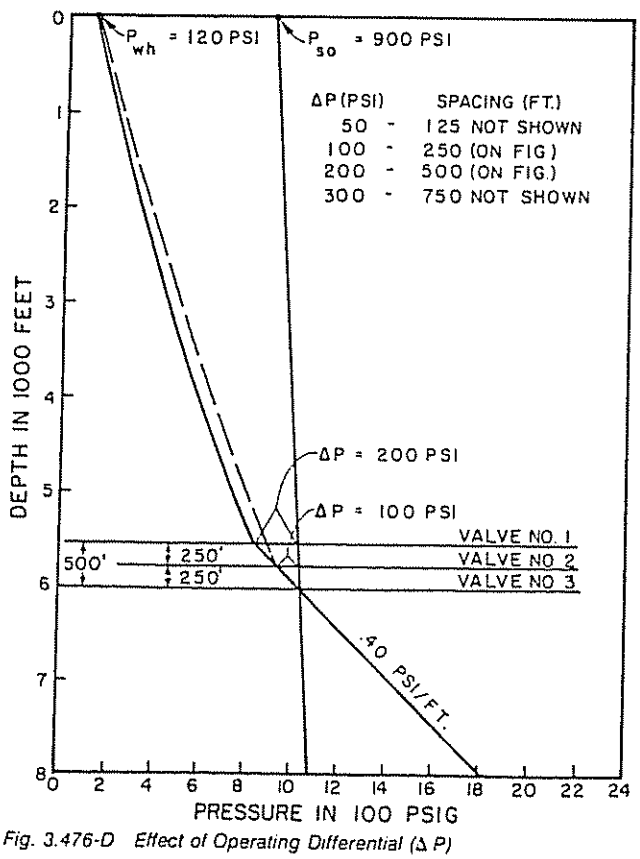


Fig. 3.476-D Effect of Operating Differential (Δ P)

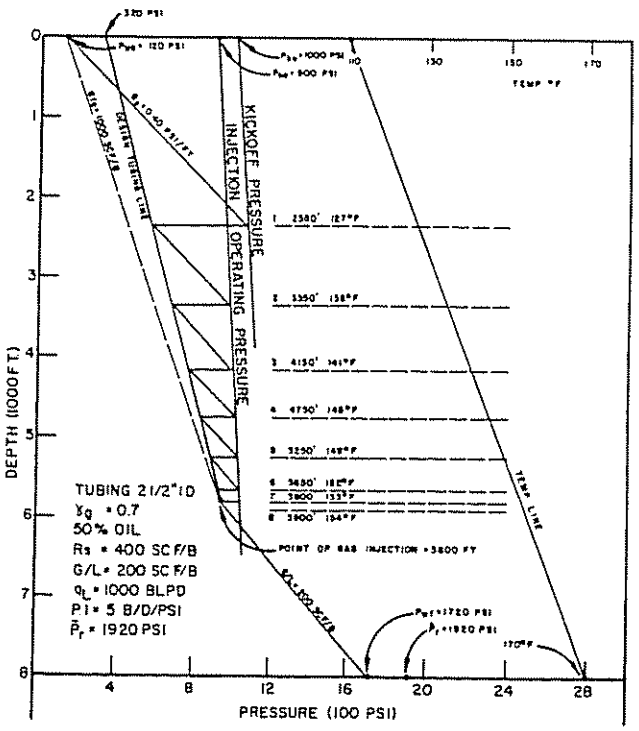


Fig. 3.477-A Universal Spacing for Continuous Flow Gas Lift

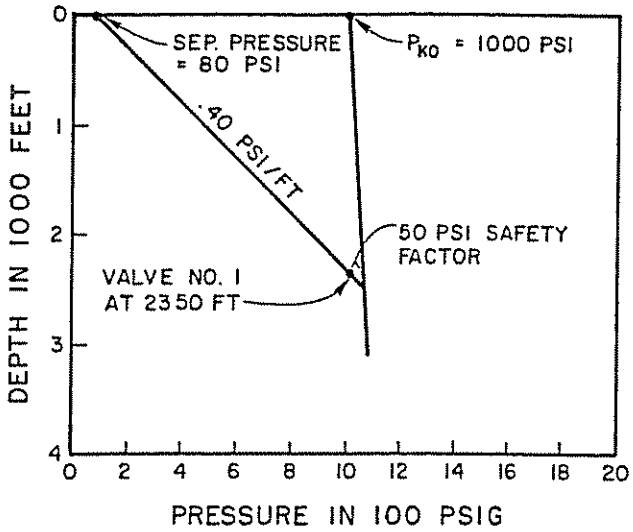


Fig. 3.477-B Location of First Valve

(4) Locate the first gas lift valve by drawing a line having the kill fluid gradient of 0.40 psi/ft. This line begins at the wellhead pressure of 120 psi and is extended downward until intersecting the kick-off gas pressure line of 1000 psi which must also be drawn in downward from P_{ko} = 1000 psi, including the gas column weight of 26.2 psi/1000 ft. The first valve is located at 2350 ft. Since 1050 psi is available we are assured of sufficient safety factor. Figure 3.477B shows another commonly-used procedure for locating Valve #1.

(5) The rest of the valves are now spaced to the injection point as follows:

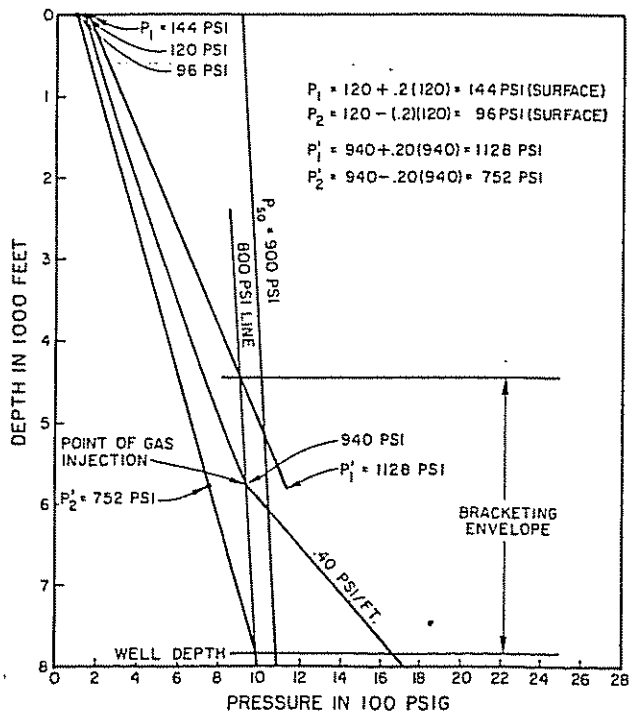


Fig. 3.478-A Bracketing Procedure

(a) From the location of Valve #1 (2350'), extend a line horizontally to the left until intersecting the design tubing pressure line.
(b) From (a) extend a line downward parallel to the previous downward line of gradient = .40 psi/ft until intersecting the operating pressure line which has 900 psi at the surface. This locates Valve #2 at 3350 ft.
(c) Repeat the procedure of (b) until reaching the point of injection or below. This locates valves as follows:

#1 — 2350'
#2 — 3350'
#3 — 4150'
#4 — 4750'
#5 — 5250'
#6 — 5650'
#7 — 5900'

(6) Bracket the valve spacing to permit a valve to be located at the point of gas injection and with at least one valve below this point. Depending upon well characteristics such as decreasing static bottom hole pressure and changing productivity, several additional valves may be needed below the present point of injection. The following adjusted spacing is recommended for this example:

1 — 2350
2 — 3350
3 — 4150
4 — 4750
5 — 5250
6 — 5650
7 — 5800
8 — 5900

A more realistic design procedure is offered for bracketing the point of gas injection. Since there are inherent inaccuracies in both horizontal and vertical multi-phase pressure predictions, the point of injection should be bracketed with the desired spacing as indicated by the pressure flow rate diagram. Figure 3.478 illustrates how this could be accomplished by assuming a certain error in the pressure predictions. Although 20% error has been assumed to illustrate this example, any percentage error could be used, and ±10% is more realistic.

The procedure for bracketing the point of injection is as follows:

(a) Plot the gas gradient line for the operating casing pressure.
(b) Subtract the desired operating differential (100 psi) from the surface operating pressure (900 psi). This gives 800 psi. Draw a line parallel to the surface operating pressure starting at 800 psi at the surface.
(c) Construct the flowing tubing pressure gradient for the desired production rate and GLR. Draw this gradient down from the flowing wellhead pressure (120 psi) until it meets the 800 psi casing pressure differential line.
(d) Calculate pressures which are 20% greater than the flowing wellhead pressure (P_1) and 20% less than the flowing wellhead pressure (P_2).

$P_1 = P_{wh} + 0.2\ P_{wh}$
$P_2 = P_{wh} - 0.2\ P_{wh}$

Locate and mark P_1 and P_2 on the pressure vs. depth graph.

(e) Find the pressure where the flowing tubing pressure gradient intersects the 800 psi casing pressure differential line (940 psi). This is the desired point of injection. Calculate pressures which are 20% greater than this pressure P_1' and P_2' on the pressure vs. depth graph.

(f) Draw a straight line between points P_1 and P_1' and P_2 and P_2'.

(g) Locate and mark the point where the line $P_1 P_1'$ intersects the 800 psi differential line (4450 ft). This depth is approximately where the bracketing envelope should start.

(h) Locate and mark the point where the line P_2P_2' intersects the 800 psi differential line (7850 ft.) This depth is approximately where the bracketing envelop could stop. However, for deeper wells, especially on offshore platforms, it may be advisable to valve all the way down to the packer.

Figures 3.478B and 3.478C show completed designs for 20% and 10% bracketing envelopes.

(7) Determine the operating temperature at each valve. This information should be placed on the same Fig. 3.477 as the spacing as follows:

(a) Mark the bottom hole temperature of 170°F. at 8000 ft.

(b) From Fig. 3.44 and a flow rate of 1000 B/D, tubing size of 2½", and geothermal gradient of 1.2°F./100 ft, surface flowing temperature is 109°F. Mark this temperature at the top (zero depth) on Fig. 3.477 and connect 109°F. at the top by a straight line with 170°F. at bottom. For high flow rate wells, the procedure recommended by Shiu should be used to allow for the non-linear change in temperature.

The temperature at each valve is:

Valve no.	Temp.	Adjsuted temp.
1	127	127
2	134.5	135
3	140.6	141
4	145	145
5	149	149
6	152	152
7	153	153
8	154	154

(8) Determine the injection gas volume necessary. The required total gas-liquid ratio is 1000 scf/B for a rate of 1000 B/D. The injection gas volume needed is (1000 − 200) (1000) = 800,000 scf/D.

(9) Determine the port size of the operating gas lift valve. Refer to Figure 3C.1 or Figures 3C.2 through 3C.26 in the Appendix. Note that a correction is required for both gas gravity and temperature.

Correction factor = $0.0544\ \sqrt{\gamma_g T}$ for Figure 3C.1 (Appendix) where T = °R

The temperature at the operating valve is 153°F. or 613°R.

Correction factor = $0.0544\ \sqrt{(.7)(613)}$
= 1.12688
Corrected gas volume = (1.127)(800,000)
= 901,504 scf/d

To read the choke charts, an upstream and downstream pressure is needed, which is the casing pressure

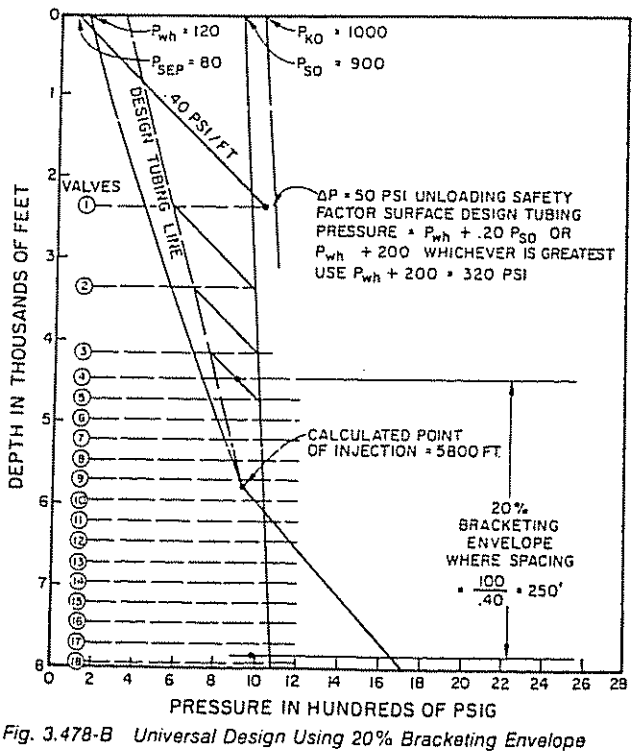


Fig. 3.478-B Universal Design Using 20% Bracketing Envelope

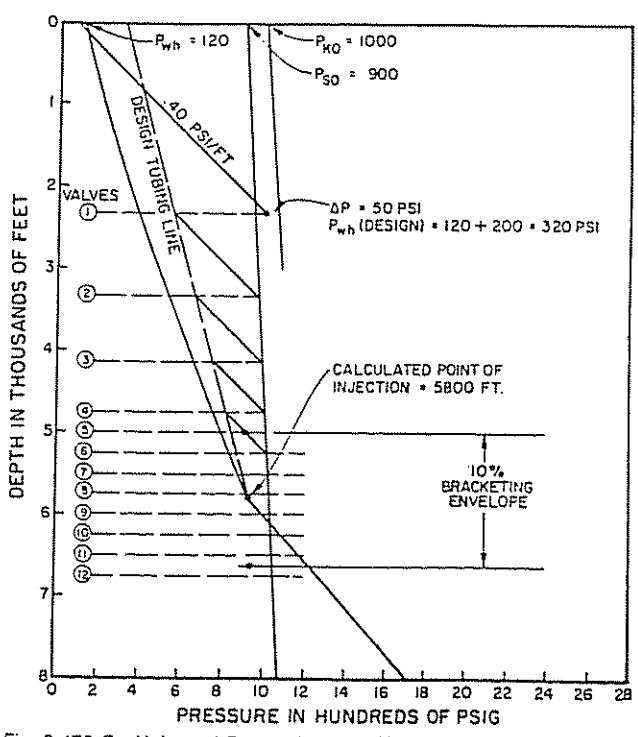


Fig. 3.478-C Universal Design Using 10% Bracketing Envelope

and tubing pressure at the valve, respectively. These values are found to be 1037 psi in the casing and 937 psi in the tubing at 5800 ft and can be found from Figure 3.477.

The nearest valve port size is slightly greater than a $^{16}/_{64}$", from Figure 3C.1, which corresponds to a ¼" diameter port. The next largest manufactured size of $^{5}/_{16}$" will be selected for this valve or a $^{9}/_{32}$" may be used if available. The correct amount of gas flow can still be controlled through a "throttling" type of gas lift valve or by lowering the differential across the operating gas lift valve. Also, use a $^{5}/_{16}$" ported valve for the lowermost Valve #8 and for Valve #6. Use of ¼" ports gives less flexibility in the valve if greater volumes are needed. Use gas volume charts to select the valve port if available.

(10) Select the port sizes for the upper unloading gas lift valves. It is possible to change port sizes or to maintain the same port sizes on all valves. Since a $^{5}/_{16}$" ported valve is relatively small, a $^{5}/_{16}$" port can be used on all gas lift valves. There is less chance for any valve setting errors if all valves are selected with the same port size. This example will be worked using all $^{5}/_{16}$" valves and also using $^{9}/_{32}$" valves for the bottom 3 and ¼" unloading valves.

(11) Determine the calibration of the valves in the shop so they will perform satisfactorily in the well. The first illustration uses a standard 1½" diameter valve with a $^{5}/_{16}$" port. For a bellows area of 0.77 in.^{2} this valve has the following values for R, (1 − R) and T.E.F.

$$R = \frac{A_p}{A_b} = 0.0996 \qquad (3.417)$$

$$1 - R = 0.9034$$

$$\text{T.E.F.} = \frac{R}{1 - R} = 0.11025 \qquad (3.418)$$

Prepare a table giving the valve number, depth, temperature, surface opening pressure, opening pressure at valve depth, and design tubing pressure. From this information it is possible to calculate the valve closing pressure at depth, the dome pressure at 60°F., and the test rack opening pressure.

These valves are charged with nitrogen at 60°F. All of this information is tabulated as noted in Tables 3.464 and 3.465. Table 3.464 shows one calibration of valves with port sizes all $^{5}/_{16}$". Table 3.465 shows ¼" unloading valves and $^{9}/_{32}$" operating valves. Since $^{9}/_{32}$" ports are not standard, it is likely that $^{5}/_{16}$" ports would be used on the bottom three valves and ¼" ported unloading valves.

The same general procedure would be used for 1" diameter valves. Many of these are used in the field because of space limitations.

TABLE 3.465
VALVE INFORMATION FOR PRESSURE OPERATED VALVES

Valve no.	Port size	Depth	Temp	P_{so}	P_{vo}	P_t tubing design	P_{vc} = P_d at Depth	P_d 60°	P_{tro} 60°F	Final calib.
1	¼	2350	127	900	955	570	930	810	865	865
2	¼	3350	135	900	979	676	960	825	881	880
3	¼	4150	141	900	998	762	983	832	889	890
4	¼	4750	145	900	1012	825	1000	843	900	900
5	¼	5250	149	900	1024	879	1015	845	903	905
6	$^{9}/_{32}$	5650	152	900	1033	921	1024	855	930	930
7	$^{9}/_{32}$	5800	153	900	1037	937	1029	860	936	935
8	$^{9}/_{32}$	5900	154	900	1039	948	1032	860	936	935

For ¼" Port, R = 0.0638, 1 − R = 0.9362
For $^{9}/_{32}$" Port, R = 0.08068, 1 − R = 0.9193
$P_d = P_{vc} = P_{vo}(1 - R) + P_t(R)$

An explanation for the determination of the values found in Tables 3.464 and 3.465 is given in the following summary.

Explanation of Columns in Summary Tabulation
(Table 3.464 and 3.465)

DESCRIPTION	EXPLANATION
Valve number	Self-explanatory
Port size	See step 9 design procedure
Depth	From Fig. 3.477
Temperature	From Fig. 3.477
P_{so}	Given data
P_{vo}	From Fig. 3.477
P_t (Tubing design)	From Fig. 3.477
$P_{vc} = P_d$ at depth	From Eqn. $P_d = P_{vo}(1 - R) + P_tR$
P_d 60°	From Fig. 3D.1 or Table 3D.1 in Appendix 3D.
P_{TRO} (60°F.)	From Eqn. $P_{TRO} = \frac{P_d}{1 - R}$
Final calibration	Self-explanatory

TABLE 3.464
VALVE INFORMATION FOR PRESSURE OPERATED VALUES

Valve no.	Port size	Depth	Temp	P_{so}	P_{vo}	P_t = Design tubing pressure	P_{vc} = P_d at Depth	P_{sc} P_t(Design)	P_d 60°F	P_{tro} 60°F	Final calib.
1	$^{5}/_{16}$	2350	127	900	955	570	917	862	800	886	885
2	$^{5}/_{16}$	3350	135	900	979	676	949	876	820	908	910
3	$^{5}/_{16}$	4150	141	900	998	762	975	881	830	919	920
4	$^{5}/_{16}$	4750	145	900	1012	825	993	884	840	930	930
5	$^{5}/_{16}$	5250	149	900	1024	879	1010	888	845	935	935
6	$^{5}/_{16}$	5650	152	900	1033	921	1022	890	855	946	945
7	$^{5}/_{16}$	5800	153	900	1037	937	1027	892	860	952	950
8	$^{5}/_{16}$	5900	154	900	1039	948	1027	892	860	952	950

R = .0996, 1 − R = .9004, T.E.F. = .1106

Another set of useful charts can be found in Appendix 3E for determining dome pressures (closing pressures) or opening pressures with varying tubing pressures. Also, additional charts are found in Appendix 3D for valve settlings at 80°F. and valves charged with natural gas.

3.4293 Design procedure for pressure operated valves—taking 10-20 psi drop in surface closing pressures between valves

If the universal spacing is not desired, then one recommended method of design is to take a drop in surface closing pressures between valves, even though some continuous flow valves are designed to have non-constant closing pressures.

To ensure that all upper unloading valves will close and remain closed in sequence as the well is unloaded to the point of injection, the surface closing pressure of each sequential valve down the hole should be lower than the valve immediately above. To understand this, it is necessary to understand the unloading process as shown in Figures 3.479. Figure 3.479A shows that gas is being injected through the top valve and the pressure conditions are such that gas flow is just beginning into the second valve. The process is called transferring from the top valve to the second valve. Figure 3.479B shows that transfer has taken place into the second valve and the flowing gradient is becoming lighter toward the point where transfer to the third valve will take place.

However, at this point notice that both valves one and two are still open. Both valves must be open, if both valves were closed at this time the unloading process would stop. Therefore, only the top valve should be closed by allowing the casing pressure to drop to the surface closing pressure of the top valve.

The surface closing pressure of the top valve should be set as high as possible and still allow the valve to be opened with kickoff casing pressure and transfer tubing pressure. If it is possible that the top valve could be the operating valve, then the surface closing pressure of that valve should not be set above the available casing operating pressure. Of course, the surface closing pressure of the second valve must be lower, or both valves close at once and the unloading process would stop. Experience has shown that a 10-20 psi downsetting of surface closing pressures for each valve gives good results. The larger the port size, the smaller the downsetting required.

The example problem of section 3.42421 is a contin-

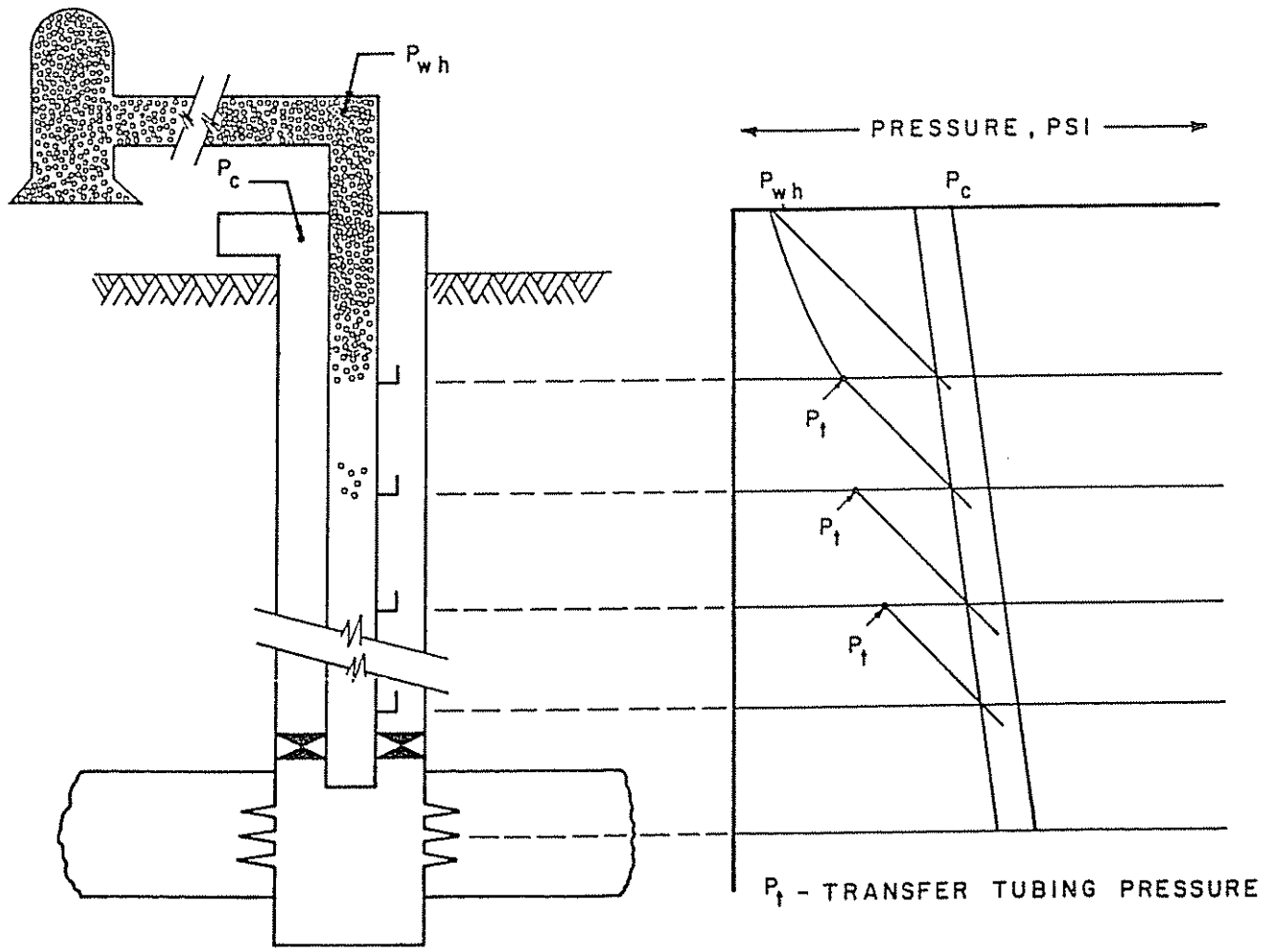


Fig. 3.479-A Unloading Process

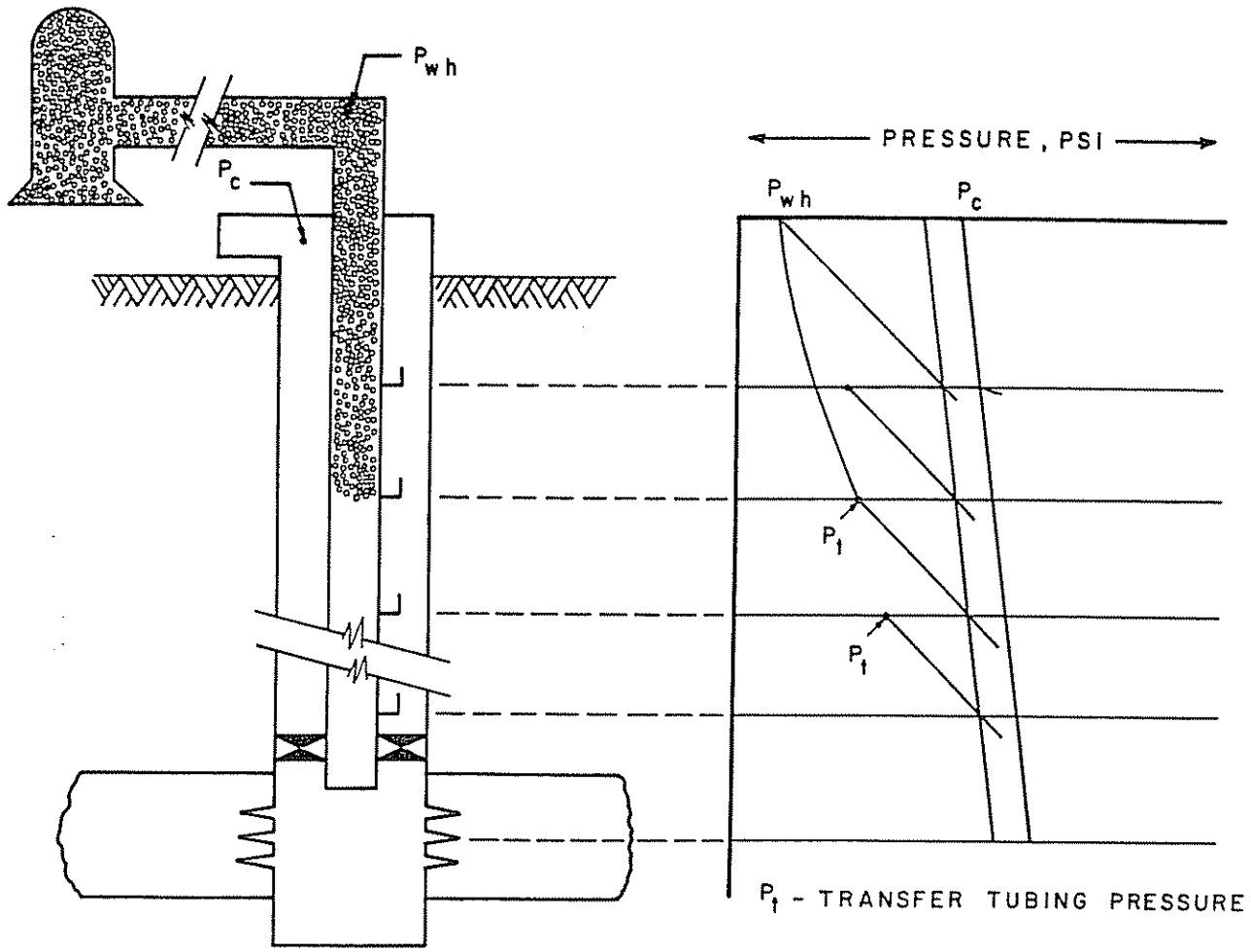


Fig. 3.479-B Transfer Process

uous flow design example using surface closing pressure as the setting criterion:

SPACING PROCEDURE: (Figure 3.479C)

The point of injection is located at approximately 5800 ft

(1) Plot the gas gradient line for kickoff and operating pressures.

(2) Construct the flowing tubing pressure gradient for the desired production rate and GLR as shown in Figure 3.479C.

(3) Construct the kill fluid gradient line of 0.4 psi/ft. This is done by drawing a straight line from zero to 400 psi at 1000 ft. (working line only)

(4) Extend the 0.4 psi/ft line from 80 psi @ 0 ft. until the tubing pressure is 50 psi less that of the kickoff casing pressure gradient line at depth. This locates the first valve at about 2350 ft.

(5) Select the transfer tubing pressure on or to the right of the 1000 BLPD gradient pressure at the first valve. Figure 3.479D shows possible transfer points.

(6) Set the surface close of the first valve at 950 psi (the surface operating pressure) and calculate the surface opening pressure based on the transfer tubing pressure of 420 psi (Fig. 3.479C). If the surface opening pressure is less than or equal to the available kickoff pressure, the selection of 950 psi for the surface closing pressure is fine. If not, the surface close pressure must be lowered until this criterion is satisfied.

Since no feed in from the formation is occuring when transfer from the first to the second valve will take place, the selection of the port size of the first valve can be based on other considerations than gas passage. For our example a ¼" port, 1½" OD valve was selected.

For a surface close of 950 psi:

$$P_d = P_{vc} = 1010 \text{ psi:} \qquad R = 0.0720$$

$$P_{vo} = \frac{1010 - 420\,(0.072)}{0.928} \qquad 1 - R = 0.9280$$

$$P_{vo} = 1056 \quad P_{so} = 996 \quad \text{OK!} \quad (P_{ko} = 1000)$$

(7) Space the second valve by extending a 0.4 psi/ft line from the transfer pressure of 420 psi at the first valve to the 950 psi surface pressure line. Move up the hole until the tubing pressure is 50 psi less than the casing pressure and place the second valve at 3750'. Set the surface close pressure of this valve 15 psi less than the first or 935 psi.

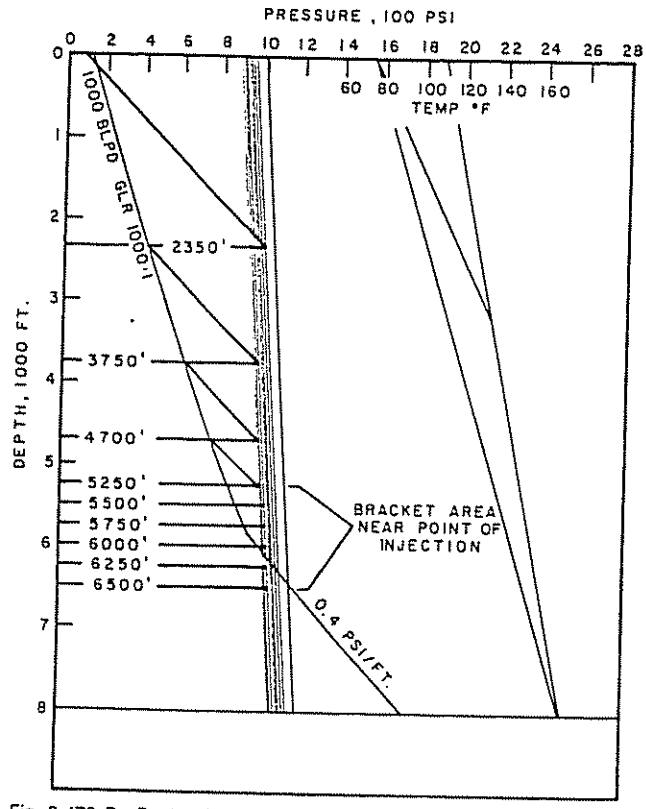


Fig. 3.479-C Design Procedure - Pressure Operated Valves

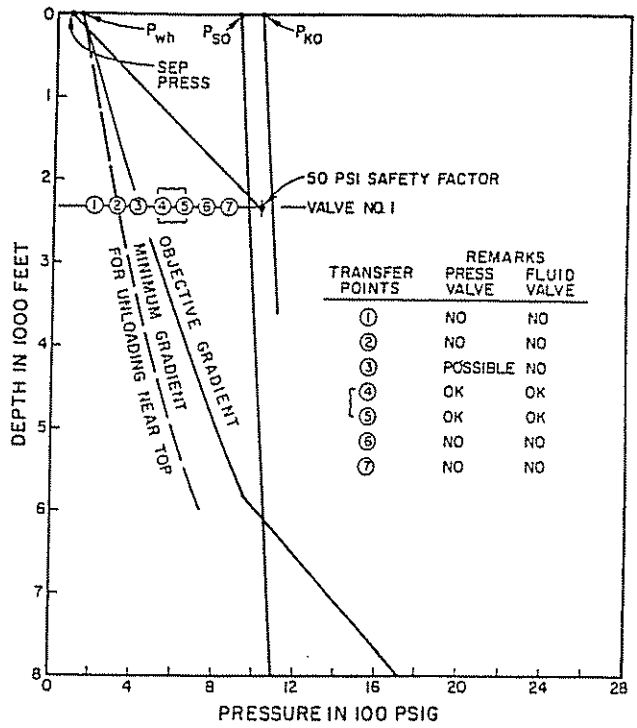


Fig. 3.479-D Selection of Transfer Points

TABLE 3.466
SUMMARY TABULATION—PRESSURE OPERATED VALVES
(15 psi drop in surface closing pressure)

Valve no.	Depth	T_v	TCF	Port size	R	1 – R	ΔP_c	P_t	P_tR	P_{sc}	P_d P_{vc}	P_{vo}	P_{so}	P_d at 60°F	P_{tro}
1	2350	118	.889	¼	.0720	.9280	52	420	30	950	1052	1101	1049	935	1008
2	3750	138	.856	¼	.0720	.9280	84	630	45	935	1019	1050	966	872	940
3	4700	145	.845	¼	.0720	.9280	105	780	56	920	1025	1044	939	866	933
4	5250	148	.841	¼	.0720	.9280	117			905	1022			860	926
5	5500	151	.836	¼	.0720	.9280	123			890	1013			847	913
6	5750	153	.833	¼	.0720	.9280	128			875	1003			836	900
7	6000	155	.830	¼	.0720	.9280	124			860	994			825	889
8	6250	156	.829	¼	.0720	.9280	140			845	985			817	880
9	6500	158	.826	¼	.0720	.9280	145			830	975			805	868

(8) Select the transfer tubing pressure on or to the right of the 1000 B/D gradient pressure at the second valve.

(9) Space the third valve by extending the 0.4 psi/ft line from the transfer pressure of 620 psi at the second valve to a line 15 psi less than the 950 psi casing pressure line. This is done because the casing pressure has been dropped to 15 psi to close the first valve. Move up the hole until the tubing pressure is 50 psi less than the casing pressure (now 935 psi) and place the third valve at 4700'. Set the surface close of this valve at 920 psi.

Continue this procedure until the desired bracketing envelop is reached. P'ace the valves every 250' and continue to drop the surface closing pressures 15 psi per valve. The completed design is noted in Figure 3.479-C.

(10) Prepare a summary tabulation showing the spacing, depths, settings, etc. of all valves (Refer to Table 3.466).

3.4294 Spacing and design procedure for fluid-operated valves (pressure-charged dome)—universal design

For this design procedure the spacing of the valves will be exactly the same as for the pressure-operated valves: hence, Figure 3.477 will be used. However, to

TABLE 3.467
VALVE INFORMATION FOR FLUID OPERATED VALVES

Valve no.	Depth	Temp	P_{so}	Tubing opening P_t design	P_{vc} Casing	Port	(Tubing) $P_d = P_{vc}$	P_d 60°F.	$P_{tro} = P_t$ at 60°F.	Adjusted P_{tro} (tubing press)
1	2350	127	900	570	955	1/4	595	520	555	555
2	3350	135	900	676	979	1/4	695	600	641	640
3	4150	141	900	762	998	1/4	777	655	700	700
4	4750	145	900	825	1012	1/4	837	707	755	755
5	5250	149	900	879	1024	1/4	888	745	796	795
6	5650	152	900	921	1033	9/32	930	780	849	850
7	5800	153	900	937	1037	9/32	945	790	859	860
8	5900	154	900	948	1039	9/32	955	800	870	870

insure the closing of upper valves, run 1/4" ported valves for unloading and size the last three valves according to the choke selections of 9/32". The tubing opening pressure for each valve is the design tubing pressure for each valve. In this case use pressure-charged fluid operated valves. For fluid valves, think of opening and closing pressures in terms of the tubing pressures. This creates no problem for operation but may confuse those accustomed to thinking in terms of having a closing pressure lower than the opening pressure.

All valves have the same surface casing injection pressure of 900 psi. As far as the casing is concerned, the operating pressure to open each valve is 900 psi, but the valve is principally opened by pressure in the tubing. Once the valve is open, the tubing pressure is then acting on the total bellows area. The valve has less pressure acting on the seat area; hence, the closing tubing pressure increases and the tubing closing pressure is higher than the tubing opening pressure.

Test rack opening pressures are given in terms of tubing pressure to open the valve, not casing pressure. Table 3.467 shows a summary and test rack settings of the pressure charged fluid operated valves.

Note that the P_{vc} in this case represents pressure in the tubing at which the valve closes and is a value higher than P_{vo} (pressure in the tubing at which the valve opens). For example, the operating valve #7 opens with 937 psi in the tubing and closes with 945 psi in the tubing.

Refer to Table 3.467 to find $P_d = P_{vc}$ at depth using the following equation:

$$P_d = P_{vc_t} = P_T(1 - R) + P_{vo}(R) = \text{closing tubing pressure.} \quad (3.419)$$

Note that P_T and P_{vo} have changed positions as compared to a pressure operated valve.

$$P_{TRO} = \text{pressure in tubing to open valve} = \frac{P_{vo_{(T)}}}{1 - R} \quad (3.420)$$

Fluid valves are commonly charged with spring loadings; this same example problem is worked for a spring loaded valve in Section 3.4295. Also, there are other methods for setting the gas lift valves. Section 3.4295 discusses the fluid spring loaded valve and gives a better design procedure. In this case the goal is to show how fluid valves can be set using a spacing procedure that is also good for pressure valves.

3.4295 Spacing and design procedure for combination pressure-closed fluid-opened valves

This valve is described in Section 3.42. It incorporates a spring differential section that can operate independently as a differential valve being controlled (opened) by sufficient pressure in the tubing. There are similar valves (called two valves in one) that can be similarly spaced. These valves have a pressure or spring charged section that operates like a standard bellows charged or spring charged gas lift valve. The differential section can open independently but will not pass gas unless the pressure or spring charged section is open, and vice versa. The upper pressure charged section assures the valve's closure even though the differential section may be open.

The same universal spacing procedure can be used and the valve selections are made as follows. Assume a bellows charged upper valve. Calibrate these valves with 5/16" ports the same as noted in Section 3.4292. The spring selections for the differential section of the valve will be the difference between the operating casing pressure and the design tubing pressure. For good practice, run 1/4" ported unloading valves and 9/32" valves for the last three valves which are near the operating depth. Both possibilities are shown in Tables 3.468 and 3.469. However, gas flow rate performance curves should be available to certify that sufficient gas volumes will pass with the desired differential.

3.4296 Design example for fluid-operated, spring-loaded gas lift valves

The techniques used in a continuous flow fluid valve design are derived from the gas passage characteristics of the spring-loaded fluid valve described in Section 3.42. The dome of the valve is not charged. Therefore, the bellows acts only as an element of area, while the spring supplies the closing force. Note that the port size must be small enough to create a pressure drop across the valve. This results in the entire bellows area sensing only tubing pressure when the valve is in the open position.

Figure 3.480A illustrates the flow rate response of the valve as a function of tubing pressure. If the tubing pressure is within the throttling range of the valve, the pressure drop across the port, coupled with the spring effect, causes the valve to respond proportionally to

TABLE 3.468
VALVE INFORMATION FOR COMBINATION FLUID OPENED-PRESSURE CLOSED VALVES

Valve no.	Port size	Depth	Temp	P_{so}	P_{vo}	Design tubing pressure	P_{vc} = P_d at Depth	P_{sc} P_t (Design)	P_d 60°F.	P_{tro} 60°F.	Final Calib.	$P_{vo} - P_t$ (Design) Spring Setting	Final Calib.
1	5/16	2350	127	900	955	570	920	865	800	886	885	385	385
2	5/16	3350	135	900	979	676	952	879	820	908	910	303	305
3	5/16	4150	141	900	998	762	978	884	830	919	920	236	235
4	5/16	4750	145	900	1012	825	996	887	840	930	930	187	190
5	5/16	5250	149	900	1024	879	1013	891	845	935	935	145	145
6	5/16	5650	152	900	1033	921	1025	893	855	946	945	112	110
7	5/16	5800	153	900	1037	937	1030	895	860	952	950	100	100
8	5/16	5900	154	900	1,039	948	1033	895	860	952	950	91	100

TABLE 3.469
VALVE INFORMATION FOR COMBINATION FLUID OPENED—PRESSURE CLOSED VALVE

Valve no.	Port size	Depth	Temp	P_{so}	P_{vo}	P_t Tubing design	P_{vc} = P_d at Depth	P_d 60°	P_{tro} 60°F.	Final calib.	$P_{vo} - P_t$ (Design) spring setting	Final Calib.
1	1/4	2350	127	900	955	570	930	810	865	865	385	385
2	1/4	3350	135	900	979	676	960	825	881	880	303	305
3	1/4	4150	141	900	998	762	983	832	889	890	236	235
4	1/4	4750	145	900	1012	825	1000	843	900	900	187	190
5	1/4	5250	149	900	1024	879	1015	845	903	905	145	145
6	9/32	5650	152	900	1033	921	1024	855	930	930	112	110
7	9/32	5800	153	900	1037	937	1029	860	936	935	100	100
8	9/32	5900	154	900	1039	948	1032	860	936	935	91	100

increases and decreases in tubing pressure. Gas flow tests indicate that in the interval between Points 1 and 2 of Figure 3.480, some gas flow occurs; but for all practical purposes, the quantity is negligible in relation to the overall capacity of the valve. As shown in the following design example, the valve closing pressures are chosen so the opening pressures will be greater than the design flow rate gradient. This ensures against valve interference when the tubing pressure falls in this range.

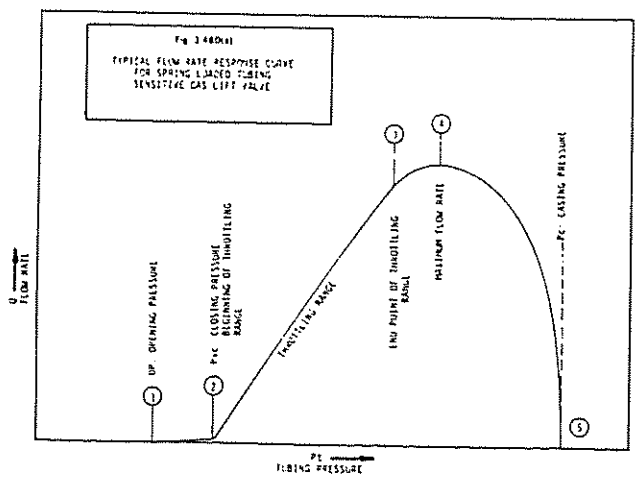


Fig. 3.480-A Typical Flow Rate Response Curve for Spring Loaded Tubing Sensitive Gas Lift Valve

The fluid valve acts as a variable orifice with the orifice size increasing with increasing tubing pressure and decreasing with decreasing tubing pressure. This proportional response maintains the established flowing tubing gradient. Since valve closure is independent of casing pressure, full operating pressure can be used at the point of injection regardless of the depth of that point. These are desireable operating characteristics for continuous flow applications.

Following is a design example for a continuous flow fluid valve installation:

Given:
Depth = 8000 ft (Mid Perf)
$\overline{P}_R$ = 1920 psi
q_L = 800 B/D (50% water)
Tubing size = 2 7/8 in. O.D.
2.5 in. I.D.
P_{wh} = 120 psi
J = 4 B/D psi
Gas Gravity = 0.7
BHT = 170 °F @ 8000 ft
Surface flowing temp = 110 °F.
GOR = 300 scf/B
P_{ko} = 1000 psi
P_{so} = 900 psi
Well kill fluid gradient = .468 psi/ft
Well loading = all the way to top
Well unloading = to be unloaded to 80 psi separator. P_{wh} = 120 psi under flowing conditions

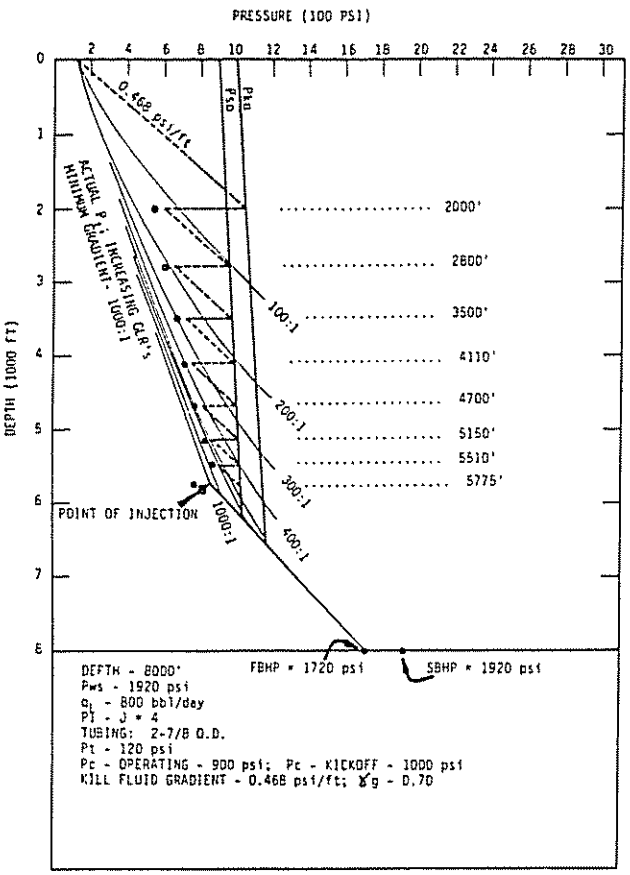


Fig. 3.480-B Continuous Flow. Graphical Procedure for Spacing Fluid-Operated Valves

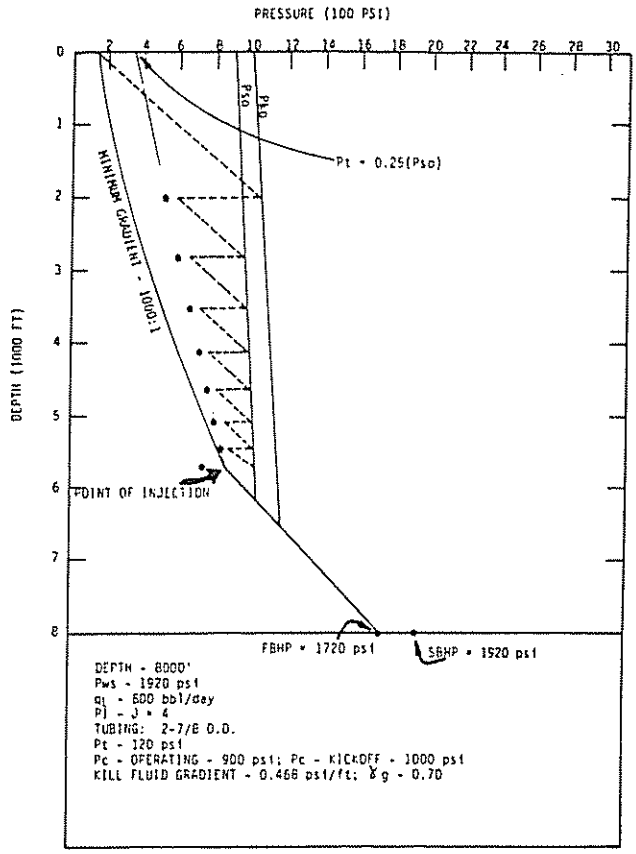


Fig. 3.480-C Spacing Procedure

Spacing procedure: (Refer to Figure 3.480 B)

The point of injection is located at approximately 5775 ft

(1) Plot the gas gradient lines for kickoff and operating pressures.

(2) Construct the flowing tubing pressure gradient for several GLR's at the desired producing rate.

(3) Construct the kill fluid gradient line of 0.468 psi/ft. Do this by drawing a straight line from zero to 468 psi at 1000 ft.

(4) Extend the 0.468 psi/ft line from 120 psi @ 0 ft until it intersects with 1000 psi gas gradient line. This locates the first valve at 2000 ft.

(5) Select the valve closing pressure for the first valve. This Pvc should be as high as possible yet allow for realistic spacing to the next valve. Selecting the valve closing pressure as high as possible serves two purposes. First, if the actual back pressure is greater than the anticipated back pressure, the possibility of valve interference from an upper valve is minimized. Second, increasing the Pvc reduces the injection gas required to transfer to the next lower valve. Experience in an area and the known accuracy of the design information dictate the degree of safety factor used in the selection of Pvc. Except for the operating valve at the point of injection, Pvc should be high enough that the opening pressure of the valve will be greater than or equal to the minimum flowing gradient pressure at that location. Use the following technique for the selection of Pvc for all but the operating valve: add 25% of the surface operating pressure to the design flowing tubing pressure. Extend a line from this point down to the point of injection. Use the intersection of this line with the horizontal line at each valve location as the valve closing pressure, Pvc. (See Fig. 3.480C.)

(6) Draw a horizontal line from the point located in Step 4 to the point located in Step 5. Locate the point 50 psi greater than Pvc. This is the transfer point used to space the next lower valve. Using a point 50 psi greater than Pvc as the transfer point ensures the upper valve will still be passing gas when the next lower valve is uncovered. (See Fig. 3.480-D)

(7) Draw a line parallel to the 0.468 psi/ft line from the transfer point of Step 6 until it intersects the 900 psi surface operating pressure line. This locates Valve 2 at 2800 ft.

(8) Pvc and the transfer point are determined in the manner previously discussed.

(9) Continue this procedure until a valve is located at or near the point of injection.

(10) Determine the closing pressure for the operating valve. To ensure adequate gas passage and throttling of the operating valve, the Pvc of the

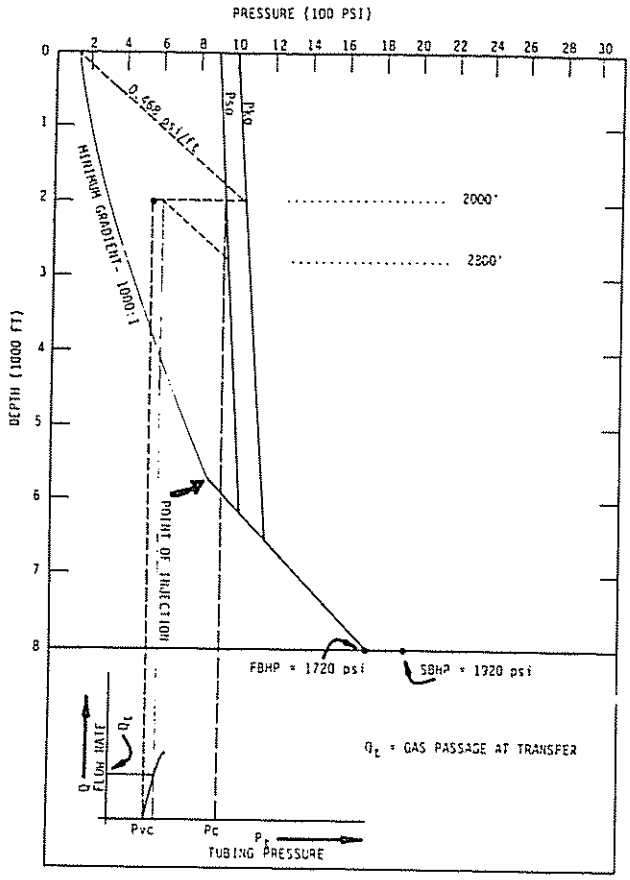


Fig. 3.480-D Illustration of Gas Passage at Transfer

operating valve should be set approximately 80 psi less than the minimum flowing gradient pressure. This will vary with port size and gas requirements.

(11) Determine port sizes of the valves. Do this for each valve as it is spaced. Use the smallest port size required for transfer from each location. Table 3.470 illustrates these selections.

3.4297 Additional spacing procedures

3.4297 Introduction

Additional methods of spacing are available. In particular, a drop in surface opening pressure between valves is commonly used. This eliminates valve interference completely, assuring single point injection. For single installations where ample injection pressure is available, this procedure is common.

Additional safety factors in spacing are sometimes used. For example, 0.50 psi/ft is commonly used for the value of G_s. For unloading heavy brine, this may be increased as high as 0.60 psi/ft, where G_s represents the gradient of the kill fluid to be unloaded. Also, the gas column pressure may be neglected, particularly if an analytical procedure is being used.

3.42972 Common procedure for analytical spacing of pressure-operated gas lift valves

The same example will be worked as given in Section 3.4291. If sufficient data is available, the point of gas injection is determined as outlined previously and is found to be at 5800 feet. Equations used for the analytical determination of valve spacing are as follows:

$$D_{v_1} = \frac{P_{ko} - P_{wh}}{G_s} \tag{3.421}$$

$$D_{v_1, v_2 \ldots} = D_{v_1, v_2} + \frac{P_{so_1, 2 \ldots} - P_{wh} - D_{v_1, v_2 \ldots}(G_u)}{G_s} \tag{3.422}$$

D_{v_1} = Depth first valve. This top valve may be placed at this depth or at the fluid level, whichever is deeper.

$D_{v_2, v_3 \ldots}$ = Depth Valve 2, Valve 3, etc.

$P_{so_1}, P_{so_2} \ldots$ = Surface opening pressure of Valve 1, 2, etc.

G_u = Unloading gradient (psi/ft)

P_{wh} = Wellhead pressure (psi)

For a balanced valve installation, this pressure must be dropped 25 psi per valve for the given installation; 25 psi between valves may also be used for unbalanced valves. Where high injection pressure is available, use 50 psi or more between valves. Some clarification on the spacing gradient to use is probably in order. The unloading gradients of Figures 3F.1 and 3F.2 (Appendix) are based on experience factors and minimum flowing gradients possible by continuous flow. Also use the average gradient between the point of injection and the wellhead pressure obtained from Figure 3.476. This is 0.14 psi/ft, which checks very closely with the unloading gradient of 0.13 psi/ft

TABLE 3.470
SUMMARY TABULATION—FLUID VALVE CONTINUOUS FLOW DESIGN

Valve no.	Depth (ft.)	Casing Pressure at Depth (psig)	Pvc Valve Closing Pressure (psig)	GLR at Transfer Point	Tubing Pressure at Transfer Point (psig)	Required Gas Input to Transfer (MCF/D)	Port Size (64th)	Orifice Capacity (MCF/D)	70% of Orifice Capacity (MCF/D)	1-R	OP (psig)	P_{tro} (psig)
1	2000	1050/945	550	150	600	120	8	320	224	0.9614	530	572
2	2800	970	610	190	660	152	8	300	210	0.9614	596	635
3	3500	980	675	250	725	200	10	470	329	0.9373	654	720
4	4110	1000	720	350	770	280	10	460	322	0.9373	701	768
5	4700	1010	780	420	830	336	12	630	441	0.9053	756	862
6	5150	1020	830	480	880	384	16	750	525	0.8263	790	1004
7	5510	1030	870	600	920	480	16	720	504	0.8263	836	1052
8	5775	1035	780	1000	860	800	16	850	-	0.8263	727	944

obtained from Figure 3F.2 (Appendix). I recommend using 0.14 psi/ft.

Therefore, start with the kick-off pressure of 1000 psi at the surface and take a 25 psi drop per valve. Neglect the gas column pressure and use an unloading gradient of 0.40 psi/ft. The kill fluid gradient was previously given as 0.40 psi/ft. Work the same example as given in Section 3.4292.

Valve spacing calculations

$$D_{v_1} = \frac{P_{ko} - P_{wh}}{G_s} = \frac{1000 - 120}{.40} = 2200 \text{ ft}$$

Note that this compares to 2350 ft for the previous graphical spacings. The difference is due to the gas column weight.

$$D_{v_2} = D_{v_1} + \frac{P_{so_1} - P_{wh} - G_u(D_{v_1})}{G_s}$$

$$D_{v_2} = 2200 + \frac{1000 - 120 - (.14)(2200)}{.40} = 2200 + 1430 = 3630 \text{ ft}$$

Note that we can use the opening pressure of Valve #1 to space Valve #2. This is sound design procedure. Available in the casing is the pressure of Valve #1 until reaching Valve #2; that is, the pressure in the casing does not drop to the pressure of Valve #2 until reaching Valve #2. However, common field practice is to use P_{so_2} in spacing Valve #2.

$$D_{v_3} = D_{v_2} + \frac{P_{so_2} - P_{wh} - G_u(D_{v_2})}{G_s}$$

$$= 3630 + \frac{975 - 120 - .14(3630)}{.40}$$

$$= 3630 + 867 = 4497 \text{ ft} \qquad (3.423)$$

$$D_{v_4} = D_{v_3} + \frac{P_{so_3} + P_{wh} - G_u(D_{v_3})}{G_s}$$

$$= 4497 + \frac{950 - 120 - .14(4497)}{.40}$$

$$= 4497 + 501 = 4498 \text{ ft} \qquad (3.424)$$

$$D_{v_5} = D_{v_4} + \frac{P_{so_4} - P_{wh} - G_u(D_{v_4})}{G_s} \qquad (3.425)$$

$$= 4998 + \frac{925 - 120 - .14(4998)}{.40}$$

$$= 4998 + 263 = 5261 \text{ ft}$$

$$D_{v_6} = D_{v_5} + \frac{P_{so_5} - P_{wh} - G_u(D_{v_5})}{G_s} \qquad (3.426)$$

$$= 5261 + \frac{900 - 120 - .14(5261)}{.40}$$

$$= 5261 + 108 = 5370 \text{ ft}$$

The distance between Valve #5 and Valve #6 is only 108 feet, and the calculation for distance between Valve #6 and #7 is only 8 feet. There is no more spacing and strictly following this procedure, no valves can be placed below 5380 feet, which is 420 feet from the determined point of injection. Therefore, beginning with Valve #5, space valves 200 feet apart (common practice is to space valves 200 feet apart after reaching a spacing of less than 200 feet) until reaching one valve below the point of injection. This places valves at 2200, 3630, 4497, 4998, 5261, 5461, 5661, 5861 and 6061. Then readjust the last two valves to depths of 5800 feet and 5950 feet. If desired, gas column pressures can be accounted for in this analytical procedure and will permit reaching the point of injection with no trouble. Use port sizes of 5/16" on the bottom three valves and ¼" on the upper unloading valves.

Note Table 3.471 for a summary tabulation for this design. In this example, good use is made of the additional kick-off pressure of 1000 psi to start valve settings and almost reached bottom with the 900 psi operating pressure. If the kick-off and operating pressure are the same, over 100 psi pressure would be lost, resulting in a higher injection point and a lower production rate.

For this method of spacing there is a drop in surface opening pressure between valves. The spacing is based on the average operating flowing gradient in the tubing string while the previous procedure used a design tubing pressure line starting with P_{wh} + 200 psi = 320 psi at the surface and drawn to the point of injection. If valve spacing is based on the operating flowing gradient line, a drop in pressure is necessary. The drop in casing pressure keeps the valves closed. Note that the final shop calibrations do not necessarily reflect this 25 psi drop between the surface opening pressures of the valves.

TABLE 3.471
SUMMARY TABULATION FOR PRESSURE OPERATED VALVES—ANALYTICAL SPACING
(25 PSI DROP IN SURFACE OPERATING PRESSURE BETWEEN VALVES)

Valve no.	Port size	Depth	Temp	P_{so}	P_{vo}	P_t	Adjusted P_t's*	$P_d = P_{vc}$	P_d 60°F.	P_{tro} 60°F.	Final calib. Adj. P_{tro}
1	¼	2200	126	1000	1058	428	428	1018	885	945	945
2	¼	3630	137	975	1068	628	628	1040	890	951	950
3	¼	4497	143	950	1062	750	750	1042	885	945	945
4	¼	4998	147	925	1046	820	820	1032	870	929	930
5	¼	5261	149	900	1024	857	857	1013	850	908	910
6	¼	5461	151	875	1000	884	884	993	827	883	885
7	5/16	5661	152	850	977	912	877	967	810	897	895
8	5/16	5800	153	825	951	932	851	941	785	869	870
9	5/16	5950	154	800	925	953	825	915	762	843	845

*Tubing Pressures are re-adjusted so that valve opens with a tubing pressure 100 psi less than casing pressure.

3.42973 Graphical spacing of pressure-operated valves—(25 psi drop in pressure between valves)

The same problem will now be worked graphically taking a 25 psi drop in surface opening pressure between valves.

Refer to Figure 3.481. The point of gas injection is determined in the same manner described earlier and is found to be at 5800 feet.

The following procedure is used to complete the design.

(1) Locate the first valve by drawing in the kill fluid gradient line of .40 psi/ft until intersecting the 1000 psi kick-off pressure line at 2350'.
(2) Starting from (1000 − 25) = 975 psi at the surface; construct the next valve operating pressure line including gas column pressure.
(3) Extend a line horizontally to the left from the depth of Valve #1 at 2350' until intersecting the operating flowing gradient line. From this point draw a line parallel to the previously-drawn kill fluid gradient line of 0.40 psi/ft until intersecting the 975 psi gas line. This locates Valve #2 at 3900 feet.
(4) Repeat this procedure until reaching the point of gas injection by continuing to take the 25 psi drop between valves. This locates valves at 2350, 3900, 4950, 5500 and 5800 feet.
(5) Place one valve below the point of injection at 6000 ft.
(6) Construct the temp gradient line as previously explained and note the temperature opposite each valve.
(7) Determine the gas volume necessary and port size of the operating valve to be 5/16". Use the bottom three valves with 5/16" ports and the upper four valves with 1/4" ports.
(8) Prepare a summary tabulation and determine the shop setting of the valves (Table 3.472).

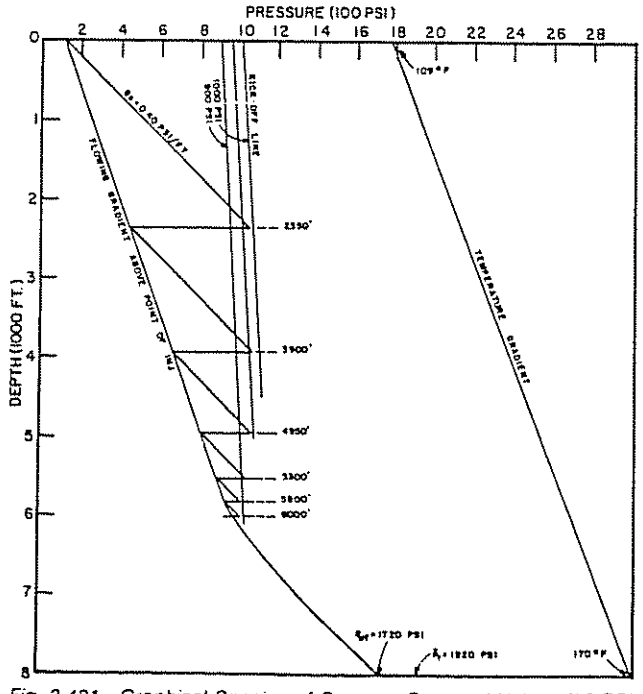


Fig. 3.481 Graphical Spacing of Pressure Operated Valves (25 PSI Drop in Surface Pressure Between Valves)

For the graphical spacing of valves whereby a pressure drop of 25 psi per valve is taken, the point of injection is reached with five valves. This also means there must be 1000 psi available to unload the well. If the maximum pressure available was the operating pressure of 900 psi, operating pressure would be lost moving down the well from one valve to the next. However, by unloading with 1000 psi, the point of injection can be reached with a surface operating pressure of 900 psi. This avoids sacrificing any pressure by taking a drop in pressure between valves. Circumstances dictate whether or not this can be done with a particular installation. Also, a change in the dropping of pressures between test rack opening pressures occurs when seat sizes are changed. This is normal and is to be expected.

3.4298 Discussion on spacing

Numerous procedures can be found for spacing gas lift valves; most of them will work. Probably, the greatest fallacy is the failure to space valves close enough near the point of gas injection. Some companies have set up universal spacing procedures for all wells, that is, the same distance between valves. This type procedure can restrict production severely in high productivity wells. For wells that have PI's above 5, and especially above 10, the uncovering of another valve some 100 feet lower may mean another 150-300 B/D of production.

There is certainly no justification for not spacing valves sufficiently close in offshore wells. These wells are very expensive to complete and a work-over may be almost prohibitive depending on location. It is good practice to run an extra two to three valves rather than take a chance on too few valves.

To illustrate the various valve settings for unbalanced pressure operated valves, Brown worked the following problem:[22]

Given: (Fig. 3.482):
Depth = 8000 ft
$\overline{P}_R$ = 2900 psi
q_L = 700 BD (95% water)
Tubing size = 2⅞" OD (1.995 in. ID)
P_{wh} = 100 psi
J = 7 B/D/psi
Bottom hole temp = 210°F.
Surface flowing temp = 150°F.
Surface operating pressure available = 900 psi
Kick-off pressure = 950 psi
Well "kill" fluid gradient = 0.50 psi/ft
Well loaded all the way to top
First valve only unloaded against zero surface pressure

The point of gas injection was 3820 feet, as noted in Fig. 3.482, and the spacing of the valves is noted in this figure. Tables 3.473, 3.474, and 3.475 show three different design possibilities and port sizes.

In the example of Table 3.473, the surface opening

TABLE 3.472
SUMMARY TABULATION GRAPHICAL SPACING OF PRESSURE OPERATED VALVES
(25 PSI DROP IN SURFACE OPERATING PRESSURE BETWEEN VALVES)

Valve no.	Port size	Depth	Temp	P_{so}	P_{vo}	P_t	Adjusted P_t	$P_d = P_{vc}$ (Depth)	P_d 60°F.	P_{tro} 60°F.	Adj. P_{tro}
1	¼	2350	127	1000	1058	451	451	1019	885	945	945
2	¼	3900	139	975	1075	669	669	1049	890	951	950
3	¼	4950	147	950	1074	817	817	1058	885	945	945
4	¼	5500	151	925	1058	895	895	1048	875	935	935
5	5/16	5800	153	900	1036	937	936	1029	857	949	950
6	5/16	6000	155	875	1013	965	913	1006	835	924	925

pressure was dropped 10 psi per valve for ¼" ports starting from 50 psi less than the available operating pressure. Note the following points:

(1) The drops in surface opening pressure from valve to valve down the well were 16, 13, 11, 13, and 12 psi under operating conditions. This added safety factor is the result of setting the valves on the basis of the design tubing pressure instead of the operating tubing pressure. Therefore, additional safety can be incorporated by shifting the design tubing pressure nearer to the casing pressure. However, additional valves will be required.

(2) Note that if operating from Valve 6, the required surface opening pressure (795 psi) is lower than the surface closing pressure (802.8 psi) of Valve 5. Therefore, the closing of upper valves is assured. However, this condition—not a necessity for variable orifice valves which do not have a constant closing pressure—only occurs when lifting from Valve 5 or (6). If lifting from valve 4 which is 820 psi to open, then Valves 2 and 3 could conceivably be open since they have closing pressures of 819.6 psi and 815.5 psi, respectively. The variable orifice feature allows these valves to close at or near their opening pressures of 844 and 831 psi. This assures single point injection.

To illustrate some very important points, this problem has been worked for the following conditions:

(1) ⅜" port, dropping the surface opening pressure by 10 psi per valve (Table 3.474)
(2) ⅜" port, holding the surface opening pressure constant (Table 3.475)
(3) ¼" port, holding the surface closing pressure constant (Table 3.476)

The valve number, depth, temperature at the valve, design tubing pressure, and operating tubing pressure are the same as the previous example in discussing these three sets of conditions. The summary tabulations, therefore, will not include these five columns.

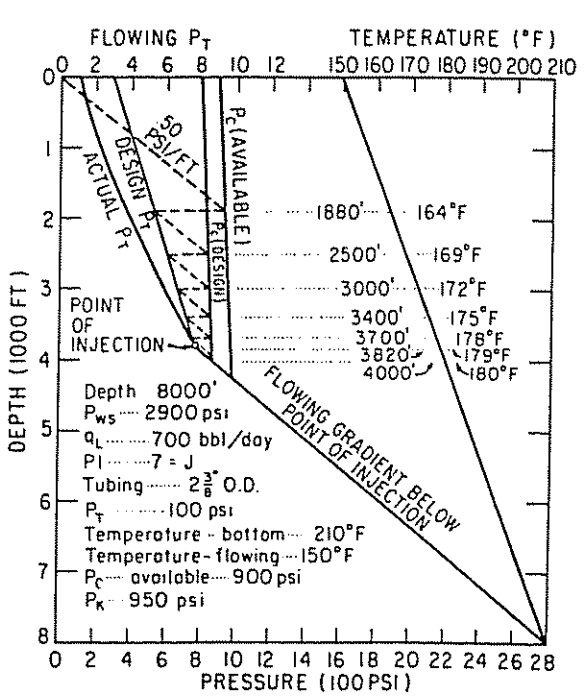


Fig. 3.482 Continuous Flow—Graphical Procedure for Spacing Unbalanced Gas Lift Valves

EXAMPLE PROBLEM #1—⅜" PORT; SURFACE OPENING PRESSURE DROP OF 10 PSI PER VALVE

The procedure is the same as the preceding example except that a ⅜" port is used instead of a ¼" port. Table 3.474 summarizes the results.

Note drops in surface opening pressures of 12, 20, 23, 16, 9 and 7 psi. Although greater differences generally occur with the ⅜" ports than for the ¼ in. port, the surface closing pressures are always less because of the greater port area. For example, the 806 psig opening pressure of Valve 5 is the same or greater than the closing pressure of all other valves. Therefore, the variable orifice feature of nonconstant closing is needed. With the variable orifice feature incorporated, the ⅜" port offers more safety in keeping the upper valves closed than does the ¼" port. Therefore, a ⅜" ported intermittent valve would give difficulty in continuous flow operations. If Valve 4 were the operating valve, Valves 1, 2, and 3 would probably never close. If they should close by some chance, the installation would function well since the opening pressures of Valves 1, 2, and 3 are all greater than the 822 psig opening pressure of Valve 4. Although it would not be impossible to continuous flow through Valve 4, it might take some careful surface regulation. If the well could be unloaded by intermittent cycles, perhaps Valve 4 could be reached with Valves 1, 2, and 3 closed. A true continuous flow valve will not be confronted with this problem since it does not have to drop to dome pressure to close.

EXAMPLE PROBLEM #2—⅜" PORT: SURFACE OPENING PRESSURE CONSTANT

Table 3.475 results from a design surface opening pressure held constant at 800 psi.

TABLE 3.473
SUMMARY TABULATION—SURFACE OPENING PRESSURE DROP OF 10 PSI PER VALVE (¼ IN. PORTS)

Valve no.	Depth (ft)	T_v (°F)	Design P_t (psig)	Operating P_t (psig)	Design P_{so} (psig)	Design P_{vo} (psig)	P_d at T_v P_{vc} (psig)	P_{sc} (psig)	P_d at 80°F (psig)	P_{tro} at 80°F (psig)	Operating P_{vo} (psig)	Operating P_{so} (psig)
1	1880	164	540	380	850	887	864.5	827.5	710	958	898	860
2	2500	169	615	490	840	889	868.6	819.6	712	760	895	844
3	3000	172	680	595	830	888	873.5	815.5	715	764	892	831
4	3400	175	725	680	820	885	876.3	811.3	715	764	890	820
5	3700	178	765	750	810	880	872.8	802.8	708	756	881	807
6	3820	179	775	775	800	873	866.5	793.5	704	752	873	800
7	4000	180	800	800	790	864	859.0	786.0	695	743	864	790

TABLE 3.474
SUMMARY TABULATION—⅜ IN. PORT; SURFACE OPENING PRESSURE DROP OF 10 PSI PER VALVE (R = 0.1434)

Valve no.	Design P_{so} (psig)	Design P_{vo} (psig)	P_d at T_v P_{vc} (psig)	P_{sc} (psig)	P_d at 80°F (psig)	P_{tro} at 80°F (psig)	Operating P_{vo} (psig)	Operating P_{so} (psig)
1	850	887	833	798	680	795	915	877
2	840	889	855	806	700	825	915	865
3	830	888	860	802	705	830	905	845
4	820	885	860	795	700	825	890	822
5	810	880	860	790	695	820	880	806
6	800	873	860	787	693	817	873	797
7	790	864	855	775	685	800	864	790

TABLE 3.475
SUMMARY TABULATION—⅜ IN. PORT; SURFACE OPENING PRESSURE CONSTANT

Valve no.	Design P_{so} (psig)	Design P_{vo} (psig)	P_{vc} (psig)	P_{sc} (psig)	P_d at 80°F (psig)	P_{tro} at 80°F (psig)	Operating P_{vo} (psig)	Operating P_{so} (psig)
1	800	836	793.5	758	660	770	865	828
2	800	848	813.4	766	662	773	865	816
3	800	857	832.5	775	680	795	872	814
4	800	865	845.0	780	685	800	875	809
5	800	870	854.5	784	695	812	873	801
6	800	873	858.0	785	695	812	872	797
7	800	876	864.0	790	695	812	872	797

The surface opening pressures of all valves can be held constant when based on design tubing pressure, but surface opening pressure drop will occur under operating tubing pressures. Drops in surface opening pressures of 12, 2, 5, 8, 4 and 0 psi are noted totaling 31 psi. However, if each valve had a constant closing pressure, all the closing pressures would be less than the 797 psig surface opening pressure of the bottom valve. Therefore, this installation would work only with valves that do not have to drop to dome pressure to close. Even so, the drop of 10 psi per valve should be taken to ensure good operations.

EXAMPLE PROBLEM #3—¼ IN. PORT; SURFACE CLOSING PRESSURE CONSTANT

The following tabulation results from the design surface closing pressure of each valve held constant at 800 psi.

TABLE 3.476
SUMMARY TABULATION—¼" PORT; SURFACE CLOSING PRESSURE CONSTANT

Valve no.	P_{sc} (psig)	P_{vc} (psig)	P_d at 80°F. (psig)	P_{tro} at 80°F. (psig)	Operating P_{vo} (psig)	Operating P_{so} (psig)
1	800	836	698	746	867	830.0
2	800	848	700	747	872	823.0
3	800	857	705	753	875	816.5
4	800	865	707	755	878	812.0
5	800	870	707	755	879	807.0
6	800	873	707	755	880	805.0
7	800	876	705	753	880	803.0

In this example the surface closing pressures were set constant. This would not be true for variable orifice

valves but would be true for a standard ¼" port intermittent valve. The example points out several factors: if these valves must drop to the dome pressure to close, conceivably if operating from Valve 6 at 3820 feet, all the other valves in the string are open. Note that the 805 psig surface opening pressure of Valve 6 exceeds the 800 psig surface closing pressure on all other valves. Therefore, if none of the other valves are closed and the system is operating on Valve 6, we would also pass gas through all other valves. If all the other valves are closed, it is possible to have one point injection at Valve 6 without interference from other valves. Note the drop in surface opening pressure from one valve to the next; these pressure drops are quite close and represent only a 25 psi drop from top to bottom. Any fluctuation in line pressure could cause valve interference. Note the test rack opening pressures at 80°F show an increase in pressure with the bottom four valves remaining essentially constant at 755 psi.

Again, nothing is wrong with this installation if it uses a variable orifice valve that does not have to drop to dome pressure to close. However a drop in surface pressures should be taken for this type of installation when high system pressure is available.

3.4299 Continuous flow design for differential valves

3.42991 Introduction

Although differential valves such as the spring-loaded are seldom used, their spacing principle is used with other valve types. To understand the spacing for these valves, recall from Section 3.42 that this valve operates on the difference between tubing pressure (fluid load) and casing pressure (injection pressure). Normal spring settings in the past were on the order of 150 psi, but might vary from 50-500 psi.

When running these valves with a packer, make provisions for U-tubing the "kill" fluid out of the annular space. If this is not done, all the valves may lock closed and manual manipulations will be necessary to unload the well. A sliding side door, pressure-operated valve, or reverse check on bottom will suffice. Once the well is unloaded, close the sliding side door by wire line tools.

To explain why this is necessary, assume that 800 psi is available for operation and that the well is standing full of "kill" fluid of 0.50 psi/ft. If the first valve is at 1880 feet (as located graphically from 900 psi kick-off pressure) and there is 800 psi on the surface, 800 + 0.50 (1880) = 1740 psi in the casing at the first valve. The tubing pressure at this point with no back pressure is 940 psi or 1040 psi with 100 psi back pressure. With a differential valve with a 150 psi spring, it cannot open until the tubing pressure is within 150 psi of the casing pressure. Therefore, it would take 1740 − 50 = 1590 psi in the tubing to open the valve allowing gas entry. The tubing pressure would need to be boosted or casing pressure bled off. Although this well can eventually be unloaded, communication between the tubing and casing should be provided in another manner to facilitate easier unloading.

Differential valves are normally used for continuous flow operations only, and the following design procedure should be followed:

3.42992 Design procedure for differential valves (continuous flow) (refer to Fig. 3.477)

(1) Determine the point of gas injection previously outlined. Use a design injection pressure of 100 psi less than available operation pressure.

(2) Deduct the spring setting pressure from the design injection pressure and extend both this pressure and the pressure of (1) down the well.

(3) Draw in the "kill" fluid gradient line from the well head tubing pressure or from zero depth depending upon how the well is to be unloaded. Extend this line until it intersects the kick-off pressure less 50 psi line and/or to the available operating pressure line. This intersection locates Valve 1.

(4) Draw a horizontal line back to the left until it intersects the line constructed in (2).

(5) From this line draw a line parallel to the "kill" fluid gradient line until it intersects the design injection pressure line established in (1). This intersection locates Valve 2.

(6) Continue this spacing procedure until the point of injection is reached and place at least one valve below this point.

(7) Since these valves are spring loaded, further valve selection detail is unnecessary, other than deciding on choke sizes for gas entry. As a precaution, keep ports small in differential valves (use less than $^{10}/_{64}$" where possible). This makes operation much smoother. Exact choke sizes can be selected from Figures 3C.1 through 3C.26 (Appendix).

3.42993 Design example for differential valves (continuous flow)

The same continuous flow problem will be worked for differential valves with all information noted in Section 3.4297 (Refer to Fig. 3.483).

(1) The point of gas injection is at 3820 ft and the design injection pressure is 800 psi.

(2) Use a spring differential of 150 psi which, when deducted from 800 psi, leaves 650 psi to be used for spacing purposes. The two pressures of 800 psi and 650 psi are extended down the well (Refer to Fig. 3.483).

(3) The "kill" fluid gradient of 0.50 psi/ft is drawn from zero tubing pressure until it intersects the 900 psi line. This locates Valve 1 at 1880 feet.

(4) Draw a horizontal line from this point back to the left until it intersects the 650 psi gas line.

(5) Draw a line parallel to the 0.50 psi/ft line until it intersects the 800 psi line, locating valve (2) at 2190 feet.

(6) Continue spacing in this manner between the 800 and 650 psi line locating valves at 2500, 2800, 3100, 3400, 3700, and 4000 feet.

(7) Select port sizes from charts prepared in Appendix 3C.

Finally, an installation with a packer is recommended utilizing a sliding side door above the packer. This

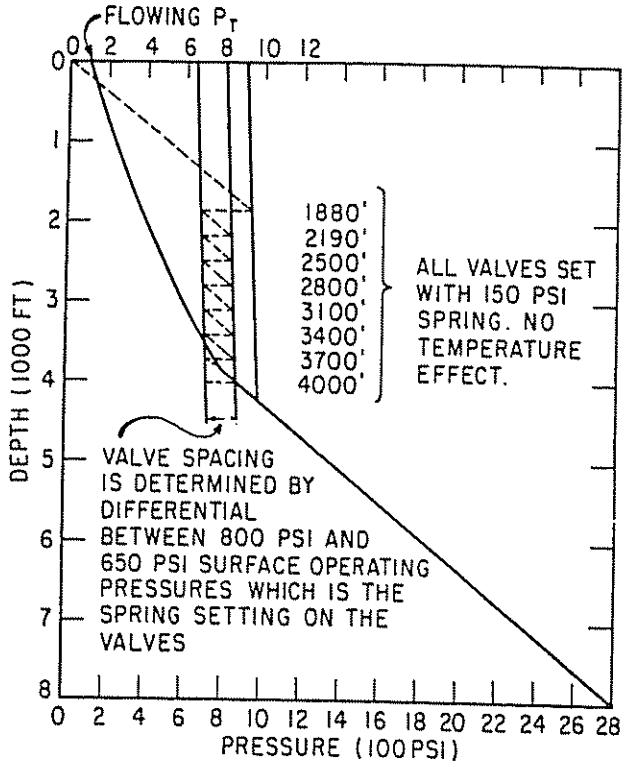


Fig. 3.483 Continuous Flow—Graphical Procedure for Spacing Differential Valves

sliding side door, left open until the well is unloaded, is then closed.

3.42994 Analytical procedure for spacing differential valves

The analytical procedure for spacing differential valves is simple and will be illustrated by working the problem in Section 3.4297.

Gas column weight is neglected and all but the topmost valve are spaced on the spring settings. For example, Valve 1 is placed at 900/0.50 = 1800 ft. Valve 2 = 1800 + (spring setting/G_s) = 1800 + (150/0.50) = 2100 ft. Spacings for Valves 4, 5, 6, and 7 = 150/0.50 = 300 ft.

Valve no.	Depth (ft)
1	1800
2	2100
3	2400
4	2700
5	3000
6	3300
7	3600
8	3900

This gives essentially the same spacing as the graphical procedure and is an efficient design.

The formulas for spacing all valves except Valve 1 are:

$$D_{v_2} = D_{v_1} + \frac{S_r}{G_s} \quad (3.427)$$

$$D_{v_3} = D_{v_2} + \frac{S_r}{G_s} \quad (3.428)$$

$$D_{v_4} = \text{etc.}$$

where:

S_r = spring setting = 150 psi for this example.

3.42910 Spacing of completely balanced continuous flow valves

Recall from Section 3.42 that this valve is the concentric flexible sleeve valve and has no tubing pressure effect. It therefore requires a drop in surface opening pressure between valves. It is also charged with a natural gas of specific gravity = 0.60 and at 80°F.

The following example problem was worked by Davis, et al.[23]

Given Data:

Depth = 8000′ to perforations
Desired production rate = 500 BFPD, q_L
Tubing size = 2⅜" OD
Casing size = 5½" — 17#
Salt water content = 90%
Oil = 40° API
Specific gravity gas = 0.65
Separator pressure = 50 psig.
Flowing surface tubing pressure = 100 psig, P_{wh}
Kill fluid gradient = 0.5 psi/ft
Static reservoir pressure = 2600 psi, $\overline{P}_r$
Productivity index = 5, PI
Kick off pressure = 950 psi, P_{ko}
Operating pressure = 900 psi, P_{so}
Volume lift gas available = 300 MCFPD
Surface flowing temperature = 120°F, T_s
Reservoir temperature = 200°F, T_r
Gas/oil ratio = 500 scf/B, GOR
Gas-liquid ratio = 50 scf/B, GLR

The point of gas injection is found as previously described and is noted in Figure 3.484. Considering additional safety factors, use P_{ko} – 50 for kick-off pressure.

Refer to Figure 3.484 for the graphical spacing of the valves locating valves at 1650, 2800, 3550, 4000, and 4250 feet. Space on the flowing gradient line and the casing pressure lines with 25 psi drop between valves. Place the temperature scale on the same graph.

Table 3.477 shows the summary tabulation for these valve settings.

This concentric valve is charged with natural gas (γ_g = 0.60) at 80°F in the shop (Figs 3D.5 through 3D.10—Appendix). Once the valve spacing for the installation has been determined, the test rack set pressure is then calculated.

This determination is made using Figure 3D.9 (Appendix). Determine this test rack set pressure @80°F. by entering the chart at the proper "wellhead casing pressure required to open valve." Then proceed horizontally to intersect the "valve depth" line. Proceed vertically downward to the intersection of the corresponding "temperature at valve °F." point. From this point, proceed parallel to the diagonal lines to the "opening pressure at 80°F." line. Then read vertically downward to obtain the proper value for the set pres-

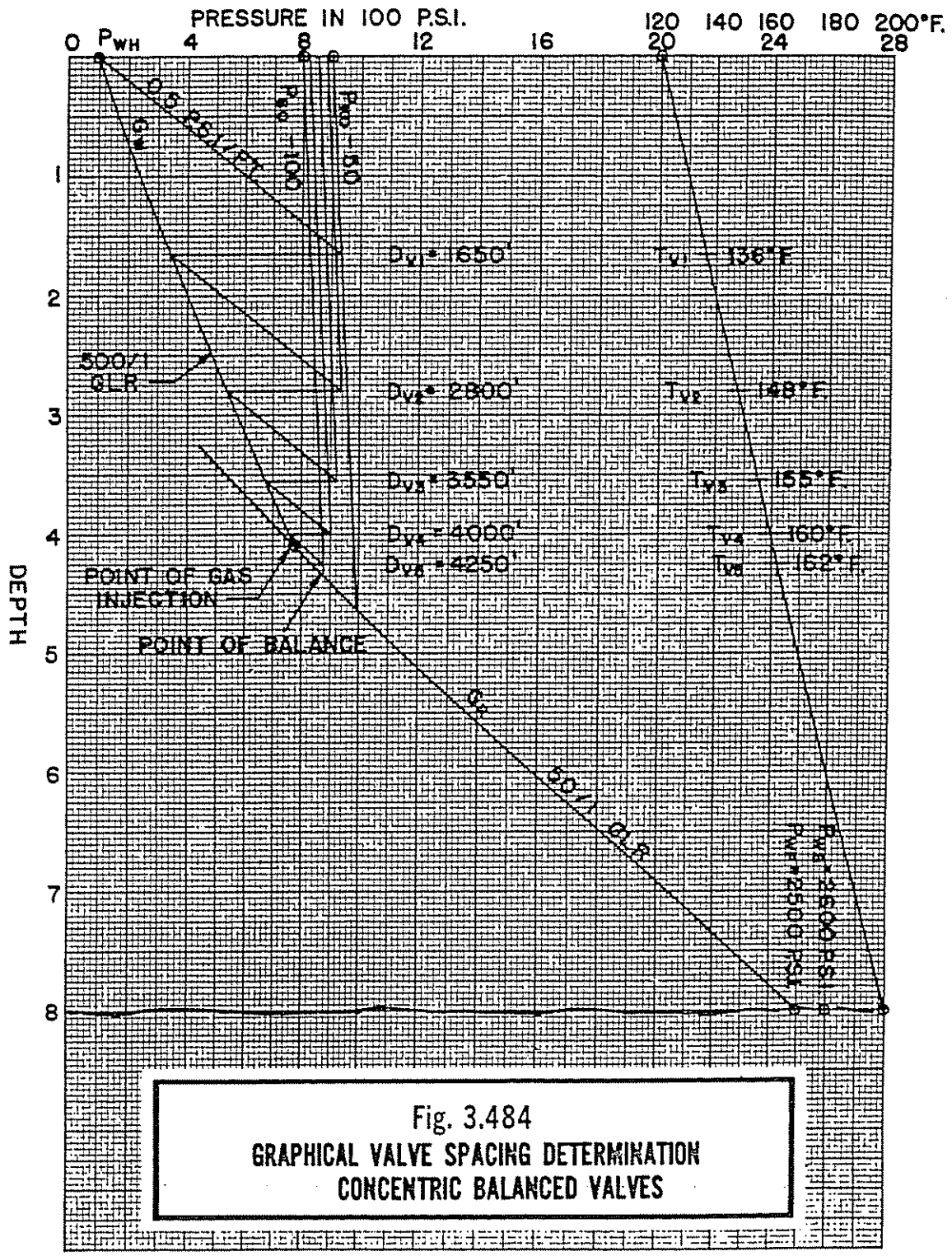


Fig. 3.484 Graphical Valve Spacing Determination, Concentric Balanced Valves (after Davis, et al)

TABLE 3.477
SUMMARY TABULATION—BALANCED VALVES
(25 PSI DROP BETWEEN VALVES)

Valve no.	Surface opening pressure (PSI)	Valve depth (ft)	Depth temp. (°F.)	Test rack set pressure (80°F) (Psi)
1	900	1650	136	820
2	875	2800	148	800
3	850	3550	155	780
4	825	4000	160	760
5	800	4250	162	740

sure at 80°F for the valve. Calculations for this determination from the spacing on Figure 9 are noted in Table 3.477. The calculations may also be made by first determining the opening pressure at the valve from Figure 3A.1 (Appendix) and then making the temperature correction from Figure 3D.7.

These valves serve as excellent continuous flow valves as long as wells are selected so no damage occurs to the flexible element. Also where space becomes a factor, the concentric feature may be desirable.

3.42911 Proportional response system

3.42911 Introduction

Merla has offered a design system that effectively accounts for the throttling action of the valve depending upon tubing pressure. To illustrate the need for a throttling type valve, Merla gave the following discussion regarding an orifice in the tubing (refer to Figure 3.485). In general, this shows that as the tubing pressure increases, the volume of gas that the orifice can pass decreases. This is opposite as to what is needed in the flow configuration, because as the tubing load increases—indicating a greater rate and a lower gas liquid ratio—the valve should pass more gas.

Quoting from Merla:[24] The amount of gas required is illustrated in Figure 3.485. Let us assume for a minute that we believe the formation productivity index is such that a gas-liquid ratio of 1000 scf/B will provide the proper formation pressure to cause the well to produce 400 B/D. Now, suppose we put a proper size hole in the tubing to pass the exact amount of gas needed to produce the well at 1000 scf/B ratio and 400 B/D. This is 400 Mcf. Provided all our data is correct, we have an

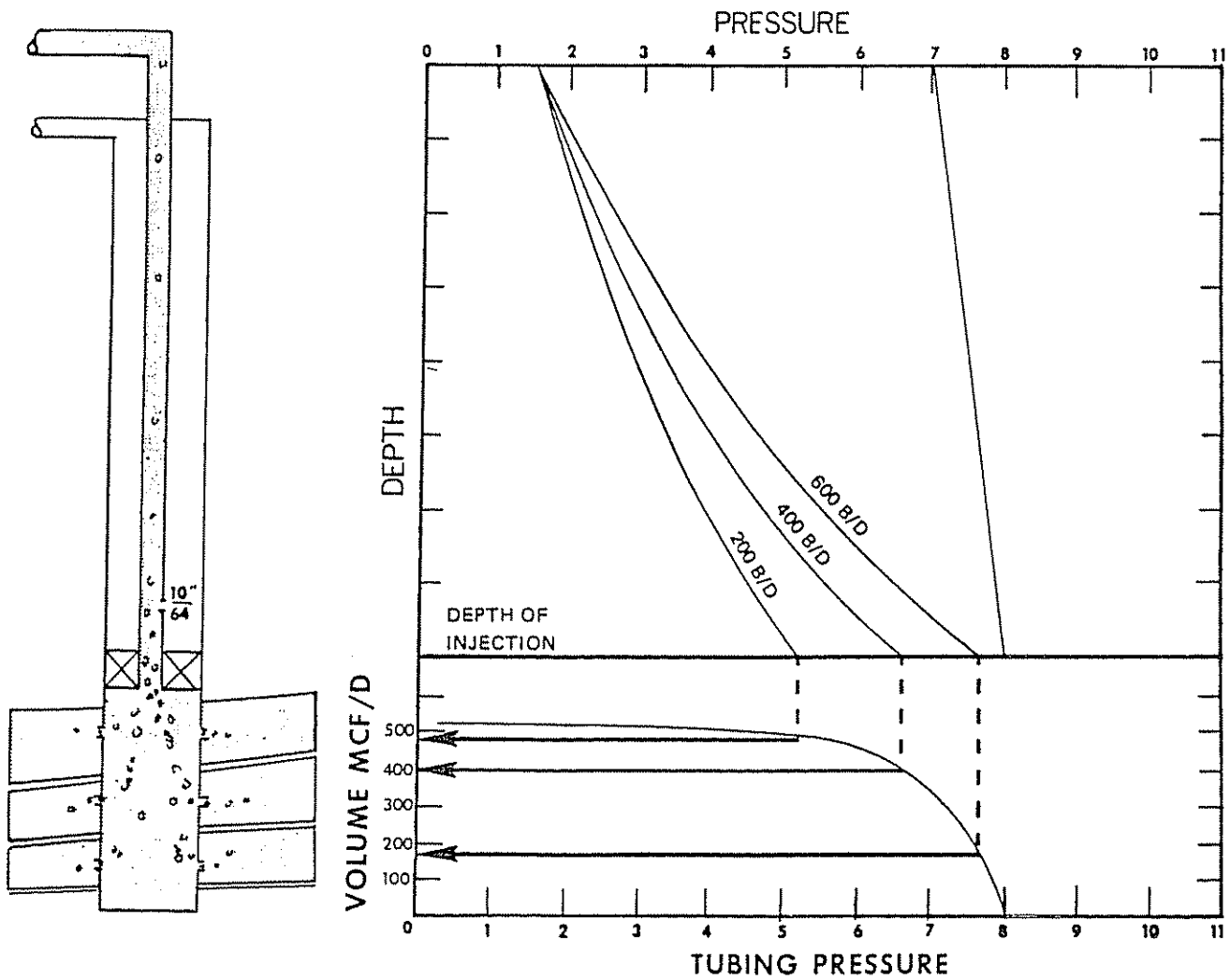


Fig. 3.485 Effect of Tubing Pressure on Gas Volume Passage Through An Orifice (Courtesy Teledyne Merla)

ideal continuous flow installation. However, we seldom have this exact information. Then assume that instead of the well having producing capabilities of 400 B/D, something happens in the formation and it only has the capability of producing 200 B/D. This causes our tubing pressure to be somewhat less than we calculated. What does our orifice do? It passes too much gas as is noted on the gas rate vs. tubing pressure chart. (Approximately 480 Mcf for an injected gas-liquid ratio of 2400 scf/B). What happens if the production rate is higher than we estimated? The orifice does not pass enough gas. (Approximately 180 Mcf for an injected gas-liquid ratio of 300 scf/B). Therefore, our orifice is working contrary to requirements.

From the preceding discussion, a valve is needed that is responsive to the tubing pressure; that is, the valve should act as a variable orifice. It should pass more gas at the higher tubing pressures and less gas at the lower tubing pressures. Such a valve performance is illustrated in Figure 3.486 with its corresponding flow performance curve. Continuing:

Let's look at the example as illustrated in Figure 3.486. From this example, we have a well designed for maximum production, thought to be 400 B/D. The valve is designed to pass the correct amount of gas for 400 B/D which is 200 Mcf. Now suppose the well will only produce 200 B/D. The result is a lower tubing pressure which causes the valve to throttle to a more closed position. In this particular instance the lower production requires 100 Mcf and the valve supplies this exact amount. Conversely, if the rate is 600 B/D, the tubing pressure

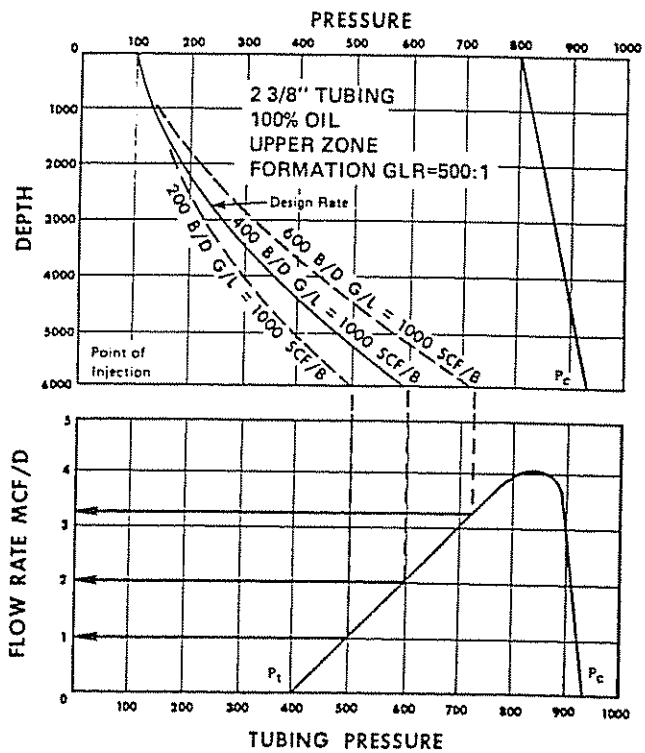


Fig. 3.486 Gas Volume Passage Through A Continuous Flow Valve As A Function of Tubing Pressure (Courtesy Merla)

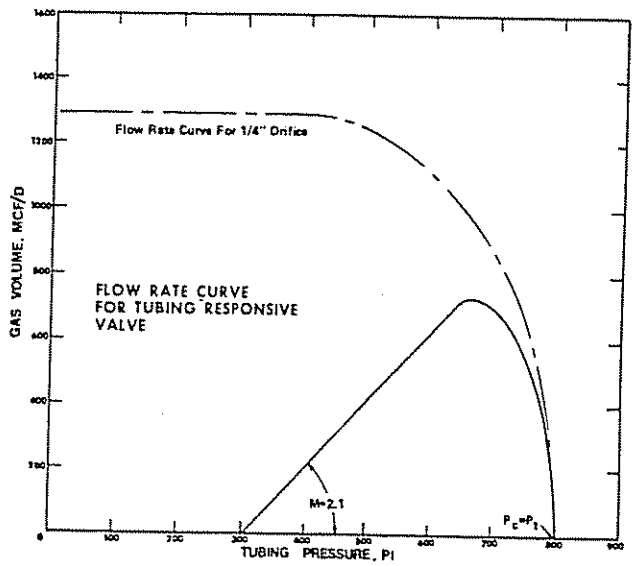


Fig. 3.487 Flow Rate Performance Curve for a Pressure Operated-Tubing Sensitive Valve (Courtesy Teledyne Merla)

increases causing the valve to open further. The gas requirement is 300 Mcf and the valve will pass 320 Mcf. The valve does not always correct to pass the exact amount of gas required but does react to correct in the right direction.

Figure 3.487 illustrates the curve which is generated during actual flow tests. This valve is adjusted for an operating casing pressure of 800 psi and to close when the tubing pressure reaches 300 psi. During the entire test, the casing pressure is held constant at 800 psi and the tubing pressure is varied. The flow rate curve for a ¼" orifice is also shown for comparison.

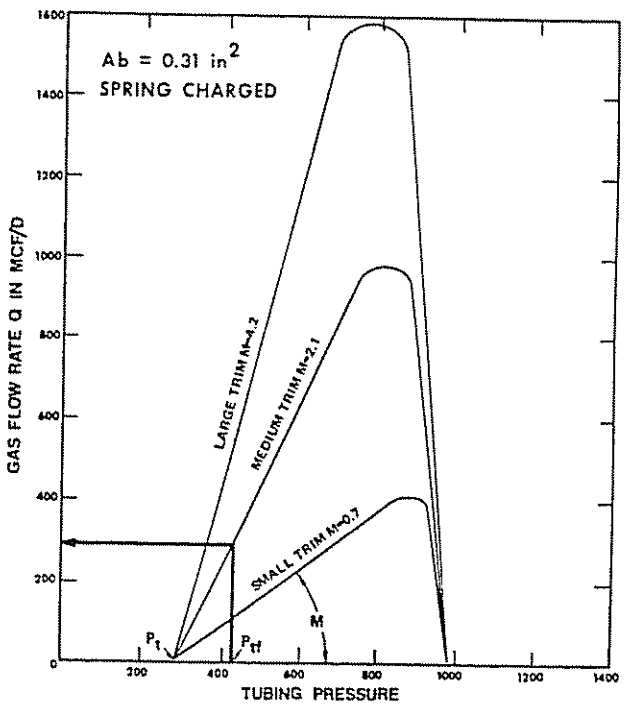


Fig. 3.488 Flow Rate Performance Curves for Different Sized Valves (Courtesy Teledyne Merla)

Figure 3.488 shows how the flow rate performance curves differ for three different gas lift valve trims. The left side of each curve contains the useful information. This portion of the curve is a straight line. Fix the left side of the curve as the tubing pressure at which the valve should close P_t and if the slope of the curve is known the difference in flow rate for a known change in tubing pressure can be determined.

From Figure 3.488, determine the gas passage for any tubing pressure. As an example, if the tubing pressure is 440 psi and the valve contains medium trim, the flow rate through the valve is 295 Mcf. Gas flow can also be calculated by the formula:

$$Q_{inj} = M(P_{tfp} - P_{tf}) \qquad (3.429)$$

where:

P_{tfp} = flowing tubing pressure
M = slope of valve performance curve (straight line portion)
P_{tf} = tubing pressure at which the valve closes

Merla offers the following discussion concerning the design of a continuous flow gas-lift installation by this system.

3.42912 Transfer operation [24]

The unloading operation should provide a smooth transfer of the point of injection from the top valve down to the lowermost valve where operation is sustained. The design procedure is such that gas injection is commenced at the upper valve with a tubing pressure P_{tfp} only slightly less than the casing pressure P_{vo} (valve essentially wide open). These conditions give the maximum effective port size and, therefore, maximum gas passage capabilities in the valve. As injection continues, the tubing pressure is reduced, resulting in a decreased gas passage through the valve. Normally, the valve is set to close completely when the tubing pressure reaches the design gradient curve that will be obtained when injection is from the operating valve.

Gas flow through the valve decreases as the tubing pressure decreases. Thus, to sustain smooth transition from valve to valve, injection must begin from the next lower valve while the upper valve is passing sufficient gas to sustain production. A smooth transition from valve to valve is the key to the prevention of intermittent action during unloading.

The term *transfer point* designates the tubing pressure that must be achieved at the upper valve to permit injection to begin from the next lower valve P_{tfp}. Figure 3.489 clarifies the following explanation.

The unloading operation begins with the upper valve supplying all the gas. The flowing tubing pressure is reduced until the transfer pressure (P_{tfp}) is reached. At this point, the next valve is uncovered and starts injection of gas. Since the total injection from the two valves exceeds the amount necessary to maintain the pressure gradient, the tubing pressure opposite the upper valve will continue to decrease until it reaches the close pressure, P_{tf}. The injection continues through the second valve alone until the transfer point for the second valve is reached.

Figure 3.489 A illustrates a condition in unloading where only the top valve is passing gas. The next lowermost valve is just ready to be uncovered and start passing gas. Figure 3.489 B shows both valves passing gas. The tubing pressure is low at the top valve and it is passing a small volume of gas compared to the next valve, which, having a high tubing pressure, is passing near maximum gas for the casing pressure available.

Each valve must supply sufficient gas to meet the well requirements up to the transfer point, and they should each close when the transfer operation is complete. Since there is no transfer from the bottom valve, this valve is not set to close but to regulate the proper amount of injection gas.

The important valve characteristic during the transfer operation is the throttling action which provides the proper distribution of gas between the two valves, i.e. a larger percentage of the gas is injected through the lower of the two valves as illustrated in Figure 3.489.

3.4291121 Transfer point selection

The selection of the transfer point has an effect on two factors:

(1) As the transfer point pressure (P_{tfp}) is increased, the gas passage ability of the gas lift valve increases.
(2) As the transfer point pressure (P_{tfp}) is increased, the required GLR decreases due to a decreased rate of production and increased flowing bottom hole pressure. Therefore, the injection gas volume required decreases.

The transfer point must be high enough to assure that the gas flow through the valve will be equal to or greater than the gas volume required to support the production rate. The actual selection of the transfer point may be done graphically or by trial and error. The following example illustrates selection of the transfer point by various methods.

3.4291122 Gas requirements

The amount of gas required at the transfer point is based on:

(1) The production rate (q_L) achieved when the tubing pressure at the valve is reduced to the transfer point pressure (P_{tfp}).
(2) The total gas liquid ratio (R_T) required to support the production rate (q_L) with a given well head pressure and a flowing tubing pressure (P_{tf}) at the valve. (R_T includes injection gas and solution gas.)

The production rate is obtained by projecting the flowing gradient line (using formation gas liquid ratio) from P_{tfp} at the valve to the formation depth to arrive at a flowing bottom hole pressure (P_{wf}). This flowing bottom hole pressure can be applied to the I.P.R. curve or P.I. equation to find the production rate (q_L).

The required total gas-liquid ratio can be calculated or determined from the two-phase pressure traverse curve for the given production rate (q_L) that passes through the surface well head pressure at zero depth and the transfer point (P_{tfp}) at the depth of the valve.

The required injection gas volume (Q_{inj}) is the product of the production rate (q_L) and the difference between

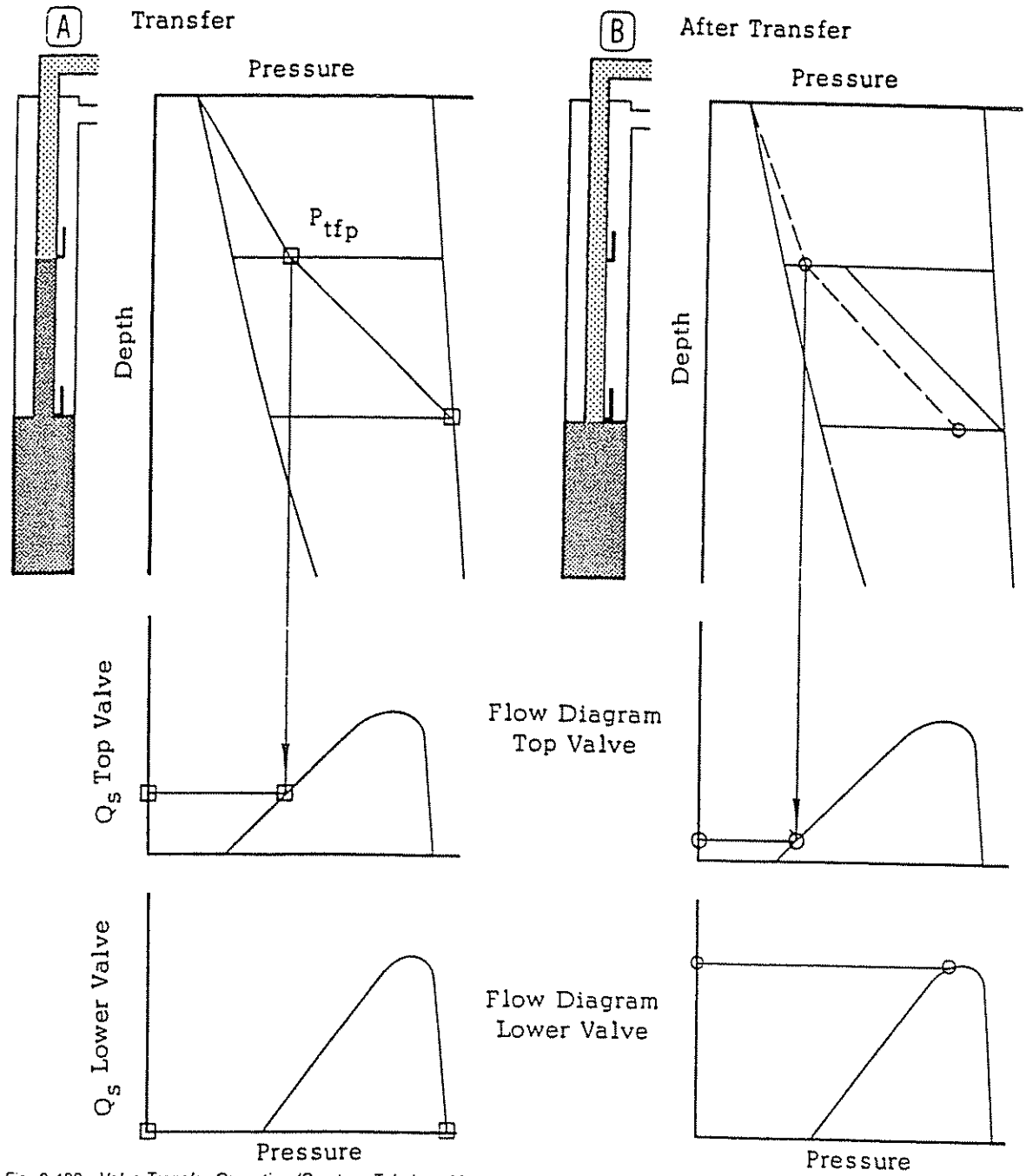


Fig. 3.489 Valve Transfer Operation (Courtesy Teledyne Merla)

formation gas-liquid ratio (R_s) and total gas-liquid ratio (R_T).

$$Q_{inj} = q_L (R_T - R_s) \qquad (3.430)$$

3.4291123 Gas supply

The amount of gas injected through the gas lift valve is determined by the difference between the transfer pressure (P_{tfp}) and fluid closing pressure (P_{tf}) as shown in the throttling slope equation.

$$Q_{inj} = M(P_{tfp} - P_{tf})$$

3.429113 Design example—detailed information available

This section describes the procedure for designing a continuous flow gas lift installation. Detailed information has been given for the design. A valve with gas passage characteristics similar to Figure 3.487 is used.

Given data:

Production rate	600 B/D (100% oil)
Depth	7200 ft
Perforations	6900 - 7100 ft
Casing	5½", 17 lbs.
Tubing	2⅜", J55, 8Rd

$\overline{P}_R$	2000 psi at 7000 ft
Static fluid level	2550 ft
Well head temp	74°F
Geothermal gradient	0.016°F/ft
Formation gas-oil ratio	100 SCF/B
Static gradient	0.37 psi/ft
Productivity index	0.8 B/D/psi
Gas lift inj. press	1000 psi
Gas specific gravity	0.65
Gas quantity available	300 Mcf/D
Flowing tbg. head press	150 psi
Load fluid	0.45 psi/ft

3.4291131 Preliminary well analysis

A preliminary well analysis determines: (1) the feasibility of continuous flow with existing conditions and (2) depth of injection required (see Figure 3.490). A step-wise development follows and the corresponding step-wise number can be found on Figure 3.490.

(1) The maximum quantity of gas available for injection is 300 Mcf/D, the desired rate is 600 B/D, and the formation gas-liquid ratio is 100 scf/B. This fixes the total gas-liquid ratio at 600 scf/B. (Injection gas-liquid ratio is 300 Mcfd/600 B/D or 500 scf/B). The total produced ratio, R_T, is 500 scf/B injected plus 100 scf/B from the formation = 600 scf/B. Construct this curve by extending the flowing gradient curve downwards from P_{wh} = 150 psi for 600 B/D and G/L = 600 scf/B in 2" tubing.

(2) Plot the static bottom hole pressure of 2000 psi at 7000 feet, which is the middle of the perforated interval.

(3) The flowing bottom hole pressure is either read from the I.P.R. curve if available or is calculated as follows for this case for an assumed constant PI.

$$PI = J = q_L/(\overline{P}_R - P_{wf})$$

$$(\overline{P}_R - P_{wf}) = q_L/J = \frac{600}{.8} = 750$$

$$P_{wf} = \overline{P}_R - 750 = 1250 \text{ psi}$$

(4) Beginning with the flowing pressure of 1250 psi, project the gradient below the point of injection upwards until it crosses the 600 SCF/B gradient traverse. A formation G/L of 100 SCF/B is used for this traverse.

(5) Draw in the 0.45 psi/ft static gradient line starting at 0 psi and 2550 feet. It should intersect the $\overline{P}_R$ of 2000 psi at 7000 feet. (Note that the 0.45 psi/ft gradient is used since this is the gradient of the load fluid.)

(6) Draw in the gas gradient for a 0.65 gravity gas and 1000 psi surface pressure until reaching total depth.

This completes the analysis. An examination of the graph in Figure 3.490 shows that continuous flow is possible at the desired conditions with a point of injection at approximately 6100 feet.

A pressure flow rate diagram can be constructed to show the maximum rate possible from this well. This procedure has been previously described in Section 3.4232.

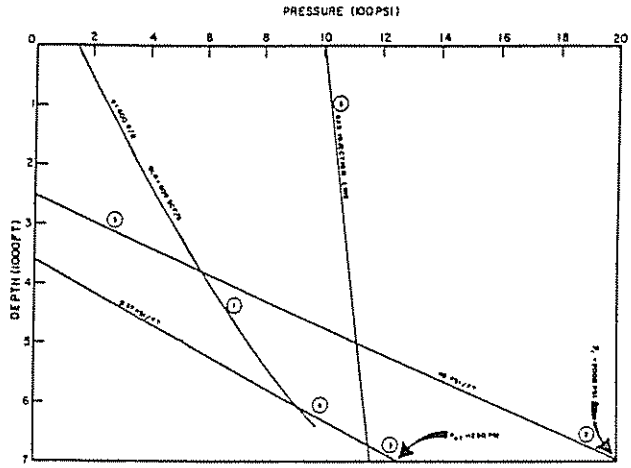


Fig. 3.490 Preliminary Well Analysis

3.429132 Gas lift valve spacing and selection

Figure 3.491 shows a graphical solution to this example problem and Table 3.478 details the resulting information.

Top Valve (First Valve)

(1) From the surface gas pressure of 1000 psi, construct the gas pressure line, including gas column weight. (γ_g = 0.65). Refer to Appendix 3A.1.

(2) Obtain the depth of the top valve by placing it at U-tube depth or at the static fluid level. Placing the valve at U-tube depth affords unloading under all conditions and is the normal procedure. Construct a 0.45 psi/ft gradient line from the expected tubing pressure of 150 psi until it crosses the gas operating casing pressure line. The top valve depth is 2000 feet.

(3) Determine gas pressure at the top valve depth from the pressure-depth graph where the depth of the valve crosses the gas gradient line. This valve pressure is 1050 psi.

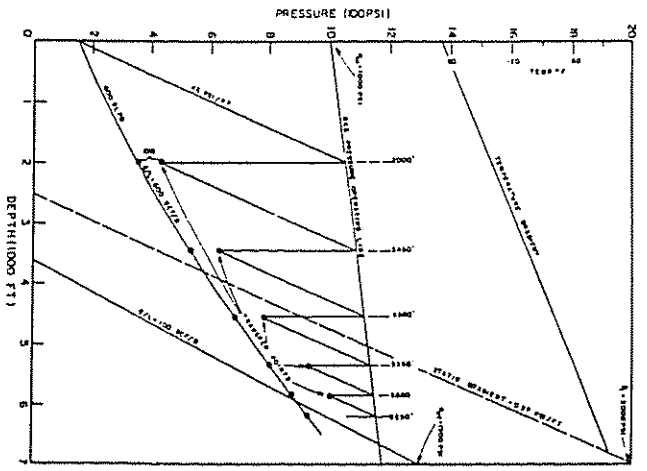


Fig. 3.491 Graphical Gas Lift Design

(4) Construct the flowing pressure traverse downwards from $P_{wh} = 150$ psi. This is for 600 B/D for a G/L = 600 SCF/B and for a 2⅜" O.D. tubing. Obtain minimum tubing fluid pressure at valve depth from the graph where the two-phase traverse crosses the depth of the valve. Pressure is 355 psi at 2000 feet.

(5) Select the transfer point (P_{tfp}) by trial and error. Normally a value of 50 to 100 psi higher than P_{tf} is used as a starting point. Should the value be too low, the valve will not pass enough gas and a higher value must be assigned. If the value is too high, the spacing will be closer than necessary. In this case, 75 psi was selected giving a value of $P_{tfp} = 355 + 75 = 430$ psi.

(6) Calculate the flowing bottom hole pressure achieved when the transfer point is reached as follows:

$$P_{wf} = P_{tfp} + (D_w - D_v)(G_s)$$
$$P_{wf} = 430 + (7000 - 2000)(0.45) = 2680 \text{ psi}$$

(7) The production rate achieved with this flowing bottom hole pressure may be obtained from the I.P.R. curve or by calculating it from the equation $q_L = J(\bar{P}_R - P_{wf})$. In this case the flowing bottom hole pressure is greater than the static and no inflow is possible. However, it is necessary to continue to unload the kill fluid from the annulus until the point where the formation inflow begins. An unloading rate of 200 B/D is assumed.

(8) The total gas-liquid ratio is that ratio required to produce the rate calculated in Step 7 with a tubing pressure of 150 psi at zero depth and 430 psi (P_{tfp}) at 2000 feet. This is obtained from the gradient curves or is calculated, and it is 300 SCF/B.

(9) The formation gas-liquid ratio is 100 SCF/B. However, since there is no inflow and the well is only producing load fluid, the formation ratio is zero while unloading from the top valve.

(10) The injection gas-liquid ratio is 300 SCF/B. $R_{inj} = R_T - R_s = 300 - 0 = 300$ SCF/B.

(11) Gas required. $Q_{inj} = R_{inj}(q_L) = (300)(200)/1000 = 60$ Mcf/D

(12) Temperature at the valve may be determined from the temperature gradient curve constructed on the design graph. The temperature at 2000 ft. is 106°F.

(13) A temperature correction factor (T_f) may be found from the equation

$$T_f = \sqrt{\frac{520}{T_v + 460}} = \sqrt{\frac{520}{566}} = .958$$

(14) The corrected gas rate corrects the gas volume from actual temperature to a temperature base of 60°F which is the basis for performance data. The corrected volume is Q/T_f or $60/0.958 = 63$ Mcf/D. A correction factor may be noted in Fig. 3C.1 in the Appendix which takes care of both gas gravity and temperature.

(15) Find the difference between the transfer point pressure and the minimum flowing tubing pressure. This difference in pressure governs the amount of gas that the valve will pass. $P_{tfp} - P_{tf} = 430 - 355 = 75$ psi

(16) Select a valve. Normally, a valve with a small trim is selected for the upper unloading valves. The small trim should be used until P_{tfp} is increased to $(P_{tfp} - P_{tf}) = 20\%$ of $(P_{vo} - P_{tf})$. Select a valve with M = 1.0, and ⅜" ball. (See Fig. 3.492).

(17) The gas passage of this valve is either read directly from a performance curve or may be calculated from the equation:

$$Q_{inj} = M(P_{tfp} - P_{tf}) = 1.0(75) = 75 \text{ Mcf/D}$$

This assumes we have a value of M on the straight line portion of the valve performance curve; hence, it is necessary to verify that the selection is on that portion of the curve.

$$\text{We find } P_{vo}/P_{tp} = \frac{1050}{355} = 2.96$$

and reading the corresponding Q_m/P_{vo} ratio from the performance data of Fig. 3.492 gives 0.66. If $Q_m/P_{vo} = 0.66$ and $P_{vo} = 1050$, we can calculate the maximum flow rate Q_m.

$$Q_m/1050 = 0.66 \qquad Q_m = 695 \text{ Mcf/D}$$

The Q_m must equal or exceed the rate shown in Step 15, which was 75 Mcf/d. Since it is much larger, we are assured that the desired gas volume does not exceed the maximum volume possible through the valve. If this occurs, a larger trim size must be selected and calculations repeated.

(18) An F_e value of 0.22 is determined from performance data. (See Fig. 3.492).

BELLOWS AREA = 0.23
ADJ PRESSURE RANGE = 600-1500 PSI

BALL SIZE	3/8	1/2	5/8
SEAT	45°	45°	45°
PORT	0.25	0.25	0.40
F_e	0.22	0.57	0.72
M	1.0	2.6	3.4

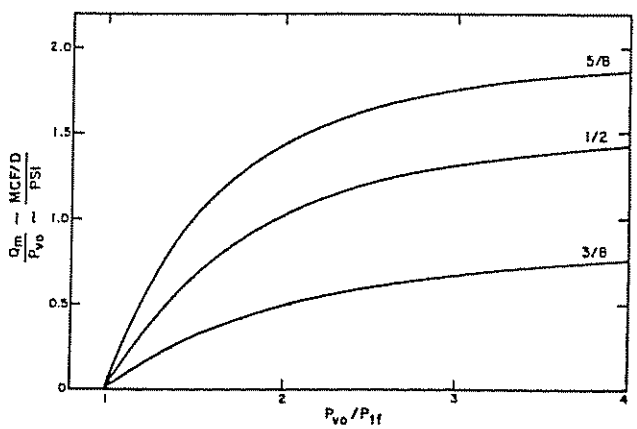


Fig. 3.492 Gas Lift Valve Performance Curves (Courtesy Teledyne Merla)

(19) Calculate the valve adjustment pressure (P_{vc}) from the formula

$$P_{vc} = P_{vo} - F_e(P_{vo} - P_{tf}) = 1050 - 0.22 (1050 - 355) \quad (3.431)$$

$$P_{vc} = 897 \text{ psi}$$

Check to determine if this adjustment pressure range is possible with the bellows selection made previously. From the performance data of Figure 3.492, the adjustment pressure range for the valve is 600-1500. Therefore, the selection is satisfactory.

Second Valve

(1) Obtain the depth of the second valve by constructing a 0.45 psi/ft gradient line from the transfer pressure (P_{tfp}) at the top valve until it crosses the gas operating casing pressure line. This depth is 3450 feet. Draw a horizontal line from the casing pressure back to the tubing two-phase gradient line from this valve depth of 3450 ft.

(2) The injection gas pressure at the valve is read directly from the graph at 3450 feet. It is 1085 psi.

(3) The minimum fluid pressure is read directly from the graph at 3450 ft and is 525 psi.

(4) *Transfer Point.* Again, a trial and error selection must be made but this time there are some clues. From the graph it is impossible to cause any drawdown from this valve depth since the minimum tubing pressure possible is above the static gradient line. This clue indicates that it will only be necessary to unload the load fluid from the annulus. Clue 2 is that the calculated gas passage at the top valve (75 Mcf/D) was only slightly higher than that required (63 Mcf/D). Since more gas is required at deeper depths, a higher transfer point must be selected at this valve. Therefore, select a transfer pressure of 100 psi higher than the tubing pressure or 525 psi plus 100 psi = 625 psi. Should this prove to be insufficient gas, a higher pressure will be used in the next trial and error selection which will cause the gas lift valve to pass more gas.

Steps 5 through 19 are to be completed in the same manner as the steps for the top valve. Note that the gas requirement is 80 Mcf. This is again determined assuming an unloading rate of 200 B/D and finding that a G/L of 400 SCF/B total is necessary to connect the P_{tfp} = 625 psi to the P_{wh} of 150 psi. Solution gas is still not helping due to no feed-in of well fluids. The gas lift valve passage rate is 100 Mcf ($Q_{inj} = M(P_{tfp}) = 1.0(100) = 100$ Mcf/D) Since this gas supply is sufficient, we may proceed to the next valve.

Third Valve

(1) The depth of third valve is determined by constructing a 0.45 psi/ft gradient line through P_{tfp} at the second valve until it crosses the gas injection pressure line. This is 4550 feet. Draw in a horizontal line to the left until intersecting the pressure traverse line.

(2) *Transfer Point.* In this case inflow from the formation will commence and higher gas rates will be required. Expect some formation gas which will reduce the injection requirements. Examination of the transfer point of the valve above is again made and 100 psi greater is assumed. The P_{tfp} is found to be 680+100=780 psi.

(3) All additional steps for valves 1 and 2 are repeated and this assumption for P_{tfp}=780 psi is satisfactory.

Fourth Valve

(1) The depth is again defined by starting at the transfer point of the third valve and constructing a 0.45 psi/ft gradient until it crosses the gas gradient line at 5350 feet. The 0.45 psi/ft is still used and is a slight safety factor due to fluid feed-in. However, it will permit unloading water if required at a later date.

(2) A transfer point of 930 psi (805 + 125) is assumed.

(3) This selection is then analyzed for previous valves.

(4) The gas passage rate for the valve (M = 1.0) is 125 Mcf/D. which is not equal to or greater than the required gas of 151 Mcf/D. Therefore, assume a higher transfer point or go to a trim size that will pass more gas. The selection of a higher value for P_{tfp} results in a lower gas requirement but gives a higher gas passage through the valve. An increased value of P_{tfp} will also reduce the spacing to the next valve. The selection of the next larger trim size without changing the P_{tfp} will cause the valve to pass more gas while the gas required remains the same. This appears to be the wiser choice; therefore, a medium trim size is selected.

(5) Note that the gas passage is considerably more than that required—325 Mcf/D vs. 151 Mcf/D required. If desired, the P_{tfp} for Valve 4 could be reduced which would increase gas requirements at the transfer point and would decrease the gas passage rate through the valve. This would also increase spacing distance to the next valve. However, in this case, it does not appear that sufficient changes could be made to place the valve below the minimum injection depth of 6100 feet; therefore, no changes are made.

Fifth Valve

The valve depth was 5850 feet. All other steps are repeated and results are tabulated in Table 3.478.

Sixth Valve (Bottom and Operating Valve)

(1) The sixth valve location is determined by constructing a 0.45 psi/ft gradient through the P_{tfp} of the fifth valve until it crosses the gas injection pressure line, or 6200 feet.

In this case, required depth of injection is at 6100 feet. Since 6200 feet is sufficiently close, use it as the depth of the bottom valve in the design. The valve could have been placed at 6100 feet; however, the accuracy of the data is not normally that good. In many cases an extra valve or more may be placed in the string below this valve depth as a safety factor.

(2) The same surface pressure of 1000 psi is maintained.

(3) The casing pressure at the valve of 1155 is read directly from the graph.

(4) Since this is to be the operating valve, consider-

able deviation takes place in the numerical sequence that has been followed. In the analysis of this design, a rate of 600 B/D with a total gas-liquid ratio of 600 SCF/B was to be maintained from 6100 feet. A multi-phase flow traverse for this rate and ratio was constructed and is the curve expected from injection at the bottom valve. In the upper valves, the minimum tubing pressure, P_{tf}, was used to fix the tubing pressure at which the valve would close. However, in this case, the valve should be open, passing the desired quantity of gas to maintain production. Therefore, leave Step 4 blank for the present.

(5) The transfer point is non-existent at the bottom valve since the injection point is not transferred any lower. This point now becomes the tubing pressure expected at the bottom valve. In addition, the bottom valve will adjust to pass the correct volume of gas to create this tubing pressure. Read the pressure (P_{tfp}) directly from the two-phase gradient on the graph (930 psi).

(6) Steps 6 through 14 are repeated as for Valve 1 and the gas volume is 342 Mcf/D. The valve is adjusted to pass this volume. A valve with a small trim on bottom is preferred since it responds gently to changes in tubing pressure. This is a trial and error selection and, should the valve prove not to have the gas flow capabilities required, the next trim size, medium trim, will be selected. In this case, the small trim is selected which has a slope of M=1.0 (⅜" ball).

(7) Since the Q_{inj} supplied by the valve and the slope M are known, determine ($P_{tfp} - P_{tf}$) from the formula as follows:

$$Q_{inj} = M(P_{tfp} - P_{tf})$$
$$(P_{tfp} - P_{tf}) = Q_{inj}/M = 342/1 = 342 \text{ psi}$$

(8) Since P_{tfp} is 930 and $P_{tfp} - P_{tf} = 342$, $P_{tf} = 930 - 342 = 588$. If P_{tf} is less than P_{tf} (minimum), the next higher trim size must be selected.

(9) $P_{vo} = 1155$ and $P_{tf} = 588$ $\frac{P_{vo}}{P_{tf}} = 1155/588 = 1.96$

(10) From Figure 3.492 $\frac{Q_m}{P_{vo}} = 0.5$ for ⅜" ball.

(11) $Q_m = (0.5)(P_{vo}) = 0.5(1155) = 577$ Mcf/D
This is a very important calculation on the bottom valve. If the maximum flow rate, Q_m, through the valve does not exceed the desired rate (342 Mcf/d), the next size trim must be selected.

(12) F_e is selected from valve specification data and is 0.22.

(13) $P_{vc} = P_{vo} - F_e(P_{vo} - P_{tf}) = 1155 - 0.22 \times (1155 - 588)$
$P_{vc} = 1030$
A final tabulation is noted in Table 3.478.

INFORMATION AND PROCEDURE NEEDED FOR SELECTION OF BOTTOM VALVE

1 Valve depth (D_v in ft)
2 Surface gas pressure (P_{so} in psig)
3 Gas pressure at valve (P_{vo} in psig)
4 Flowing tubing pressure on gradient line at valve depth (P_{tfp} in psig)
5 Flowing bottom hole press. [$P_{wf} = P_{tfp} + (D_w - D_v)(G_s)$ in psig]
6 Production rate [$q_L = J(\bar{P}_R - P_{wf})$ or from IPR curve in B/D]
7 Total gas-liquid ratio (R_T in ft³/bbl)
8 Formation gas-liquid ratio (R_s in ft³/bbl)
9 Injected gas-liquid ratio ($R_{inj} = R_T - R_s$)
10 Gas required ($Q_{inj} = R_{inj} \times q_L/1000$ in MCF/D)
11 Temperature at valve depth (T_v in °F.)
12 Temperature correction TCF
13 Corrected gas rate (Q_{inj}/TCF in MCF/D)
14 Trim selection of gas lift valve
15 Slope (refer to gas lift valve perf. data M in MCF/D ÷ psi)
16 Gas supplied by valve [$Q_{inj} = M(P_{tfp} - P_{tf})$ in MCF/D]
17 Calculate $P_{tfp} - P_{tf} = Q_{inj}/M$ and solve for unknown (P_{tf})
18 P_{tf} in psig. Needed for steps 19 and 23.
19 Check peak flow rate. (P_{vo}/P_{tf} Enter perf. curve, go to trim to find Q_{inj}/P_{vo})
20 From performance curve (Q_{inj}/P_{vo} in MCF/D ÷ psi)
21 Maximum gas rate ($Q_{inj} = Q_{inj}/P_{vo} \times P_{vo}$ in MCF/D)
22 From performance data (F_e)
23 Adjustment pressure [$P_{vc} = P_{vo} - F_e(P_{vo} - P_{tf})$ in psig]
24 Valve no. or station no.
25 Quantity of valves at station
26 Type valve
27 Spring no.
28 Bellows size

The following nomenclature is used:
P_{tf} (min) = $P_{vo} - \Delta P_{GG}$
$\Delta P_{GG} = P_{so} - P_{wh}$
P_{so} = surface operating gas pressure
P_{wh} = flowing wellhead pressure
P_{vo} = operating pressure at valve (open)

3.429114 Computer designed example

Following is a computer design of the same example as found in Section 3.42921. The first data are the input data file. The second data group is the program output at the terminal.

A graphic presentation noted in Figure 3.491 (B) is manually plotted from the information on the program output for better understanding of the completed design. This design makes use of the Ros correlation; hence, the slight difference in the rate possible, that is, 950 B/D as compared to 1000 B/D as predicted by the Hagedorn and Brown correlation.

INPUT DATA FILE

```
RED        15:37CST        01/03/78

100 UNIV. OF TULSA KERMIT BROWN EXAMPLE
110 CFDVKLEN  GARLAND  3 JAN 78
120 0  0  2  6.765  2.875  2.441  8000
130 .7  65  .7  1.05  32  0  0
140 1  18  18
150 LM-15R  (HP  SP-5)  450-1120/SAME
160 1935  0  .4  1  0  0  5  0
```

TABLE 3.478
SUMMARY TABULATION OF PROPORTIONAL RESPONSE SYSTEM

Valve no.	1	2	3	4 (Initial)	4* (Final)	5	6
1 Valve depth-D_v (ft)	2000	3450	4550	5350	5350	5850	6200
2 Surface gas pressure P_{so}	1000	1000	1000	1000	1000	1000	1000
3 Gas pressure at valve P_{vo}	1050	1085	1115	1135	1135	1145	1155
4 Minimum fluid pressure at valve depth	355	525	680	805	805	875	588
Assume							
5 Transfer point P_{tfo}	430	625	780	930	930	1000	930
Calculate gas required							
6 Flowing BHP $P_{wf} = P_{tf} + (D_w - D_v)(G_s)$	2680	2225	1685	1540	1540	1425	1225
7 Production rate $q_L = J(\bar{P}_R - P_{wf})$(BPD) or from IPR curve	Unload only Assume 200 b/d		252	368	368	575	620
8 Total gas-liquid ratio R_T (scf/B) (from gradient curves)	300	375	450	475	475	550	600
9 Formation gas-liquid ratio R_s (scf/B)	0	0	100	100	100	100	100
10 Injected gas-liquid ratio $R_{inj} = R_T - R_s$ (scf/B)	300	375	350	375	375	450	500
11 Gas required $Q_{inj} = R_{inj}(q_L)/1000$ (Mcf/D)	60	75	88	138	138	259	310
12 Temperature at valve T_v (°F)	106	130	148	161	161	169	174
13 Temperature correction TCF	0.958	0.939	0.925	0.915	0.915	0.910	0.906
14 Corrected gas rate $\frac{Q_{inj}}{TCF}$ (Mcf/D)	63	80	95	151	151	285	342
Calculate valve gas passage rate							
15 $P_{tfo} - P_{tf}$	75	100	100	125	125	125	342
16 Ball and seat description (Trim)	S	S	S	S	M	M	S
17 Slope, M	1.0	1.0	1.0	1.0	2.6	2.6	1.0
18 Q_{inj} from graph or $Q_{inj} = M(P_{tfo} - P_{tf})$	75	100	100	125	325	325	342
19 Check peak flow rate P_{vo}/P_{tf}	2.96	2.07	1.64		1.41	1.31	1.96
20 From performance curve Q_m/P_{vo}	0.66	0.52	0.48		0.59	0.48	0.5
21 $Q_m = Q_m/P_{vo} \times P_{vo}$	695	565	535		670	550	577
Valve adjustment							
22 From performance data F_e	0.22	0.22	0.22	NOT	0.57	0.57	0.22
23 Adjustment pressure $P_{vc} = P_{vo} - F_e(P_{vo} - P_{tf})$	897	962	1019	ENOUGH	947	990	1030

*Note Valve #4 — Calculations were repeated for medium trim and final selection was medium trim.
Since there may be some confusion in the calculation of the bottom valve, the following information given by DeMoss helps clarify this selection.

170 950 .5 200 109 170
180 135 135 1065 915 6000 1150
190 0 0 0

CONTINUOUS FLOW GAS LIFT DESIGN FOR:
UNIV. OF TULSA KERMIT BROWN EXAMPLE
CFDVKLEN GARLAND 3 JAN 78
DESIGN OPTION II — CONSTANT P.I.
UPPER VALVES / LOWER VALVES
LM-15R (HP SP-5) 450-1120/SAME
PRODUCTIVITY INDEX : 5.00 B/D/PSI
DESIGN PRODUCTION RATE : 950. B/D
DESIGN GAS-LIQUID RATIO : 1150. CF/B
WATER CUT : 0.500
DESIGN FLOWING WHP : 135. PSIA
GAS INJECTION OPERATING PRESSURE :
915 PSIA
GAS INJECTION RATE : 902500. CF/D
DESIGN FLOWING BHP : 1745. PSIA
STATIC BHP : 1935. PSIA
FLOW CONDUIT : TUBING
CASING ID 6.765 TUBING OD 2.875 TUBING ID
2.441
REFERENCE DEPTH (FT.) : 8000.
MAXIMUM INJECTION DEPTH (FT.) : 6000.
INSTALLATION SPECIFICATIONS :

STATION	DEPTH (FEET)	VALVE SIZE	ADJ. PRES PSIG	NUMBER OF VALVES
1	2351.	1	869.	1.
2	3446.	1	919.	1.
3	4141.	1	951.	1.
4	4602.	2	900.	1.
5	5105.	2	932.	1.
6	5503.	2	970.	1.
7	5728.	2	991.	1.
8	5870.	2	850.	1.

ADDITIONAL OPERATING DATA :

STATION	QTT (N)	QR (N)	PRF (N)	PRT (N)	PRCF (N)
1	275.	54598.	491.	566.	975.
2	275.	76133.	634.	744.	1004.
3	275.	89216.	729.	849.	1022.
4	275.	115041.	795.	845.	1033.
5	275.	128229.	848.	898.	1046.
6	275.	137118.	912.	972.	1057.
7	275.	137311.	949.	1009.	1062.
8	950.	902500.	634.	972.	1066.

3.4210 Continuous flow gas-lifting directionally drilled wells

3.42101 Introduction

Brown and Beggs presented the following discussion on the continuous flow gas lifting of directionally drilled wells.[25]

Because of the increase in the number of directionally drilled wells, it has become necessary to look at the differences in gas lift design procedures in these wells as compared to vertical wells. Directional wells may commonly be drilled at angles varying from 0-75° from the vertical. Figure 3.493 shows a well which is drilled at 83° from the vertical, having a horizontal distance of approximately twice the vertical distance.

For many years it has been assumed that these wells can be treated as vertical for purposes of gas lift design and the pressure loss in the tubing or annulus calculated as if the flow strings were vertical. Because of some of the inherent safety factors built into gas lift design, it has been possible to lift these wells successfully. Some of these safety factors have been to neglect the gas column weight and to use a 0.50 psi/ft kill fluid gradient instead of the true gradient of a lighter kill fluid, such as a crude oil, which might have a gradient of less than .40 psi/ft.

Other designers have added safety factors to the calculated pressure traverse in the tubing string based on the angles, such as using $p'_{wf} = p_{wf} + .10\ p_{wf}$ for angles between 10 and 30° and $p'_{wf} = p_{wf} + .20\ p_{wf}$ for angles between 30 and 70°.

This section presents a method for gas lift design in directionally drilled wells which will accurately account for the deviation.

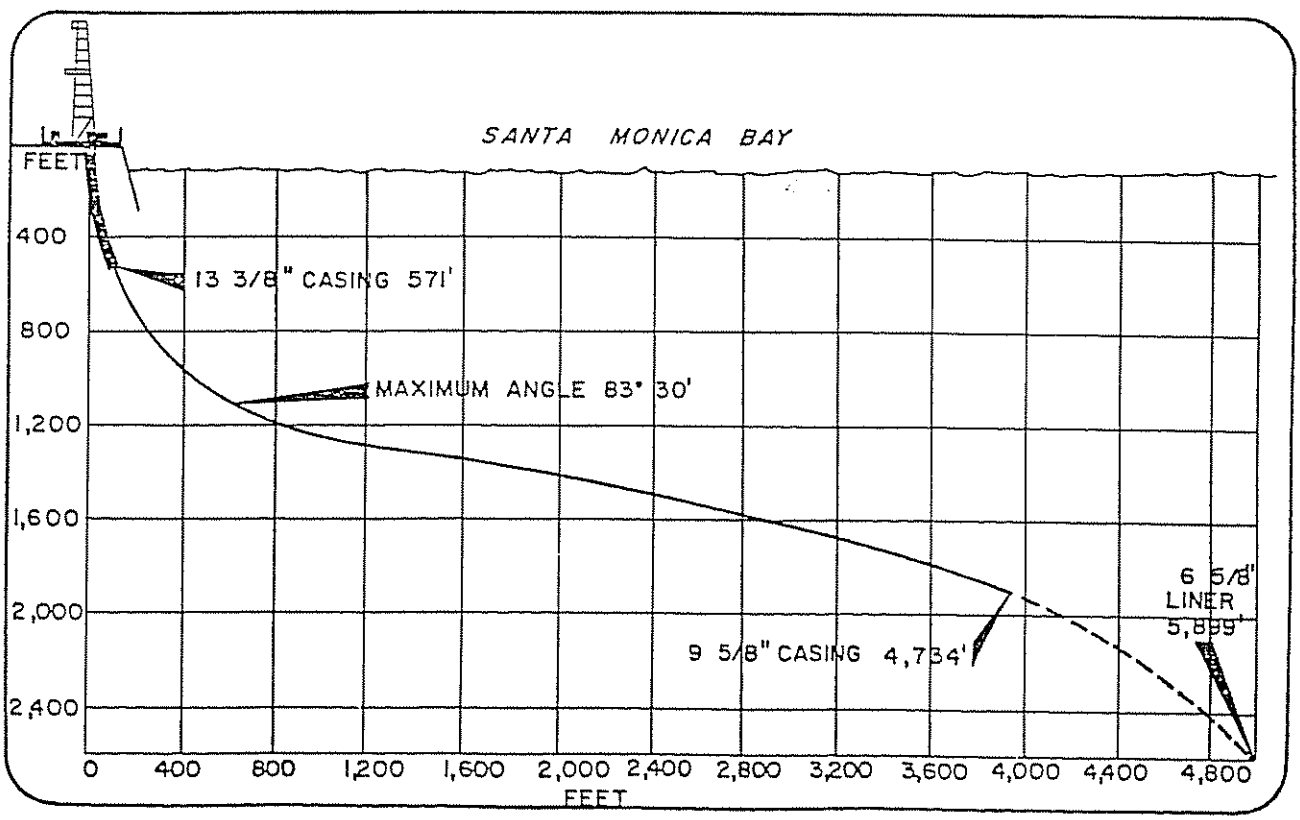


Fig. 3.493 Directional Well Profile

3.42102 Methods for calculating pressure loss in deviated wells

Brown and Beggs proposed two methods for handling the calculation of pressure losses in flow configurations for directionally drilled wells.

Use of standard multiphase flow correlations

In general, most correlations present an equation similar to the following:

$$\left(\frac{dp}{dz}\right)_{total} = \left(\frac{dp}{dz}\right)_{\substack{elev\ or \\ static}} + \left(\frac{dp}{dz}\right)_{friction} + \left(\frac{dp}{dz}\right)_{acceleration} \quad (3.432)$$

In more detailed terms this may appear as:

$$\frac{dp}{dz} = \frac{g}{g_c}\rho_m \sin\Theta + \frac{f\rho_m v_m^2}{2g_c d} + \frac{\rho_m v_m dv_m}{g_c dz} \quad (3.433)$$

where:

Θ = angle from horizontal (refer to Fig. 3.494)

If preferred, the angle may be taken from the vertical and sin Θ changes to cos ϕ as noted in Figure 3.495. This type of calculation permits the static or elevation component to be taken over the vertical distance and permits friction and acceleration to be taken over the entire length of the pipe.

The mixture density is calculated by the following equation:

$$\rho_m = \rho_L H_L + \rho_g(1 - H_L) \quad (3.434)$$

where:

ρ_m = density of the mixture of liquid and gas
H_L = holdup for vertical flow correlation (that fraction of the flow configuration occupied by liquid)
ρ_g = density of gas
ρ_L = density of liquid

A fallacy in this method for calculating pressure loss in deviated wells is that it assumes the holdup at any angle is equal to the holdup for a vertical well. However, it does account for friction and acceleration over the entire tubing length. The holdup can, in fact, change considerably depending upon the deviation angle and flow can become very inefficient at angles between 45-75°.

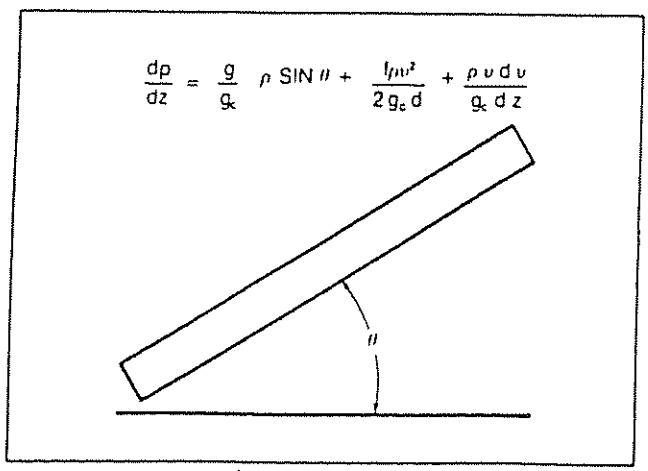


Fig. 3.494 Flow Geometry for Inclined Flow

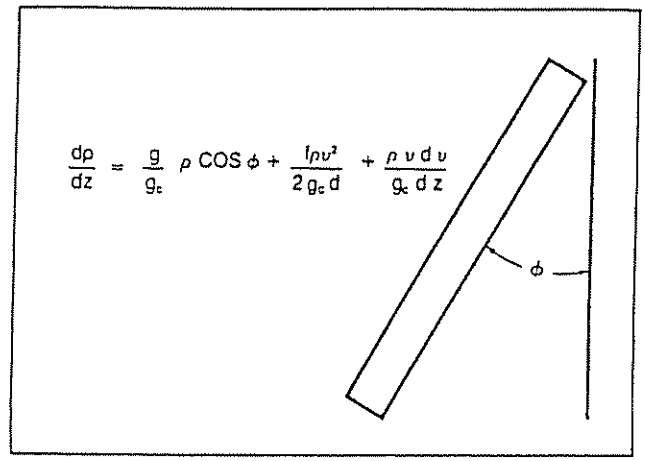


Fig. 3.495 Flow Geometry for Directional Flow

Beggs and Brill correlation

One of the few correlations that appears to be of practical field use for calculating pressure loss in deviated wells was presented by Beggs and Brill.[26] Beggs conducted numerous tests in which he measured H_L at different angles of flow for varying rates of liquid and gas.

He proposed the following equation for the pressure gradient:

$$\frac{dp}{dz} = \frac{\frac{g}{g_c}\rho_m \sin\Theta + \frac{f_{tp} G_m v_m}{2 g_c d}}{1 - \frac{\rho_m v_m v_{sg}}{g_c P}} \quad (3.435)$$

where:

$G_m = G_L + G_g$ = mass flux rate

In this method the liquid holdup used in calculating ρ_m is a function of the angle of the pipe from the horizontal.

Comparison of correlations

Several computer runs were made to show the difference in results obtained depending upon whether the Beggs and Brill correlation was used or whether one of the vertical correlations such as Hagedorn and Brown,[4] Orkiszewski,[27] or Duns and Ros[5] was used. The Beggs and Brill method is the only one of these that shows a variation in the holdup with angle. The other three merely use sin Θ in the static gradient term to correct it to vertical distance and determine friction and acceleration over the entire length of the pipe. Table 3.479 shows flowing pressures for a well having a true vertical depth of 15,000 ft. Flowing bottom-hole pressures for wells at angles of 0, 15, 30, 45, 60 and 75° from the vertical are shown for 2½" tubing for oil production rates of 300, 600, 1000 and 1500 B/D and for gas-oil ratios of 300 and 1000 scf/B. Figure 3.496 illustrates the effect of deviation angle on the calculated pressure traverse as a function of gas-oil ratio.

TABLE 3.479
CALCULATED PRESSURES AT TRUE VERTICAL DEPTH = 15,000 ft

q_o, b/d	300		600		1000		1500	
GOR	300	1000	300	1000	300	1000	300	1000
Angle				Oriszewski				
0	4462	1667	3999	1591	3845	2015	4139	2535
15	4463	1674	4003	1602	3853	2032	4152	2560
30	4463	1695	3986	1637	3883	2088	4200	2641
45	4473	1745	4038	1714	3929	2201	4275	2791
60	4493	1843	4082	1842	4116	2396	4435	3094
75	4542	2117	4224	2214	4291	2877	5025	3772
				Beggs-Brill				
0	4622	2713	4588	2711	4540	2884	4788	3185
15	4740	2931	4672	2835	4600	2944	4813	3213
30	4768	2990	4697	2888	4630	3002	4846	3296
45	4773	3007	4712	2940	4666	3099	4902	3456
60	4768	3019	4735	3031	4736	3285	4975	3775
75	4717	2989	4778	3237	4923	3755	5349	4541
				Hagedorn-Brown				
0	3979	1563	4091	1797	4250	2167	4478	2679
15	3980	1565	4093	1804	4256	2182	4489	2705
30	3985	1572	4103	1827	4276	2232	4526	2790
45	3989	1586	4121	1874	4319	2335	4602	2965
60	4001	1618	4161	1877	4414	2550	4765	3324
75	4039	1716	4284	2275	4673	3140	5227	4229
				Duns-Ros				
0	4770	2931	4621	2315	4525	2200	4609	2506
15	4771	2933	4623	2322	4531	2217	4621	2535
30	4771	2942	4630	2347	4549	2274	4658	2631
45	4775	2948	4642	2389	4641	2405	4784	2839
60	4783	2964	4670	2515	4716	2469	4941	3230
75	4805	3025	4759	2774	4944	3237	5383	4184

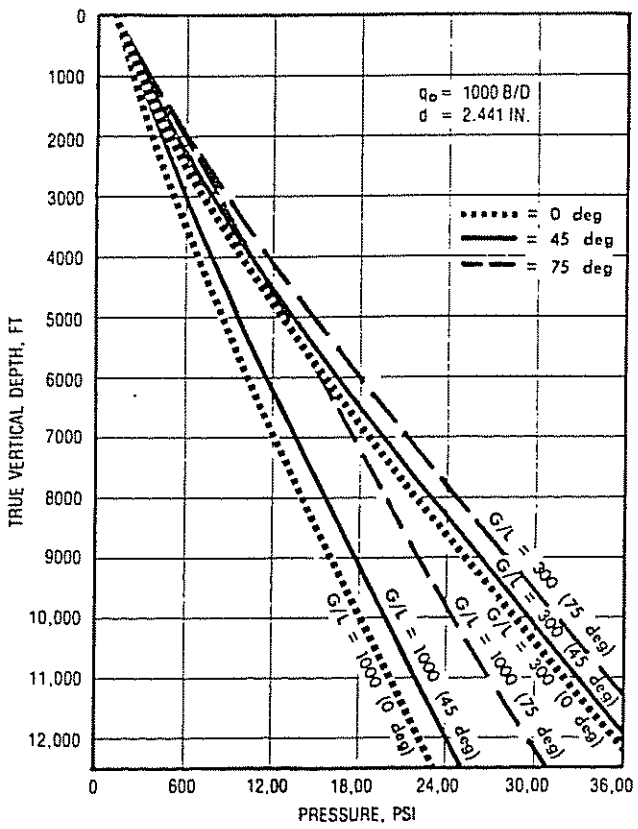


Fig. 3.496 Comparison of Projected Vertical Pressure Traverses (Courtesy Petroleum Engineer)

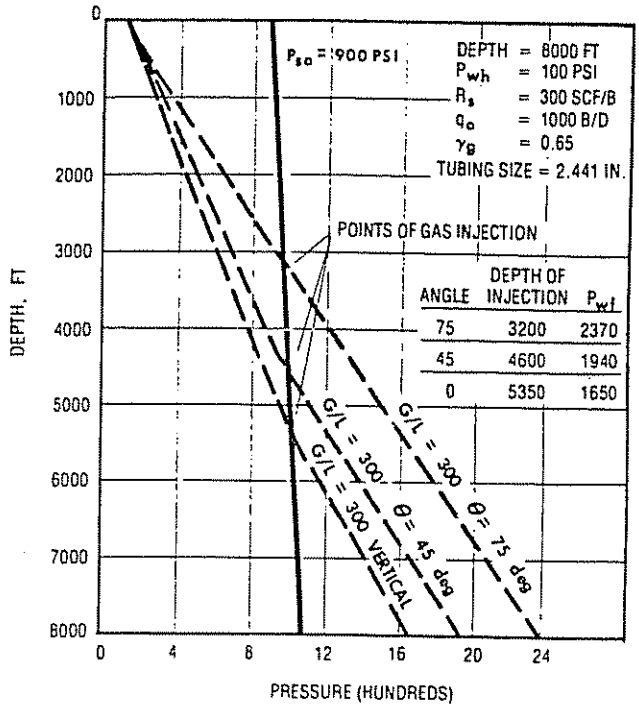


Fig. 3.497 Comparisons of Flowing Pressures Possible by Gas Lift (Courtesy Petroleum Engineer)

3.42103 Design of gas lift installations

The following procedure is proposed for the design of a gas lift installation for a directional well:

(1) Determine the vertical and measured tubing lengths along with the angle of deviation.
(2) Calculate the pressure traverse in the directionally drilled well and transpose these pressures to equivalent vertical depths.
(3) Design the gas lift installation in the normal manner using the pressure traverses as calculated in Step 2.
(4) Spacing may proceed in a normal manner.

3.42104 Example problems and discussion

To illustrate the effect of deviation angle, problems are worked to illustrate possible flow rates, required injection gas volumes, attainable and flowing pressures.

Figure 3.497 gives an example problem illustrating the possible flowing pressures to obtain for the same well conditions for angles of 0, 45 and 75°. Also, note the difference in the points of gas injection. Table 3.480 summarizes these results:

TABLE 3.480
THE DEPTH OF INJECTION VS ANGLE OF DEVIATION

Angle of deviation	Depth of injection, ft	Flowing bottom-hole pressure, psig
75	3200	2370
45	4600	1940
0	5350	1650

Table 3.480 shows that as the angle of deviation increases, the depth to the point of gas injection decreases, and the flowing bottom-hole pressure increases for the same rate. In this example, the point of injection changes from 3200 feet for a 75° angle well to 5350 feet for a vertical well. The flowing pressure changes from 1650 psi for a vertical well to 2370 psi for a 75° angle well.

If the same rate is desired from a vertical well and a deviated well having the same characteristics, then the deviated well will require more gas to maintain the same flowing bottom-hole pressure. Otherwise, the design will proceed as for a vertical well. This is illustrated in the example problem of Figure 3.498 whereby the points of injection and injection gas-liquid ratios have been determined for a vertical and a 45° deviated well. The following Table 3.481 summarizes this information.

TABLE 3.481

Type well	Point of injection	Rate, B/D	P_{wf}, psig	Total G/L	Inj. G/L	Inj. gas vol. MCF/DAY
Vertical	4500	1000	2400	520	220	220
45° Angle	4680	1000	2400	1000	700	700

To obtain the same rate, inject more gas into the deviated well (700 MCF/D as compared to 220 MCF/D). The rate of 1000 B/D cannot be obtained in a well of 75° angle for a static reservoir pressure of 2800 psig. Figure 3.499 shows the flow rates possible by gas lift from directional wells with deviations of 75° and 45°

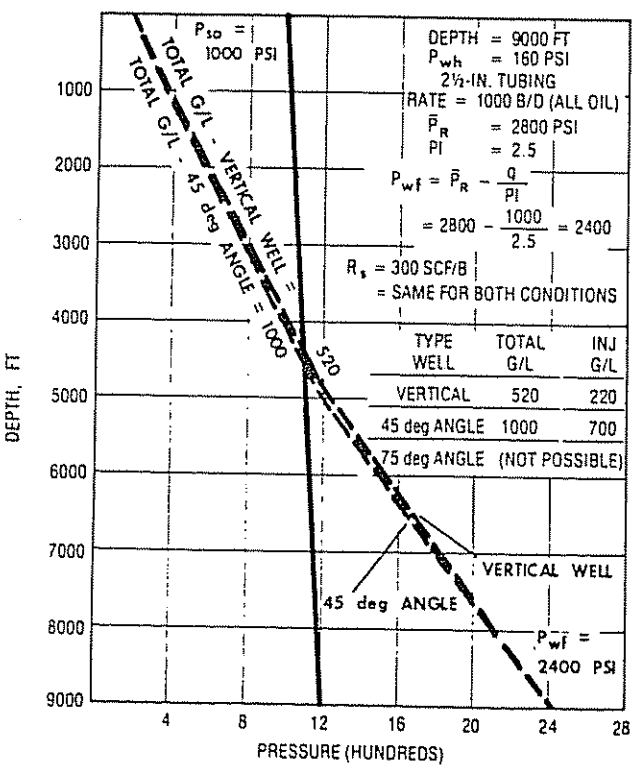


Fig. 3.498 Comparison of Injection Gas Liquid Ratios for Same Rate from Vertical and Deviated Wells (Courtesy Petroleum Engineer)

compared to an 8000 ft vertical well. These wells use the same injection gas pressure of 900 psig at the surface ($\gamma_g = 0.65$). The additional data is given on Figure 3.499. PI's of 1, 2, 4, and 8 are included and are assumed to be constant with drawdown. The following Table 3.482 shows the flow rates possible from this well depending upon the angle of deviation. (All have 8000 ft vertical depths).

These data show that the flow rates possible by gas lift decrease as the deviation angle increases. For the same conditions—that is, for 2½" tubing, PI = 2, an 8000 ft vertical depth—the vertical well will produce 1500 B/D as compared to 1020 B/D for a well deviated at 75° and 1320 B/D compared to a well drilled at an angle of 45°. On the average, the decrease in rate for a 75° angle well compared to a vertical well is approximately 500 B/D for all PI's (for this example).

For the more common deviation of 45°, the decrease in rate is approximately 200 B/D, being 125, 180, 190, and 190 B/D for PI's of 1, 2, 4, and 8, respectively.

The points of gas injection will change for each angle; for the same operating pressure these points will be higher for the higher deviations.

The rates in Table 3.482 are determined using an injection gas-liquid ratio of 700 scf/B which gives a total gas-liquid ratio of 1000 scf/B. This is nearly enough gas to obtain the minimum flowing gradient and, hence, maximum production.

TABLE 3.482
RATES POSSIBLE, B/D

Angle	PI = 1	PI = 2	PI = 4	PI = 8	Tubing length, ft
0	1095	1500	1720	1830	8,000
45	970	1320	1530	1640	11,314
75	720	1020	1200	1280	30,910

3.42105 Summary

The design of gas lift installations in directionally-drilled wells can be accomplished by projecting the pressure traverses of the deviated well to equivalent vertical depths. Once this has been done, the design proceeds in the normal manner as for a vertical well.

The maximum production rate possible from a deviated well will be less than for a vertical well due to additional pressure loss at the same operating conditions.

Once the projected equivalent vertical pressures are determined, the spacing of the valves can proceed in a normal manner and the proper valve can be selected. The only differences in valve design will be in port size selection, due to larger gas volume requirements of the directional wells for the same flow rate.

To obtain the same rate from a highly deviated well, increase either the volume of injected gas or operating gas pressure, or both.

Where possible, use tubing flow because of the inherent additional instabilities encountered in annular flow for directional wells.

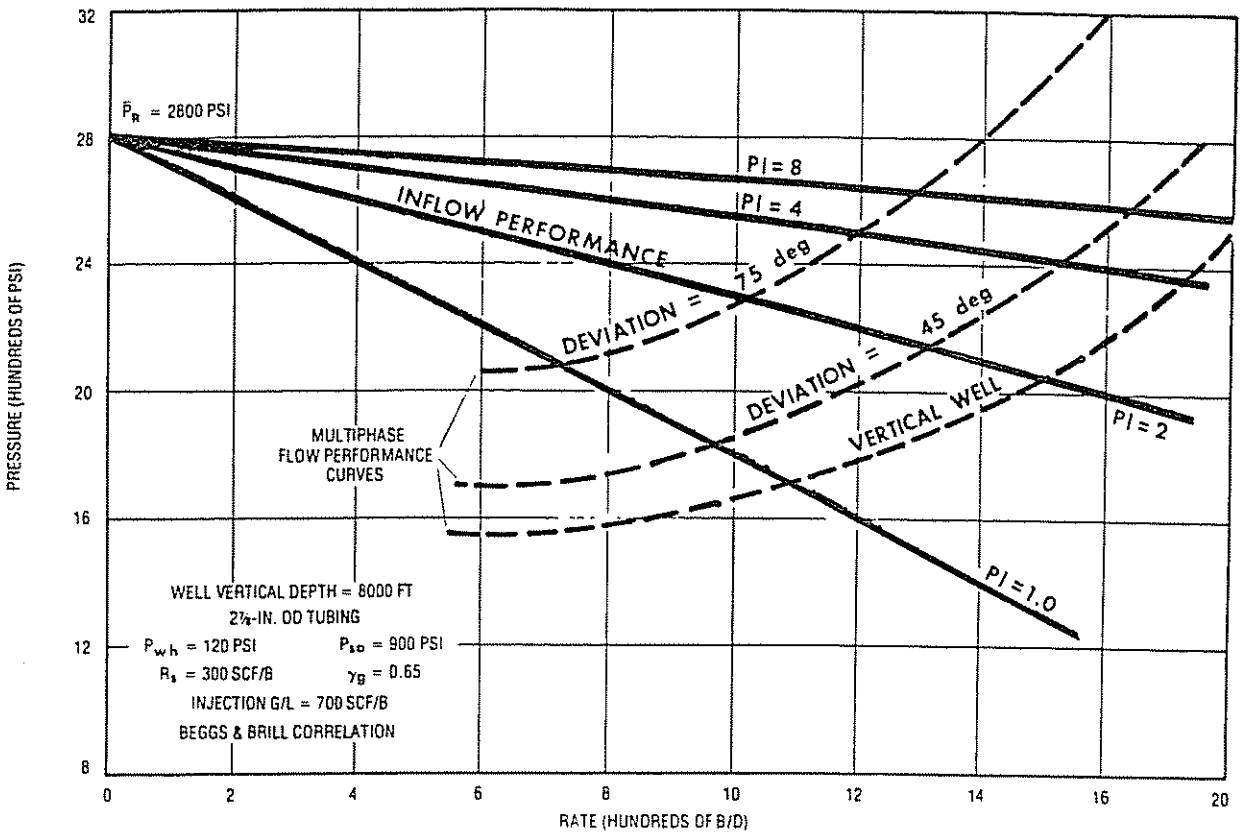


Fig. 3.499 Production Rates Possible for Various Deviation Angles (Courtesy Petroleum Engineer)

3.4211 Summary and logical sequence in continuous flow gas lift design

The following summarizes the recommended procedure for designing a continuous flow gas lift installation. To utilize this procedure, it is necessary to have good well information including inflow performance curves for the well, water cut, gas-oil ratio, etc.

Procedure:

(1) Prepare intake curves for several values of "ΔP". That is, plot rate on the ordinate vs. "ΔP" between gas injection pressure and flowing tubing pressure at the operating gas lift valve on the abscissa. A typical plot is noted in Figure 3.476(B), and several PI's can be observed on the same plot. The rate increases as ΔP decreases. For the higher productivity wells, this rate increase is more pronounced than for the lower productivity wells.

(2) Select a ΔP based on rate, spacing, PI, cost, and good judgement. For the higher productivity wells, the lowest ΔP practical should be selected. From a practical standpoint, "ΔP's" less than 50-75 psi dictate valve spacings approaching minimum values. For example, ΔP = 50 and G_s = 0.40 calls for a spacing of 50 psi/0.40 psi/ft = 125 ft. Valves should not be closer than three joints (90 feet) apart although this close spacing is used after flowing pressure surveys locate the exact point of injection.

With valves three joints apart, both above and below the point of injection, two to three valves will probably pass gas. This is not critical since the upper valve of three will be passing little gas since it is almost closed. The center valve will be passing most of the gas and the lower valve will be passing very little gas due to an extremely low differential across it. Without good well information, the selection of a 100 psi "ΔP" is a good choice.

(3) Design the gas lift installation using the graphical procedure and accounting for the separator pressure, wellhead flowing pressure, surface operating pressure, etc.

(4) Draw in the objective gradient line for the predetermined rate and injection gas volume, using ΔP of Step 2 to locate the point of injection.

(5) Construct the bracketing envelope around the point of gas injection.

(6) Locate the first valve which is the same for all types of valves. Beginning with the separator pressure at the top, construct the kill fluid gradient line and extend it downwards until reaching the kick-off pressure line. Move back 50 psi and note this as position of Valve 1.

(7) Select a transfer point for each succeeding valve which controls the spacing of valves. For pressure valves, this may be a point directly on the objective gradient line. For fluid valves refer to the section on fluid valve design. The transfer point should always be to the right of the objective gradient line.

(8) Space valves downwards to the top of the error envelope and then use $\frac{\Delta P}{G_s}$ to space valves to the bottom of the error envelope.

3.4212 CLASS PROBLEMS
CONTINUOUS FLOW DESIGN

(1) Given:

Tubing = 2⅜" O.D.
$\gamma_g = 0.65$
$G_s = 0.45$ psi/ft
G/O = 200 scf/b
Produces 100% oil
$P_R = 2250$ psi
PI = 3.5
Rate desired = Maximum
Temp bottom = 180°F.
Geothermal gradient = 1.4°F/100 ft
$P_{wh} = 150$ psi
$P_{so} = 900$ psi
Depth = 8000 ft

Find:

(a) Flow rate possible by gas lift.
(b) Space and design gas lift valves:

(1) Standard pressure valves taking 25 psi drop in surface operating pressure between valves
(2) Fluid operated valves—Pressure charged dome
(3) Fluid operated valves—Spring charged dome
(4) Combination pressure closed fluid opened valves
(5) Analytically taking 25 psi drop between valves—standard pressure charged valves
(6) Standard pressure operated valves—Constant surface opening pressure
(7) Balanced valves set at 80°F in the shop
(8) Using response system

(2) Given:

Depth = 8000 ft
$P_{wh} = 100$ psi
$P_R = 2850$ psi
GOR = 300 scf/b
$G_s = 0.50$ psi/ft
(Loaded to surface)
Temp Bottom = 160°F.
Temp Top = 120°F.
$\gamma_g = 0.65$
$q_L = 1000$ B/D (50% oil)
PI = 2.5
2⅜" O.D. tubing

Find:

(a) Space gas lift valves using universal spacing for standard pressure operated valve, fluid operated valves, and combination valves.
(b) Prepare a table showing shop settings of all valves.

(3) Given:

2½" tubing
$\gamma_g = 0.8$
50% oil
PI = 4
$P_{so} = 900$ psi
Depth = 8000 ft
$q_L = 600$ B/D
GOR = 200 scf/B
$P_R = 1900$ psi
$P_{wh} = 100$ psi
$P_{ko} = 950$ psi

Space and calibrate valves:

(1) Pressure operated valves taking 25 psi drop between valves
(2) Fluid operated valves—spring charged and bellows charged.
(3) Spring charged pressure operated valves with constant surface opening pressure
(4) Combination pressure closed—fluid opened valves

(4) Given—Continuous Flow:
4" tubing
$q = 4000$ B/D (all oil)
$P_R = 2800$ psi
$\gamma_g = 0.7$
Depth = 9200 ft
Temp Top = 120°F.
Temp Bottom = 192°F.
$G_s = 0.40$ psi/ft
(Loaded to Surface)
$P_{wh} = 120$ psi
$P_{so} = 1000$ psi
PI = 6
$R_s = 200$ scf/b

Space gas lift valves and prepare complete design for:

(1) Standard continuous flow pressure operated valves (Constant P_{so})
(2) Fluid operated valves
(3) Combination fluid opened pressure closed valves

(5) Given—Continuous Flow:
2½" tubing
$q_L = 1000$ B/D (90% salt water)
$P_{wh} = 100$ psi
PI = 1.0
Kill fluid = 0.50 psi/ft
Depth = 8000 ft
$P_{so} = 900$ psi
$P_R = 3300$ psi
$\gamma_g = 0.65$
GOR = 400 scf/B
Temp Surface = 130°F.
Temp Bottom = 200°F.

Space valves and design completely for:

(1) Pressure valves
(2) Fluid valves

(6) Given—Continuous Flow—Annular Flow:
2⅞" O.D. tubing × 7" (23#/ft) casing
$P_{wh} = 100$ psi (all oil)
$P_R = 3100$ psi
$\gamma_g = 0.8$
Depth = 8500 ft.
Rate desired = maximum
PI = 15
$P_{so} = 1200$ psi
$R_s = 200$ scf/B

Space gas lift valves and design completely for:

(1) Pressure operated valves
(2) Fluid operated valves

(7) Given—Continuous Flow:

2⅜" O.D. tubing
q_o = 1000 B/D (50% water)
P_{wh} = 100 psi
R_s = 300 scf/B
PI = 2.6
Temp surface = 120°F.
Temp buttom = 180°F.
P_R = 3000 psi
Depth = 8500 ft
G_s = 0.50 psi/ft
γ_g = 0.65
Design a complete installation for:

(a) Pressure charged valves
(b) Fluid operated valves

(8) Given data—Continuous Flow:

P_{so} = 1000 psi
2½" tubing
γ_s = 0.7
GOR = 200 scf/B
PI = 4
P_R = 2700 psi
Depth = 8000 ft
q_o = 2000 B/D (all oil)
Temp Top = 130°F.
Temp Bottom = 170°F.
Kill fluid = 0.50 psi/ft

Design a complete installation, finding injection depth, size of valve ports, gas volumes—locate all valves.

(9) Given data—Continuous flow:

Little information is available

Depth = 8200 ft
P_{so} = 900 psi
P_{wh} = 120 psi
2½" tubing
P_R = unknown
PI = unknown
Rate = unknown (Estimate between 1000 and 2000 B/d)

Design an installation and space valves for pressure operated valves.

(10) Approximately how deep can you continuous flow a well with an operating pressure of 1000 psig and a flowing tubing pressure of 150 psig (2 in. tubing size)?

(11) Why should continuous flow valves throttle?

(12) A well has a static fluid level at 3200 ft and the fluid is standing at that point (γ_w=1.07). Surface kick-off pressure for a 0.65 gas is 900 psig. How deep can the first valve be placed with 2 in. tubing in 5½ in., 17 lb/ft casing? Would you recommend placing the first valve at the static fluid level? Why?

(13) Why should the normal geothermal temp gradient as well as the flowing temp gradient be considered in a continuous flow installation?

(14) Why must balanced valves take a drop in surface opening pressure as we work down the hole? Why can unbalanced valves be set for the same surface opening or closing pressure?

(15) Why should communication be provided between tubing and casing when running differential valves?

(16) A balanced gas lift valve is charged with a 0.60 sp gr gas to 700 psi at 80°F. It is placed in an 8000 ft well at bottom which has a BHT of 200°F. Now what is its opening pressure opposite the valve? Its surface opening pressure is how much if the injection gas sp gr is 0.65?

(17) A balanced (no tubing effect) valve string is run to 7000 ft in a continuous well with the following spacing. The temp gradient is 2°F per 100 ft using a mean surface temp of 74°F.

Valve setting at 80°F (psig)	Depth of valve (ft)
800	2000
750	3500
700	4700
650	5700
600	6500
550	7000

The injection gas has a sp gr of 0.65. Calculate the surface opening pressure of each valve.

(18) A well 7600 ft deep is to be placed on gas lift. The flowing gradient below the point of gas injection is 0.45 psi/ft, the flowing BHP is 2000 psi, and the surface injection pressure is 875 psi. Temperatures are normal and the gas sp. gr. is 0.65. The flowing tubing pressure is 125 psi. Solve for the point of gas injection utilizing maximum casing pressure and check your solution graphically.

(19) How close would you allow valves to be spaced on high PI continuous flow wells? Explain.

(20) A 7200 ft well has a static BHP of 2800 psia, and a PI of 2. The well is to be placed on gas lift using 2⅞ in. OD tubing. The flowing gradient above the point of gas injection will be 0.15 psi/ft, while the flowing gradient below the point of gas injection is to be 0.45 psi/ft. The expected flowing tubing pressure is 160 psi. It is desirable to remove 1200 B/D of fluid.
Design a gas lift installation for this well showing the point of gas injection, the gas lift valve spacing, and the injection gas pressure needed. Use tapered stem ¼ in. ported valves with same surface opening pressures.

(21) A well is making 500 BLPD and has a PI of 1. It is 8000 ft deep. The flowing gradient above the point of gas injection is 0.15 psi/ft and the flowing gradient below the point of gas injection is 0.45 psi/ft. If 800 psi injection pressure is available, design a string of differential valves (125 psi) for this well. Place two valves below the required point of injection. Surface tubing pressure is 100 psi. What will the injection pressure in the casing at the point of injection be? The injection gas has a sp gr of 0.70. Design a string of balanced gas lift valves for this well (charge at 80°F).

(22) A well is to produce 400 BLPD from 2⅞ in. OD

tubing (2½ in. nominal). It has a depth of 6000 ft, a PI of 1.0, and a static BHP of 2500 psi. Using balanced gas lift valves, design an installation to continuous flow this well, determining the point of gas injection, the injection gas pressure needed, and the valve spacing. P_{wh} = 120 psi. G/L = 200 scf/B.

(23) Given:

Depth = 8500 ft
$\overline{P}_R$ = 2600 psig
PI = 1.5
P_{wh} = 150 psig
Required rate = 600 b/d (90% salt water)
Tubing size = 2 in. nominal
GOR = 1000 scf/B
γ_g = 0.70

Find: (a) Point of injection for surface operating pressures of 400, 800, 1200, and 1500 psig.
(b) Injection GLR to produce the well from each depth found in (a).

(24) An 8000 ft well is equipped with 2⅞ in. OD casing (2.441 in. ID) and is expected to produce 80 BOPD with a WOR of 4. It has a BHP of 2600 psi and a PI of 5. The BHT is 180°F and the expected flowing surface temp is 120°F. The surface flowing tubing pressure is 100 psi. The kick-off pressure is 900 psi and the operating pressure is 800 psi (gas sp gr = 0.65).
(a) Design a string of balanced continuous flow valves for this well if 1½ in. tubing is run in the well.
(b) Design a string of tapered stem bellows valves for this well.

(25) A 7000 ft well is expected to be placed on continuous flow. The following information is known:

Depth to top of packer = 6950 ft
P_{wh} = 150 psig
P_{ko} = 800 psig
P_{so} = 700 psig
$\overline{P}_R$ = 2600 psig
J = 5
2⅜ in. tubing in 7 in., 23 lb/ft casing
(γ_g = 0.65)
Surface temp = 130°F
Bottom temp = 200°F
Rate = 1000 BLPD
Well makes 95% salt water and 5% oil with a solution GOR of 500 scf/STBO.

(a) Determine the point of gas injection.
(b) Design a string of unbalanced variable orifice continuous flow valves.
(c) Design a string of balanced valves.
(d) Check all graphical solutions by an analytical solution.
(e) Design this installation for each case of balanced, unbalanced, and fluid pilot, fluid-operated valves.

(26) Space gas lift valves for the following continuous flow well.
Given:

Depth = 10,000 ft
P_{wh} = 100 psi
P_{ko} = 1000 psig
$\overline{P}_R$ = 2800 psi
P_{so} = 900 psig (operating)
PI = 5
Desired rate = 800 B/D (90% salt water)
Gas sp gr = 0.65
GOR = 400 scf/B
Tubing size = 2⅞" O.D.

(a) Differential valves
(1) Graphically
(2) Analytically (neglect gas wt)
(b) Balanced pressure-operated valves, allowing 25 psi drop per valve for each valve, starting with kick-off pressure
(1) Graphically, based on minimum gradient line
(2) Graphically, based on flowing gradient line
(3) Analytically, based on unloading gradient equivalent to average flowing gradient above injection point.
(c) Unbalanced valves, all set for the same 900 psig surface operating pressure
(1) Graphically
(2) Analytically

Assume well is loaded to the surface in each case with a kill fluid of 0.50 psi/ft

3.43 Design of intermittent flow installations

3.431 Introduction

The design of a good intermittent installation may be more difficult than the design of a continuous flow installation. In intermittent lift a liquid slug is allowed to accumulate above the gas lift valve. The valve then opens, allowing sufficient gas to enter and propel the slug to the surface in piston form.

The purpose of intermittent lift is the same as continuous flow—to create the necessary BHP drawdown so the well will produce the desired fluid rate.

The types of installations used for intermittent lift have been discussed in Section 3.3. These installations must be designed for very low BHP wells in many cases. This may necessitate the use of chambers.

The following production rates were given by Thrash and Brown as a guide as to when to consider changing from intermittent flow to continuous flow:[28]

TABLE 3.483

Tubing size (in.)	Rate (B/D)
1	25-50
1¼	50-75
1½	75-125
2	200
2½	250
3	300
Chambers	400-600

To understand intermittent gas lift, the following discussion was given by Brown.[22]

The vertical flow of liquid slugs is more complex than either multiphase continuous vertical or horizontal flow. The unsteady state nature of liquid slug flow has discouraged attempts to describe the phenomenon analytically.

Liquid slug flow normally occurs only when induced by intermittent gas lift. However, it may occur in the

latter stages of the life of a flowing well. It may also be induced in a flowing well by what is known as *stopcocking*. The material in this secion deals specifically with intermittent gas lift. However, there is no reason why the theory involved cannot be applied to the remaining cases.

3.432 Intermittent gas lift cycle

Intermittent gas lift is a cyclic production method in which a liquid slug is first allowed to build up in the tubing string (Fig. 3.4100A). (When the combination of surface back pressure, weight of gas column, and hydrostatic pressure of the slug reaches a specified value at the gas lift valve, gas is injected into the casing annulus through some type of control at the surface for a definite injection time.)

When the casing pressure increases to the opening pressure of the gas lift valve, gas is injected into the tubing string. Under ideal conditions the liquid, in the form of a slug or piston, is propelled upwards by the energy of the expanding and flowing gas beneath it. The gas travels at an apparent velocity greater than the liquid slug velocity, resulting in penetration of the slug by the gas. This penetration causes part of the liquid slug to fall back into the gas phase in the form of droplets (Fig. 3.4100B) and/or as a film on the tubing wall (Fig. 3.4100C).

When the liquid slug is produced at the surface, the tubing pressure at the valve decreases, increasing the gas injection through the valve. When the casing pressure drops to the valve closing pressure, gas injection ceases (Fig. 3.4100D). Following production of the slug, a stabilization time occurs during which the fallback from the previous slug falls or flows to the bottom of the well and becomes a part of the next slug, which is feeding in from the producing zone.

Liquid fallback can represent a substantial part of the original slug. Control of fallback determines the success of an intermittent gas lift installation. The inability to predict liquid fallback has resulted in overdesign of many installations. In may cases high recovery rates are achieved, but frequently at excessive operational costs which limit the profit-making ability of the wells.

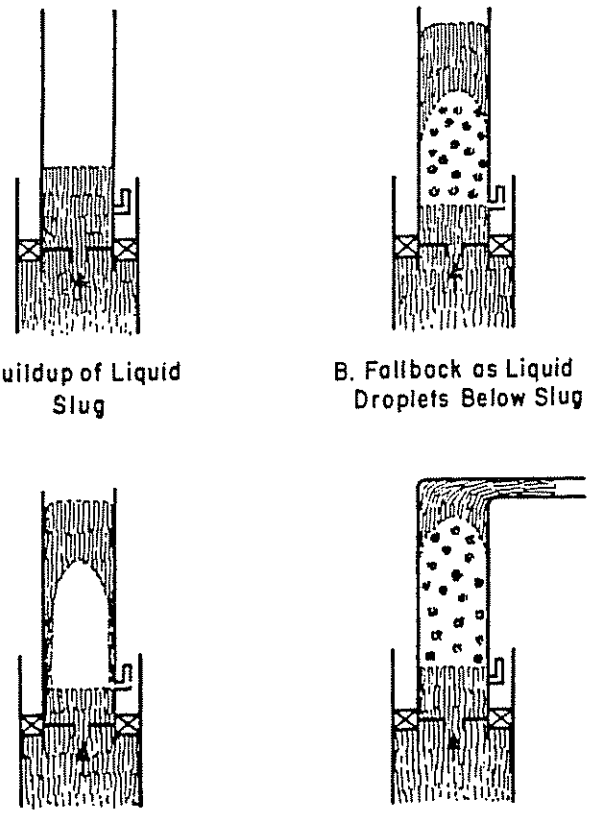


Fig. 3.4100 The Intermittent Gas Lift Cycle

3.433 Analysis of pressure recordings during intermittent gas lift cycles

Varying pressures at depths beneath the top of the moving liquid slug occur in intermittent gas lift. Numerous pressure recordings are presented for gas lift installations in both 2 and 1¼" nominal size tubing. Each recording represents one cycle, and each curve within a recording represents the pressure behavior measured at the indicated depth. Figure 3.4101 shows a typical recording in which gas was injected into 2" tubing at a depth of 5940 feet through a 1" ported gas lift valve. Pressure recordings are illustrated at depths of 5936, 4290, 2493, 1685, 967, 477, and 0 feet. The initial tubing pressure at the valve was 350 psi, and the initial slug volume was 2.345 B(95% salt water).

The following information can be obtained from Figure 3.4101. From the pressure recording at the valve at 5936 feet:

(1) At zero time the initial tubing load was 350 psi.
(2) As gas was injected, the slug accelerated until the tubing pressure at the valve reached 600 psi within 2-3 minutes.
(3) The slug reached the surface in 4 minutes, 35 seconds as noted on the zero depth curve (surface tubing pressure). The pressure at 5936 feet began to decrease at this time, although the gas lift valve had not yet closed.
(4) As the slug was produced at the surface, the pressure at 5936 feet dropped to approximately 530 psi in 8 minutes, at which time the gas lift valve closed.
(5) The pressure then dropped sharply to a minimum of 208 psi in about 12 minutes. The minimum pressure represents a combination of well back pressure, liquid fallback, and fluid feed-in into the wellbore from the producing formation.
(6) The minimum pressure remained fairly constant for 4-5 minutes during which time the fluid in droplet form was still being produced at the surface, tending to reduce the pressure at 5936 feet. However, liquid fallback and feed-in offset the pressure reduction, resulting in the constant pressure. The shape of the curve depicting minimum pressure before the pressure builds up varies, depending upon the rate of liquid production (82 B/D for this well).
(7) At approximately 18 minutes, the pressure at

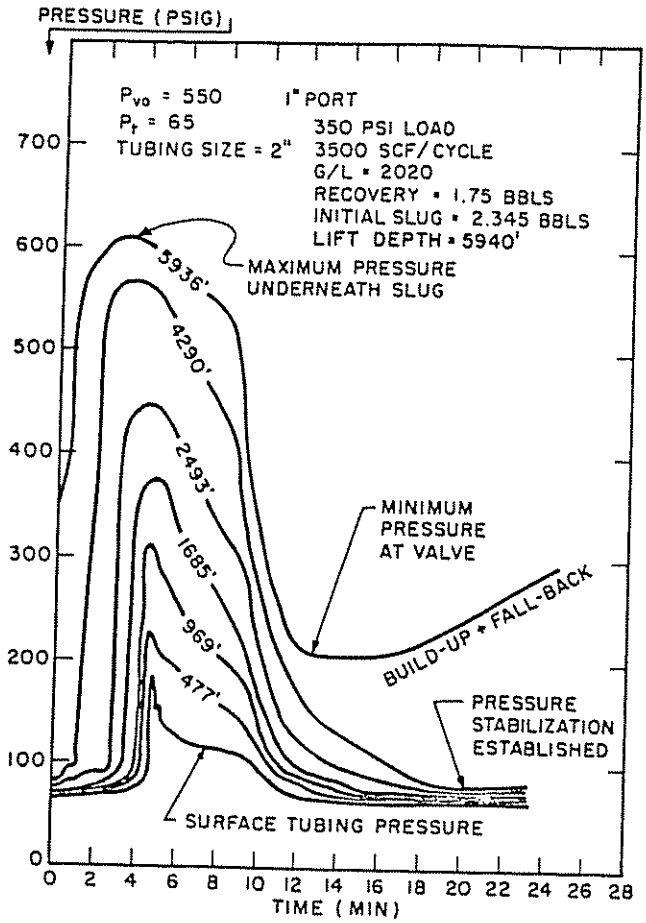


Fig. 3.4101 Typical Pressure Recordings for 1-in Port

5936 feet began to show a decided increase due to liquid feed-in.

From the pressure recording at 4290 feet, observations similar to those at 5936 feet can be made:

(1) The stabilized tubing pressure at zero time was 80 psi, indicating that the top of the slug was initially below this point.
(2) The top of the slug reached 4290 feet in about one minute, at which time the pressure began to increase.
(3) The pressure continued to increase as the slug passed 4290 feet, but did not reach the pressure level of 600 psi attained at 5936 feet. The lower maximum pressure of 575 psi is a result of part of the slug being lost as liquid fallback.
(4) The pressure dropped at about the same rate as on the 5936 feet recording when the slug was being produced at the surface and after the gas lift valve closed.
(5) After 12 minutes the pressure at 4290 feet continued to fall since liquid feed-in had not reached this level.
(6) After approximately 18 minutes, the pressure had again stabilized at 80 psi. This represents the time required for the fallback in the tubing to settle completely, and is important in determining the optimum physical cycle frequency.

The pressure recordings at 2493, 1685, 969, and 477 ft show the same general trends as do those at 4290 and 5936 feet. The times at which the slug reached these depths can easily be determined. After attaining their maximums, the pressures continually decreased, becoming constant after about 18 minutes. With decreasing depth, lower pressure maximums are discernible, indicating more liquid fallback.

The pressure curve at the surface (zero depth) shows the following:

(1) At zero time the tubing back pressure was 65 psi.
(2) The liquid reached the suface at 4 minutes, 35 seconds.
(3) The pressure reached a maximum of 195 psi.
(4) The pressure dropped immediately, indicating the major portion of the production had been recovered. Subsequent production in the form of droplets and finely-dispersed fluids follow production of the intact slug.
(5) The gas following the liquid slug (tail gas) had completely escaped within 14 minutes.

Figure 3.4102 shows another recording from 5940 feet when a 1" ported valve is used in lifting a 350 psi load. This particular recording is for the minimum gas-liquid ratio possible to attain for these conditions (2100 scf/cycle). As noted, only 55% of the slug was recovered.

Figure 3.4103 shows the same slug lifted with a 5/16" port. This port was too small and would not allow full

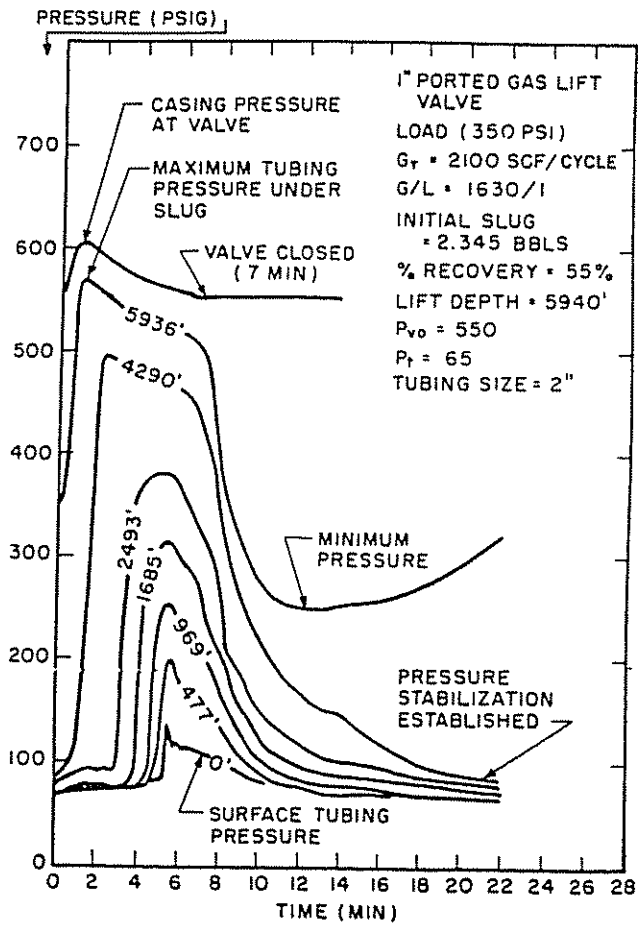


Fig. 3.4102 Typical Pressure Recordings for 1-in Port (Minimum GLR)

casing pressure to be placed underneath the slug. (3500 scf/cycle).

Figures 3.4104, 3.4105, and 3.4106 show recordings from 4072 feet with a 7/16" ported valve. The pressure placed underneath each slug was 620 psi and loads of 450 psi, 350 psi, and 250 psi were lifted, giving pressure differentials across the valve of 170, 270, and 370 psi, respectively. These recordings show that a result of lifting lighter loads is an increased velocity. The lighter load accelerates at a greater rate, preventing full casing pressure from being placed underneath the slug.

Figures 3.4107, 3.4108, and 3.4109 show recordings obtained in 2" tubing in which a 7/16 in. ported operating valve was located at 2774 feet. Approximately 660 psi pressure was available in the casing at the valve. For pressure loads of 550, 400, and 300 psi, this resulted in pressure differentials across the valves of 110, 260, and 360 psi, respectively. The lighter slugs represented in Figs. 3.4108 and 3.4109 accelerated so rapidly that maximum pressures of only 615 and 590 psi, respectively, were reached.

Figures 3.4110 through 3.4118 illustrate typical recordings in 1¼" tubing with the operating gas lift valve placed at 1450 feet. Three different pressure differentials across the valve are represented, and for each differential a graph is shown for conditions where:

(1) excess gas was injected (Figs. 3.4110, 3.4113, 3.4116)

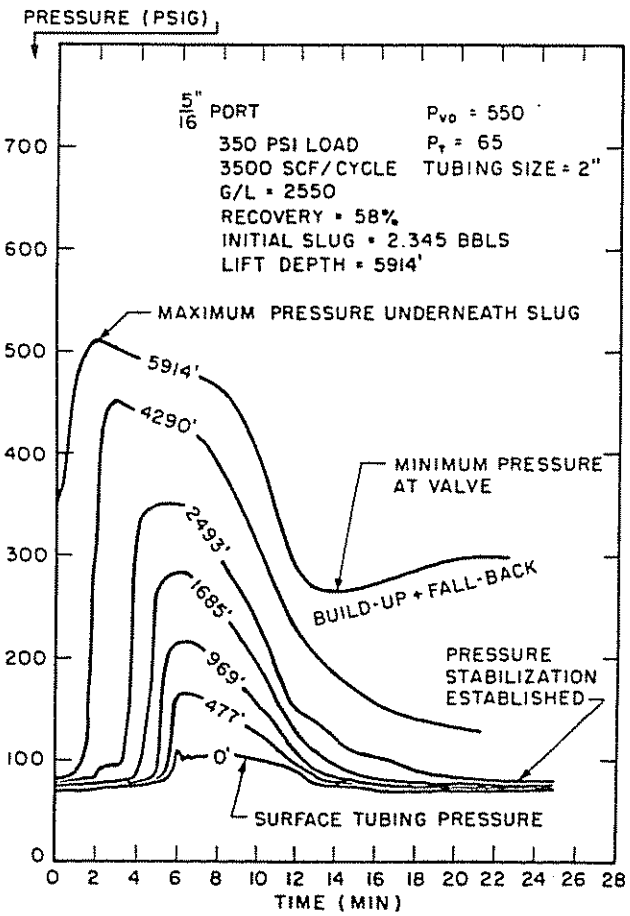


Fig. 3.4103 Typical Pressure Recordings for 5/16-in Port

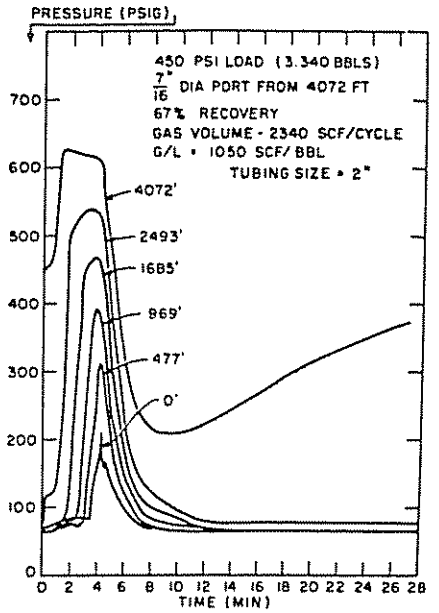


Fig. 3.4104 Typical Pressure Recordings from 4072 ft.

(2) minimum gas-liquid ratios were obtained (Figs. 3.4111, 3.4114, 3.4117)

(3) insufficient gas was injected (Figs. 3.4112, 3.4115, 3.4118).

Comparisons between graphs yield additional important information. For a given pressure differential, as the total gas injected at the surface decreases, the gas-liquid ratio decreased to a minimum (Figs. 3.4111, 3.4114, 3.4117) and then increases, although gas injection continues to decrease. This increase is because an insufficient amount of gas was injected to lift the slugs, and the small recoveries result in high gas-liquid ratios.

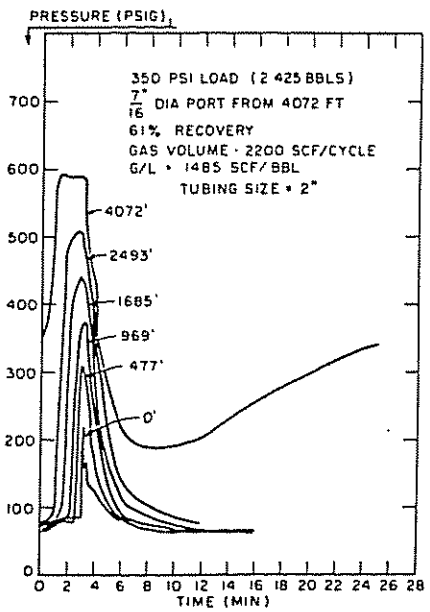


Fig. 3.4105 Typical Pressure Recordings from 4072 ft.

The effect of pressure differential and amount of gas injected on slug velocity can be seen qualitatively from the graphs. For a given differential, the time for the slug to reach the surface is seen to increase with decreasing total gas volume injected at the surface. Comparing figures for different initial pressure differentials and identical gas injection conditions, the time for slug clearance decreases with increasing initial differential. The trends can be observed in the pressure recordings at all depths.

3.434 Factors to consider in the design of an intermittent installation

The following factors are discussed in detail to help the reader design a good intermittent flow installation.

3.4341 Type of installations

Three types of intermitting installation may be considered:

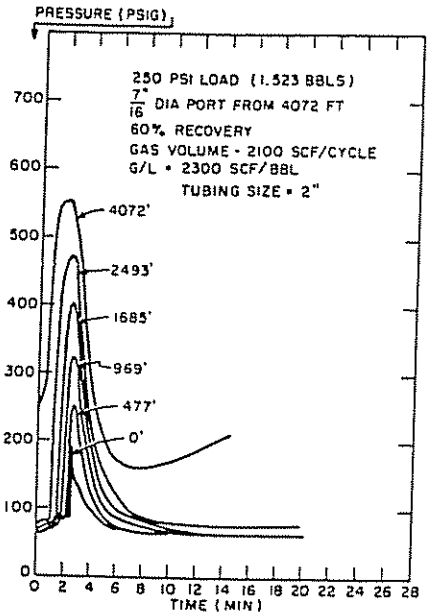


Fig. 3.4106 Typical Pressure Recordings from 4072 ft.

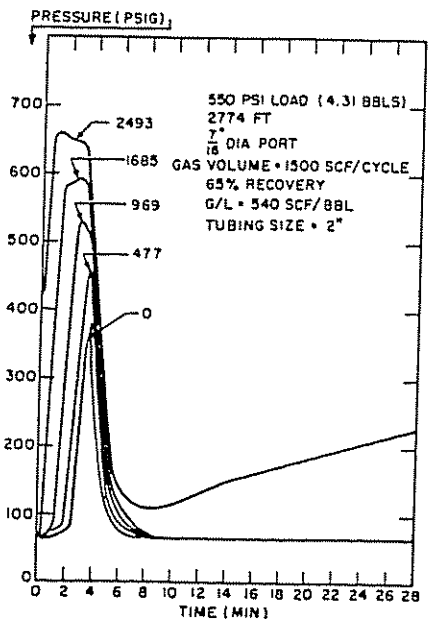


Fig. 3.4107 Typical Pressure Recordings from 2774 ft.

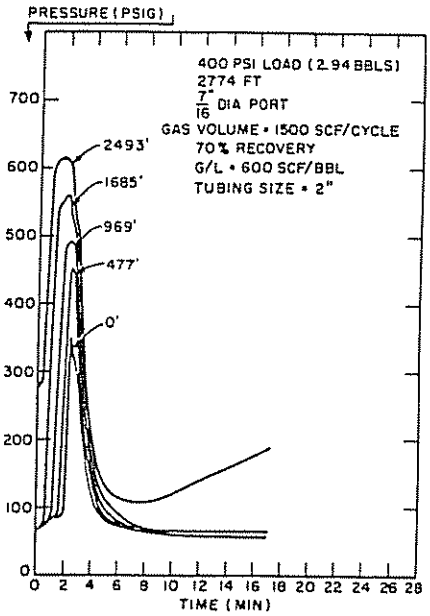


Fig. 3.4108 Typical Pressure Recordings from 2774 ft.

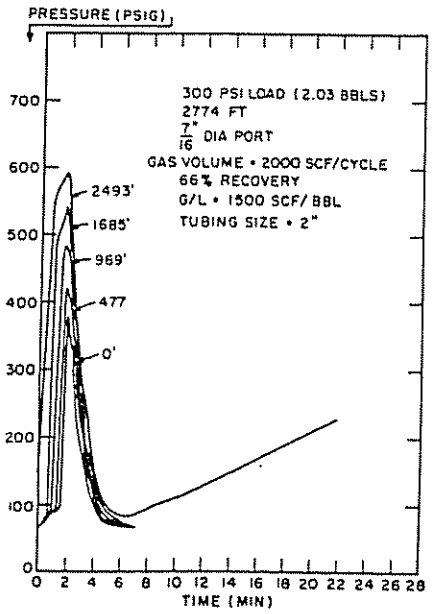


Fig. 3.4109 Typical Pressure Recordings from 2774 ft.

(1) standard tubing installation that lifts from bottom
(2) standard tubing installation that does not lift from bottom
(3) chamber installation

The majority of installations will be standard tubing installations that work from bottom. These should be run with a packer and standing valve if possible. Some installations will intermit above the bottom valve because of low available operating pressure and/or higher production rates; yet, these wells cannot be successfully placed on continuous flow.

There are many low BHP wells of fairly high productive capacity that should be designed to utilize a chamber.

3.4342 Location of top valve depth

The location of the top valve can be extremely important; it depends upon static BHP, and whether or not the well is to be loaded with "kill" fluid. Many intermittent wells have very low BHP; if they offer no attempt to flow during the workover operations, the static fluid level may be several thousand ft down the

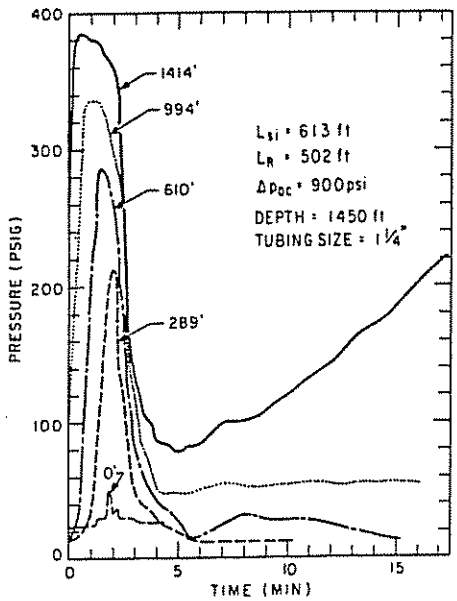


Fig. 3.4110 Typical Pressure Recordings in 1¼-in. Diameter Tubing (after Deschner [50, 51])

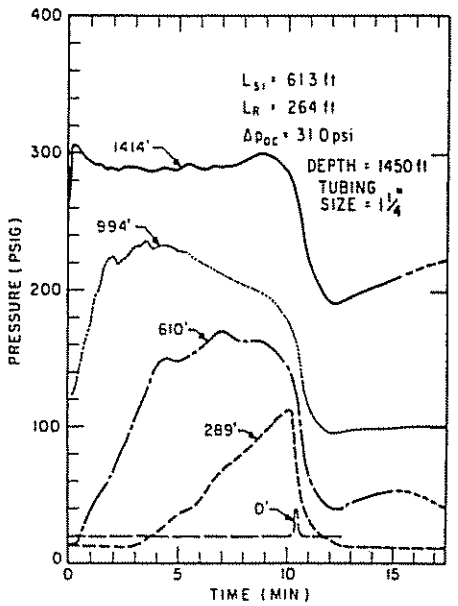


Fig. 3.4112 Typical Pressure Recording in 1¼-in. Diameter Tubing (after Deschner[50, 51])

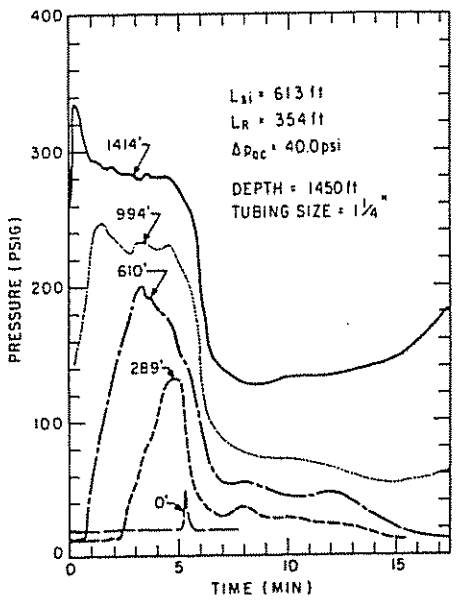


Fig. 3.4111 Typical Pressure Recording in 1¼-in. Diameter Tubing (after Deschner[50, 51])

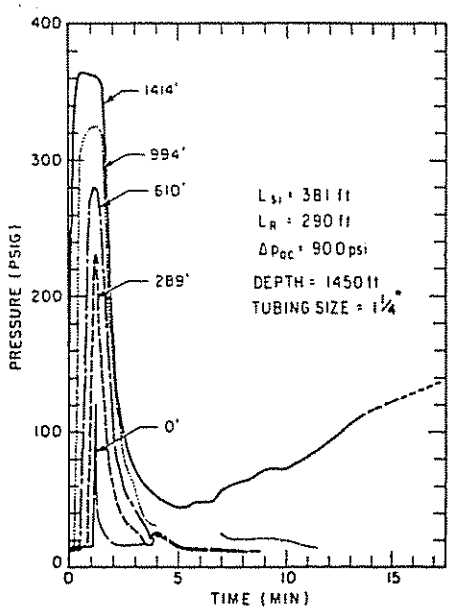


Fig. 3.4113 Typical Pressure Recording in 1¼ in. Diameter Tubing (after Deschner[50, 51])

hole. It is permissible to place the first valve at this point. Do not attempt to place the valve any lower than the static fluid level because of liquid displacement from the packer and tubing string.

If the well is loaded with "kill" fluid to the surface, then the top valve is spaced according to available kick-off pressure. For example, if 900 psi is available, and the well is loaded with a 0.50 psi/ft salt water and is unloaded into a 50 psi system, then the first valve may be placed at $\frac{(900-50)}{0.50} = 1700$ feet. If graphical spacing is being used, the first valve can be placed slightly lower due to gas column weight.

3.4343 Available pressures and valve settings

There may be a kick-off pressure available to unload a well initially that exceeds normal available operating pressure. Available operating pressure is that pressure which can be maintained at the well head under all operating conditions. If the line pressure fluctuates, consider the minimum pressure as the pressure under

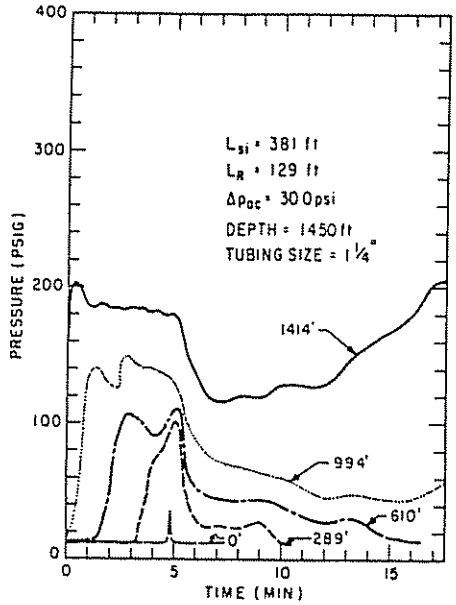


Fig. 3.4114 Typical Pressure Recording in 1¼ in. Diameter Tubing (after Deschner[50, 51])

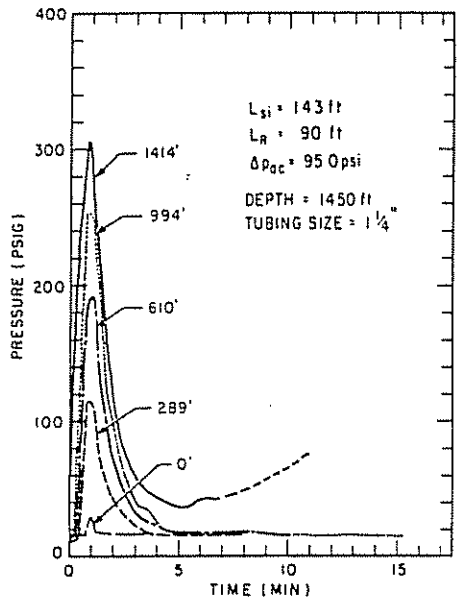


Fig. 3.4116 Typical Pressure Recording in 1¼ in. Diameter Tubing (after Deschner[50, 51])

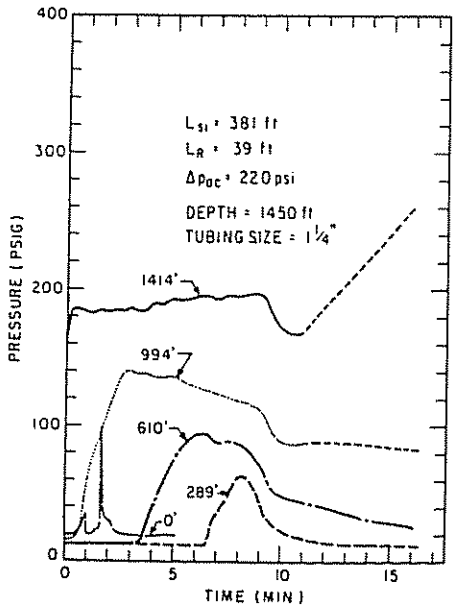


Fig. 3.4115 Typical Pressure Recording in 1¼ in. Diameter Tubing (after Deschner[50, 51])

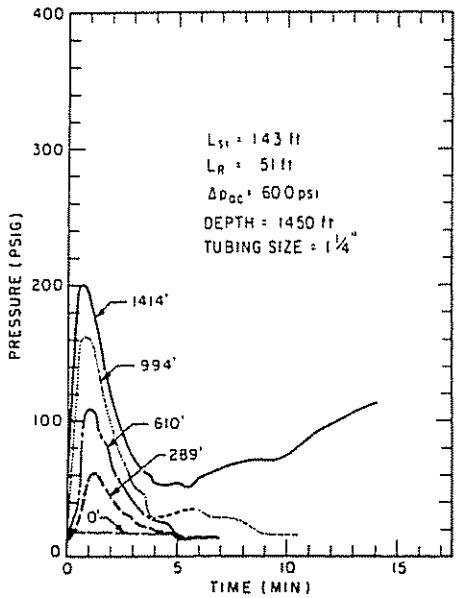


Fig. 3.4117 Typical Pressure Recording in 1¼ in. Diameter Tubing (after Deschner[50, 51])

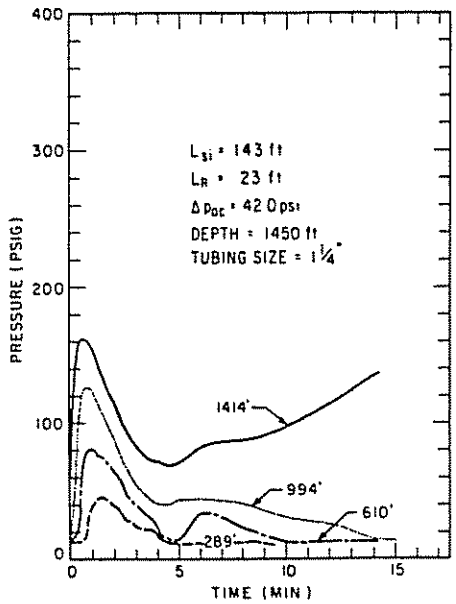


Fig. 3.4118 Typical Pressure Recording in 1¼ in. Diameter Tubing (after Deschner[50, 51])

which the installation will operate at any time during a day's or week's operation. Under no circumstances should operating valves be set to require an opening pressure greater than available line pressure. However, unloading valves may be set higher since they can take advantage of the kick-off pressure.

For example, the kick-off pressure may be 900 psi and the available operating pressure may be 800 psi. The line pressure may fluctuate between 800 and 825 or 850, but the 800 psi minimum should be used. Gas lift valve settings should not exceed the kick-off pressure less 50 psi. For unbalanced valves set on surface closing pressures, the settings should start with a closing pressure of 100 psi less than minimum available operating pressure—for this example, 700 psi. This ensures the valves can be opened.

3.4344 Unloading gradients and spacing

Some gradient must be used in the tubing string to determine distance between valves. Spacing gradients cover two phases in unloading a well. If, for example, an 8000-ft well has 1000 psi BHP, it will support a column of 0.50 psi/ft salt water 2000 feet from bottom. If the well is loaded to the surface to be killed, it will not feed in any fluids in the unloading process until the liquid level has been lowered below 8000 – 2000 = 6000 feet. Two unloading phases will exist. The well will not feed in fluids from the formation until the pressure in the tubing has been lowered below 1000 psi. Therefore, the valves can be spaced to 6000 feet without being concerned with well feed-in. These gradients vary with gas volumes available and unloading rates. They should be used with caution where limited gas volumes are available for unloading. With 100 percent excess gas, gradients as low as 0.01 psi/ft have been obtained for 80 B/D production rates. Gradients as low as 0.02 psi/ft have been used for unloading purposes. Normally, a minimum of 0.04 psi/ft in 2" and 2½" tubing is used. Use higher minimums for smaller tubing sizes (see Figs. 3F.1 and 3F.2 in the Appendix).

After lowering the liquid in the well to the point where the well starts to feed in fluid of its own accord, the spacing gradients will change, depending upon the well feed-in rate. Certainly, a well feeding in at the rate of 300 B/D cannot have valves spaced as far apart as the same well feeding in at the rate of 10 B/D. Refer to Figures 3F.1 and 3F.2 in Appendix D, which show safe gradients to use for wells that are expected to feed in at the rates indicated. These are gradients that can be obtained in continuous flow unloading and therefore offer a safety factor in intermittent unloading. In some instances, valves may be spaced with respective increasing gradients down the hole. This has been shown to be permissible by experimental work of the author. Table 3.484 shows gradients as low as 0.01 psi/ft at the top of the well and as low as 0.05-0.06 psi/ft at bottom. As noted, these are a function of valve port size. Particular attention must be given to valves restricted to port sizes less than $^5/_{16}$ inches. Unloading gradients must be increased accordingly.

Gradients lower than 0.01 psi/ft were noted when unloading to zero pressure with 100% excess gas. The operation was based on average gradients in unloading water from the surface to 6000 feet with a well feed-in rate of 80 B/D.

NOTE: Where gas volume is limited, such as one-well rotative compressor systems, gradients for excess gas must be used with caution.

Since these gradients increase with depth, the overall average gradient would be expected to increase with depth, but would not be so pronounced since it does represent an average from top to bottom. Although Figures 3F.1 and 3F.2 (Appendix F) are unloading gradients based on minimum flowing gradients for continuous flow, they may be used with confidence for intermittent unloading of either water or oil.

3.4345 Differential between valve pressure and tubing load to lift

Another important design consideration is the selection of a correct differential between tubing load and

TABLE 3.484
"AVERAGE" (WATER) UNLOADING GRADIENTS

	Port size (in.)	Gradient: 50% excess gas	Gradient: 25% excess gas
Δp = 300 psi	$^5/_{16}$	0.024	0.028
[difference	$^3/_8$	0.0235	0.0260
between lift	$^1/_2$	0.022	0.0252
pressure and	$^{11}/_{16}$	0.021	0.0236
tubing load]	1	0.020	0.022
Δp = 200 psi	$^5/_{16}$	0.039	0.044
	$^3/_8$	0.038	0.042
	$^1/_2$	0.0354	0.0390
	$^{11}/_{16}$	0.0320	0.0360
	1	0.0270	0.0310
Δp = 150 psi	$^5/_{16}$	0.0420	0.0470
	$^3/_8$	0.0410	0.0454
	$^1/_2$	0.0370	0.0420
	$^{11}/_{16}$	0.0345	0.0385
	1	0.0312	0.0337

TABLE 3.485
DIFFERENTIAL BETWEEN PRESSURE UNDERNEATH SLUG AND TUBING LOAD—$^{7}/_{16}$ IN. PORT

Depth (ft)	Opening pressure opposite valve (psig)	Most efficient load (psig)	Pressure difference between tubing load and lift pressure (psig)	Pressure placed underneath slug (psig)	Actual Δp in tubing string
2774	663	500	163	640	140
4072	629	450	179	620	170
5940	559	400	159	560	160
		(for conditions of minimum GLR)			

injection gas pressure. Table 3.485 shows the results of a series of pertinent tests.

Note that the best efficiencies occur when the differential between the pressure placed underneath the slug and the slug load (including liquid and gas load) is in the range of 140-170 psi. These represent actual test data from an experimental test well. Although this would tend to substantiate a 150 psi differential as being the most efficient, it is not the differential at which maximum production would occur.

A lower flowing BHP can be created by lifting lighter tubing loads. Differentials of 300-400 psi are common. As an arbitrary selection in the start of any design, a value of 200 to 300 psi is recommended. Figure 3.4119 shows how the recovery varies with the differential.

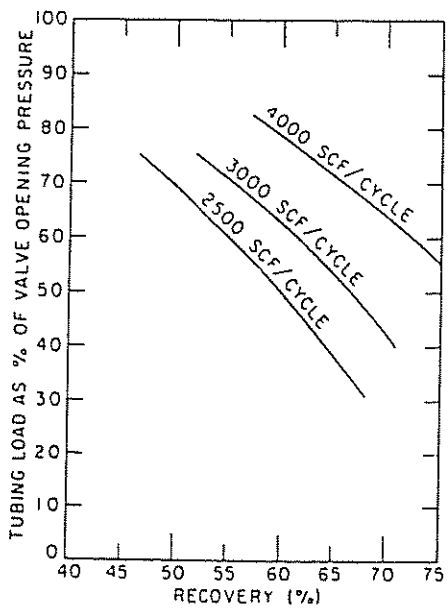


Fig. 3.4119 Percent Recovery vs. Tubing Load

3.4346 Gas lift valve port size

Numerous tests have been run to show the effect of port size in intermittent gas lift operations. The size of the port has a definite effect on the recovery and GLR for a particular set of conditions. For 2" tubing a minimum port size of ½ inch is recommended, unless multi-point is contemplated.

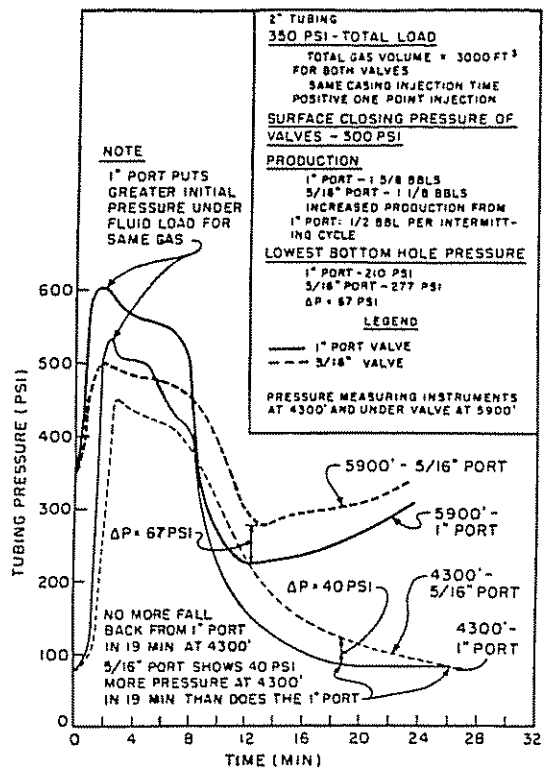


Fig. 3.4120 Comparison of A 1-in. Port with A $^{5}/_{16}$-in Port

Figures 3.4120 and 3.4121 shows typical pressure recordings of slugs lifted by 1, ½, and $^{5}/_{16}$" ported valves. Pressure recordings were made in 2" tubing at 5940 ft, 4290 ft, 1685 ft, and at the surface. All of these recordings are shown in Figure 3.100 along with a recording of the casing pressure. A tabulation on this same figure shows 58% recovery for the $^{5}/_{16}$" port and 74% recovery for the 1" port. The GLR for the $^{5}/_{16}$" port was 2550 scf/B as compared to 2020 scf/B for the 1" port.

Looking at the surface tubing recording, note that the 1" port shows the slug arriving much quicker and with a higher surface tubing pressure. This indicates a good clean slug compared to the same surface recording of the slug arriving from the $^{5}/_{16}$" port.

The BHP recording at 5940 feet shows the 1" port placing more pressure underneath the original slug. The recordings at 4290 feet show that the tubing is cleared of spray and fallback much faster with the 1 in. port than with the $^{5}/_{16}$" port.

In following the recordings at 5940 feet to a minimum, the 1" port creates a much lower minimum pressure than the $^{5}/_{16}$" port. These pressures are 208 psi and 275 psi, respectively, for the 1" and $^{5}/_{16}$" ports.

Figure 3.4122 explains why the larger port sizes do a more efficient job. It shows a plot of maximum pressure under the slug vs. port area. Although the ½" port is sufficient for the heavy loads, an $^{11}/_{16}$" port is required for the lighter loads. The 1" port is better than the $^{11}/_{16}$" port, but the gain is insignificant.

Figure 3.4123 shows how the percent of recovery increases with port size and how the size of the slug lifted affects the recovery.

Figure 3.4124 shows how the percent of recovery varies with the GLR. As noted, the 1" port has a mini-

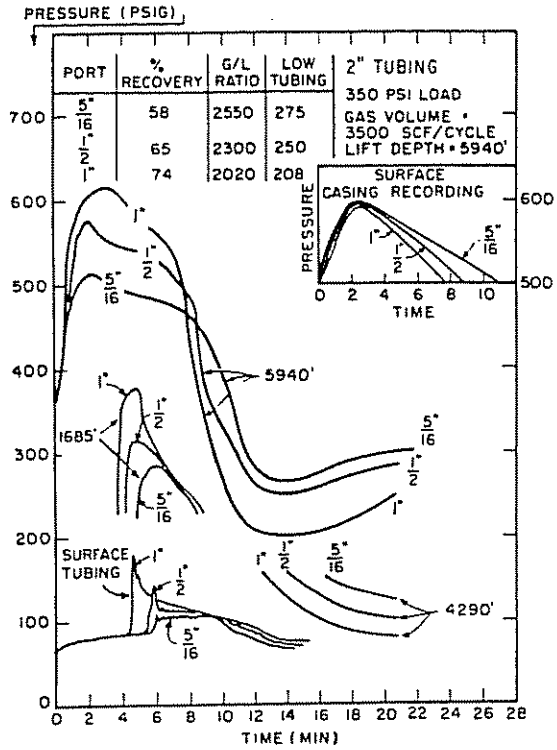


Fig. 3.4121 Comparative Pressure Recordings of 1-in, ½-in. and 5/16-in. Ports (after Brown & Jessen,[13] Courtesy Journal Petroleum Technology)

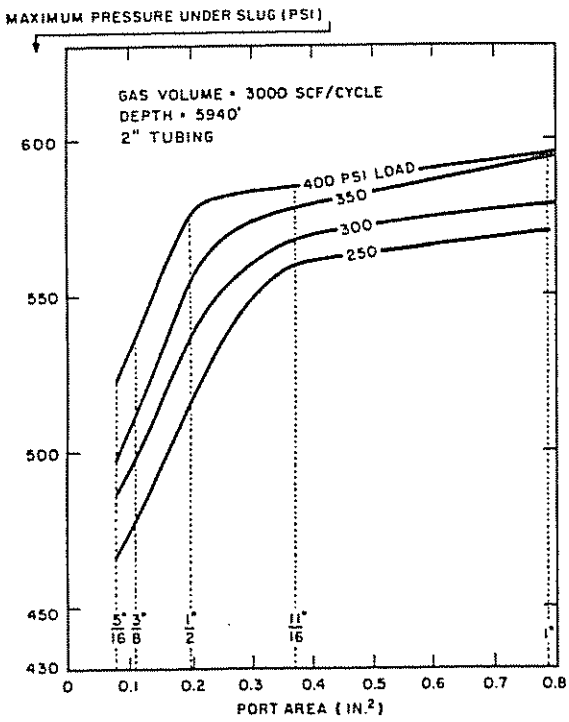


Fig. 3.4122 Maximum Pressure under Slug vs. Port Area (after Brown and Jessen,[13] Courtesy Journal Petroleum Technology)

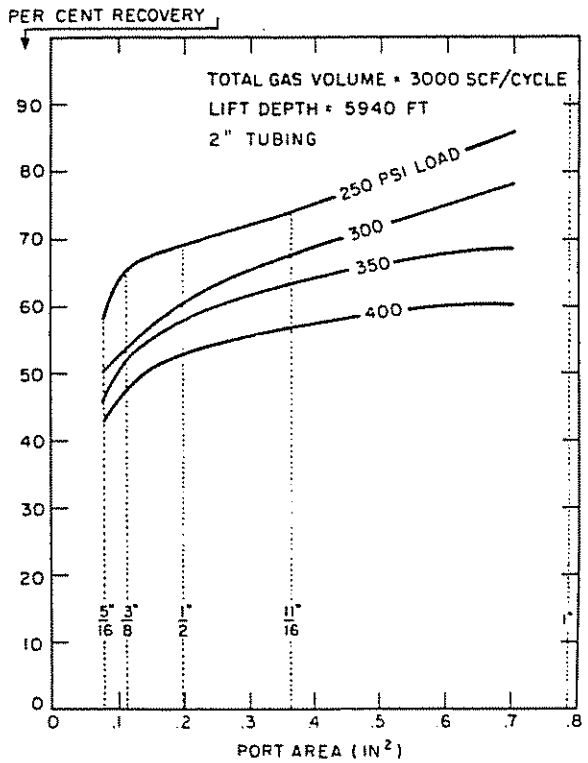


Fig. 3.4123 Percent Recovery vs. Port Area for Varying Loads (after Brown[51])

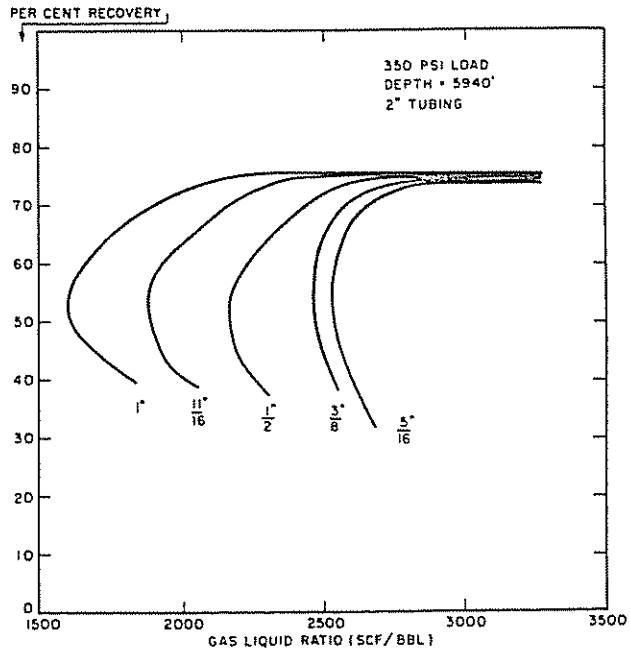


Fig. 3.4124 Percent Recovery vs. GLR (after Brown & Jessen, [13] Courtesy Journal Petroleum Technology)

mum of 1600 scf/B as compared to 2600 scf/B for the 5/16" port.

Figure 3.4125 shows the effect of port size on percent of recovery for different gas volumes per cycle.

Figures 3.4126 through 3.4132 show the effect of various variables and are listed:

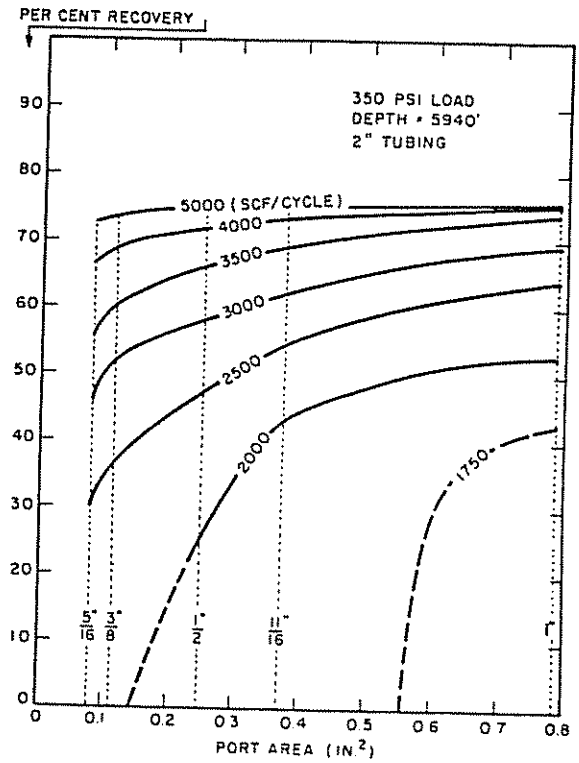


Fig. 3.4125 Percent Recovery vs. Port Area (after Brown & Jessen,[13] Courtesy Jour. Pet. Tech.)

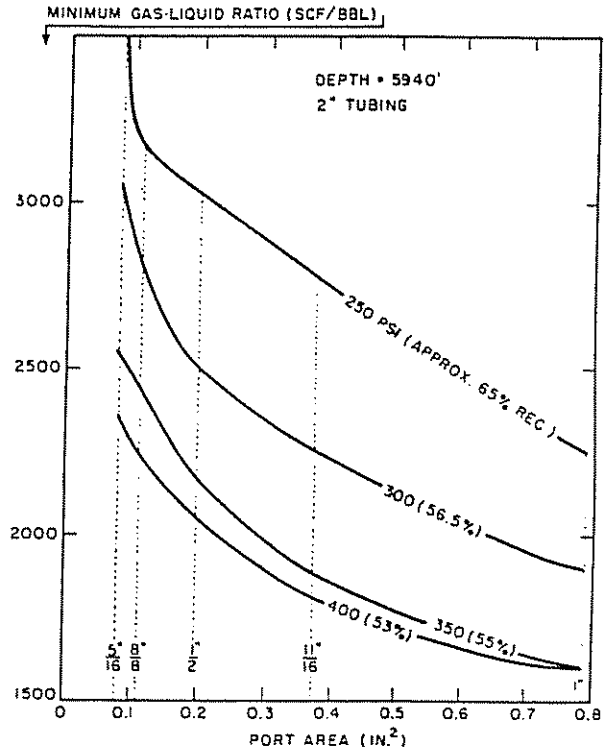


Fig. 3.4127 Minimum GLR for All Loads (after Brown & Jessen,[13] Courtesy Jour. Pet. Tech.)

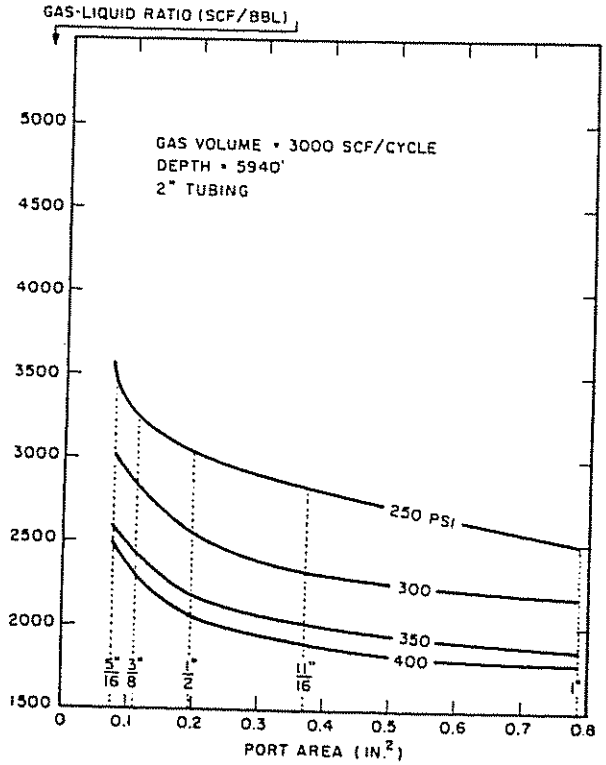


Fig. 3.4126 GLR vs. Port Area for All Loads (after Brown & Jessen,[13] Courtesy Jour. Pet. Tech.)

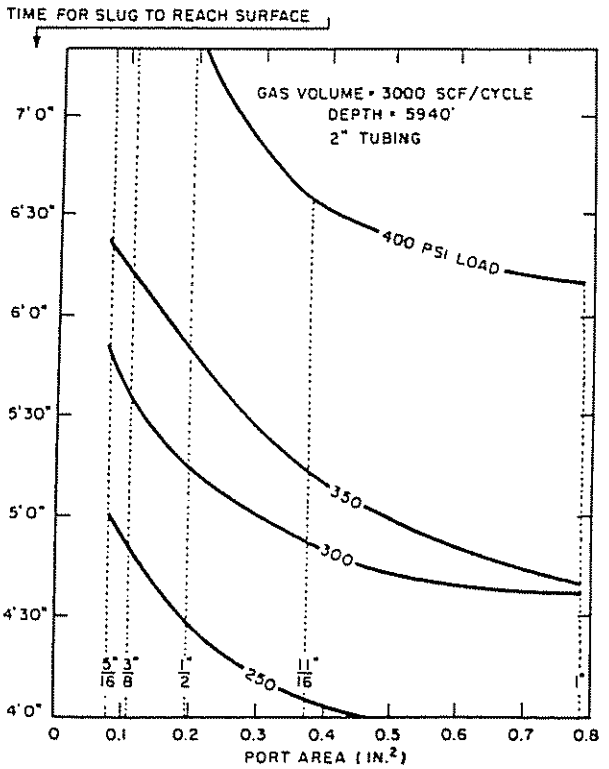


Fig. 3.4128 Time for Slug to Reach Surface for Varying Loads (after Brown and Jessen,[13] Courtesy Jour. Pet. Tech.)

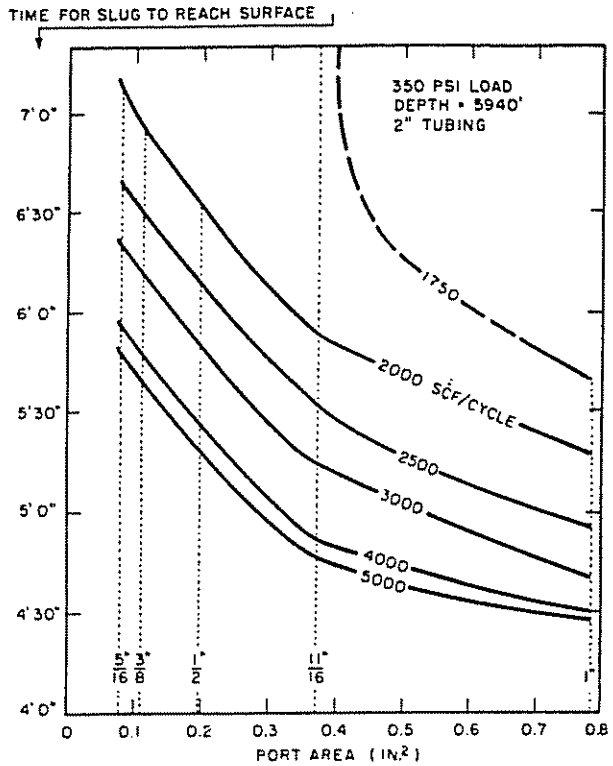


Fig. 3.4129 Time for Slug to Reach Surface for All Ports and Varying Gas Volumes Per Cycle (after Brown & Jessen,[13] Courtesy Jour. Pet. Tech.)

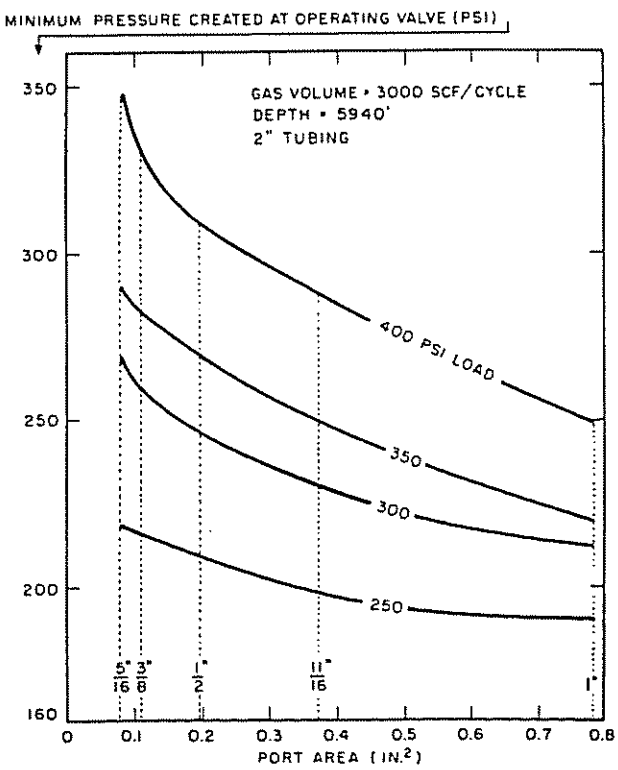


Fig. 3.4130 Minimum Pressure Created at the Operating Valve (after Brown & Jessen,[13] Courtesy Jour. Pet. Tech.)

Fig. 3.4126—Gas-liquid ratio
Fig. 3.4127—Minimum gas-liquid ratio
Fig. 3.4128—Velocity of slug
Fig. 3.4129—Velocity of slug
Fig. 3.4130—Minimum pressure
Fig. 3.4131—Minimum pressure
Fig. 3.4132—Pressure stabilization time

The figures are self-explanatory and show why the larger ports are more efficient for intermittent single point injection.

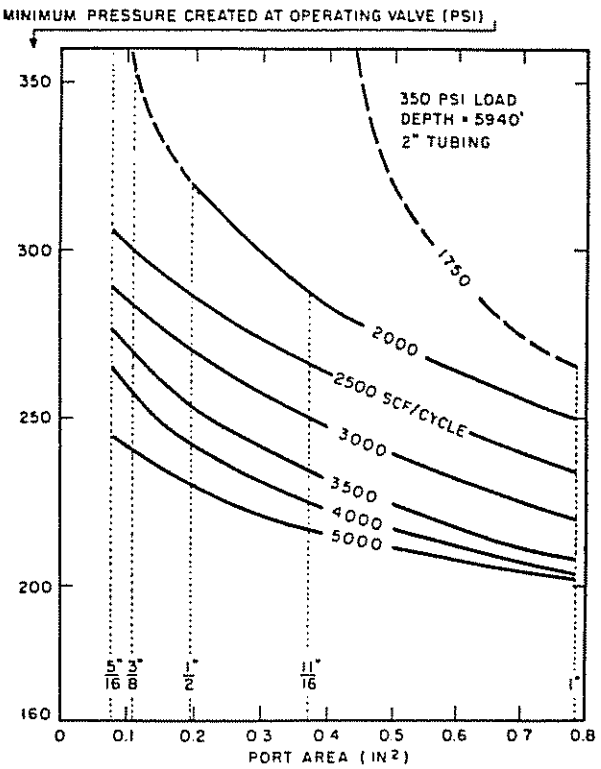


Fig. 3.4131 Minimum Pressure Created at the Operating Valve for 350 PSI Load and Varying Gas Volumes Per Cycle (after Brown & Jessen,[13] Courtesy Jour. Pet. Tech.)

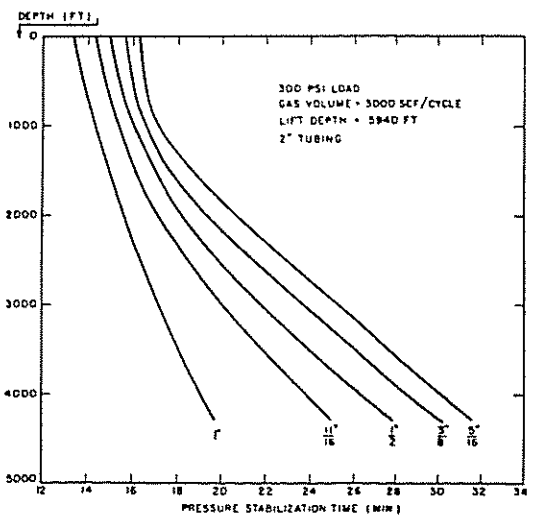


Fig. 3.4132 Pressure Stabilization Time vs. Depth for 300 PSI Load (after Brown[5])

TABLE 3.486
GAS LIFT PORT DIAMETERS (IN INCHES) IN VARIOUS TUBING SIZES GIVING SAME PORT/TUBING AREA RATIO AS THE TEST PORT SIZES (1 IN., 11/16 IN., ½ IN., 3/8 IN., 5/16 IN.) GIVEN IN THE TEST TUBING SIZE (2 IN.)

2 in.		¾ in.		1 in.		1¼ in.		1½ in.		1¾ in.		2½ in.		3 in.	
Dec.	Frac.	Dec.	Frac.	Dec.	Frac.	Dec.	Frac.	Dec.	Frac.	Dec.	Frac.	Dec.	Frac.	Dec.	Frac.
1.00	1	0.411	27/64	0.525	17/32	0.69	11/16	0.80	13/16	0.885	57/64	1.11	1 7/64	1.50	1½
0.6875	11/16	0.282	9/32	0.360	23/64	0.474	15/32	0.565	9/16	0.602	19/32	0.77	49/64	1.028	1 1/32
0.50	½	0.206	13/64	0.262	17/64	0.33	21/64	0.40	13/32	0.437	7/16	0.56	9/16	0.748	¾
0.375	3/8	0.155	5/32	0.208	13/64	0.259	¼	0.30	19/64	0.329	21/64	0.42	27/64	0.562	9/16
0.3125	5/16	0.129	⅛	0.164	11/64	0.215	7/32	0.25	¼	0.273	17/64	0.35	11/32	0.467	15/32

Since this experimental work was performed in 2" tubing, Table 3.486 has been prepared to show the port size necessary in other tubing sizes to be equivalent to the same port-tubing area ratio used in the 2" tubing.

This shows the need for a 21/64" port in 1¼" tubing to be equivalent to a ½" port in 2" tubing. In all cases, strive for at least a ½" port in 2" tubing and its equivalent in other tubing sizes.

From additional data from experimental tests, the following table has been prepared, showing how the GLR and pressure stabilization time may be changed depending upon port size. A 7/16" port has been taken as unity. For example, if a 7/16" port requires a GLR of 3000 scf/B then a 1" port should require (0.75) (3000) = 2250 scf/b.

The following Table 3.487 of multiplying factors shows the effect of port diameter other than 7/16".

TABLE 3.487
MULTIPLYING FACTORS

Port size (in.)	Factor to change GLR	Factor to change pressure stabilization for same gas volume per cycle
5/16	1.100	1.100
3/8	1.050	1.050
7/16	1.000	1.000
½	0.975	0.975
9/16	0.950	0.950
11/16	0.850	0.850
1	0.750	0.750

3.4347 Percent recovery

A portion of a liquid slug will be lost on its trip to the surface. An approximate figure is 7% loss per 1000 feet of travel. However, this varies, depending upon tubing size and amount of gas injected per cycle. The point of minimum GLR does not occur at the point of maximum recovery.

To calculate the anticipated fluid recovery per head, use the following formula:

$$B_p = \frac{C_t(P_t - P_{wh})}{G_s}\left[1 - S_f\left(\frac{D_v}{1000}\right)\right] \qquad (3.436)$$

where:

B_p = recovered liquid per cycle in bbl (barrels)
p_t = pressure in tubing opposite the valve at time of lift in psi
C_t = tubing volume factor in B/ft
S_f = slippage loss factor — 7%/1000 ft is often used
P_{wh} = tubing pressure on top of slug
D_v = depth of valve in ft
G_s = gradient of well fluid in psi/ft

EXAMPLE PROBLEM—CALCULATING INTERMITTENT FLUID CYCLE RECOVERY

Tubing pressure at valve depth at time of lift = 600 psi
Well head tubing pressure = 50 psi (assume same pressure on top of slug)
Loss factor = 0.07/1000 ft
Tubing size = 2⅜ in. OD [$C_t \doteq 0.00387$ b/ft]
Valve depth = 5000 ft
Well fluid gradient = 0.4 psi/ft
$B_p = [0.00387\,(600 - 50)]/0.4 \times [1 - 0.07\,(5000/1000)]$
$B_p = 1.37$ bbl/cycle

To calculate daily fluid production for intermittent lift, the following formula is used:

$$q_L = N\,B_p \qquad (3.437)$$

q_L = production rate in B/D.

$$N = \frac{1440}{3 \times D_v/1000} \qquad (3.438)$$

where:

N = Maximum cycles per day
D_v = Depth of operating valve in ft

Therefore:

Calculating Daily Intermittent Liquid Production Rate:

$$N = \frac{1440}{3 \times 5000/1000} = 96 \text{ cycles/d}$$

$$q_L = 96 \times 1.37 = 122.5 \text{ B/D of liquid}$$

Experimental well tests in 2" tubing show that 7% loss per 1000 feet is representative of the expected loss for minimum GLRs. Additional recovery could be obtained at higher GLRs—maximum recovery does not occur at the minimum GLR. Tests show a recovery up to 73% from 6000 feet with the minimum GLR occurring at 58% recovery. These results were measured in an experimental well.

For design purposes use a loss factor of 5-7% per 1000 feet of travel for 2" tubing. In general this is comparable with other tubing sizes. This loss factor is more on the order of 3-5% for chamber installations and is further reduced when plungers are used in conjunction with gas lift.

3.4348 Gas volume requirements for intermittent lift

In general the necessary gas volume to remove an intermittent slug of fluid from a well is the volume of gas needed to fill the tubing string at the average gas pressure underneath the fluid slug from bottom to top.

To eliminate tedious longhand calculations, a series of figures have been prepared to allow a quick determination of gas volumes needed (Figs. 3G.1 through 3G.1b in Appendix G). These are the same figures given by Davis, et al.[23]

Following is a derivation of the formula used in these calculations and average conditions used:

API = 35° (Gradient = 0.3689 psi/ft)
Gas sp gr = 0.65
T_{grad} = 1.6°F/100 ft
T_{surf} = 80°F

$$p_{avg} = \frac{\{p_c + P_{wh} + [1 - (0.07 \times L)/1000]S \times G_s\}}{2.0} + 14.7$$

$$= \left\{\frac{p_c + P_{wh}}{2} + 14.7 + 0.18445\,S[1 - 7 \times 10^{-5}L]\right\} \quad (3.439)$$

$$\frac{p_{avg}}{p_s} = \frac{p_{avg}}{14.7} = \frac{p_c + P_{wh}}{29.4} + 1 + (0.01255)S\,[1 - 7 \times 10^{-5}L] \quad (3.440)$$

$$T_{avg} = \frac{T_{surf} + (T_{grad}/100)(1 - 7 \times 10^{-5}L)S + T_{surf} + (T_{grad}/100)L}{2.0} + 460 = (0.008)SL\left(\frac{1}{S} + \frac{1}{L} - 7 \times 10^{-5}\right) + 540 \quad (3.441)$$

$$\frac{T_{sc}}{T_{avg}} = \frac{520}{T_{avg}} = \frac{520}{(8 \times 10^{-3})SL(1/S + 1/L - 7 \times 10^{-5}) + 540} \quad (3.442)$$

$$H_{gc} \times Cap. = \{L - [1 - 7 \times 10^{-5}L]S\} = 5.4541 \times 10^{-3}(d)^2 \quad (3.443)$$

$$H_{gc} \times Cap. = (5.4541 \times 10^{-3})(d)^2(LS)\left[\frac{1}{S} - \frac{1}{L} + 7 \times 10^{-5}\right] \quad (3.444)$$

$$\text{Gas volume} = (H_{gc} \times Cap.)\left(\frac{p_{avg}}{p_s}\right)\left(\frac{T_s}{T_{avg}}\right)\left(\frac{1}{z}\right) \quad (3.445)$$

$$= (5.4541 \times 10^{-3})(d)^2(LS)\left[\frac{1}{S} - \frac{1}{L} + 7 \times 10^{-5}\right]\left[\frac{p_c + p_t}{29.4} + 1 + (1.255 \times 10^{-2})(LS)\left(\frac{1}{L} - 7 \times 10^{-5}\right)\right] \times \left[\frac{520}{(8 \times 10^{-3})(LS)(1/S + 1/L - 7 \times 10^{-5}) + 540}\right]\left[\frac{1}{z}\right] \quad (3.446)$$

where:

S = submergence in ft
L = depth to valve in ft
d = tubing ID in in.
p_c = casing pressure at valve psi
P_{wh} = surface tubing pressure in psi
z = compressibility factor at p_{avg}, T_{avg}
Cap. = capacity of tubing = 0.0054541 $(d)^2$ cu ft/ft
H_{gc} = length of gas column in tubing string in ft

Figures 3.4133 A, B and C show additional charts prepared from these experimental tests. These figures show that although the recovery can be increased, additional gas is required to obtain this recovery.

3.4349 Cycle frequency and pressure stabilization time

How fast is it permissible to cycle an intermittent installation? This question becomes important only when a well must be cycled faster than the pressure stabilization time. Pressure stabilization time is the time required for all the spray and fallback to settle in the tubing string. There are many wells that need not be cycled faster than one to two hours. In these cases all the liquid fallback has settled and there is no worry about cycle frequency. However, some wells must be cycled faster than their pressure stabilization time.

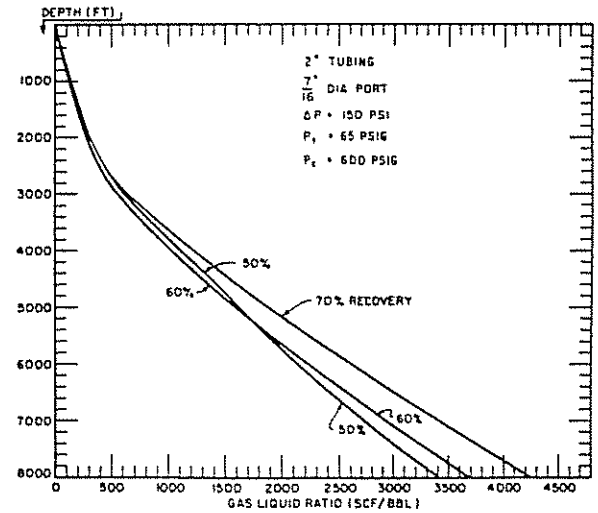


Fig. 3.4133-A GLRs

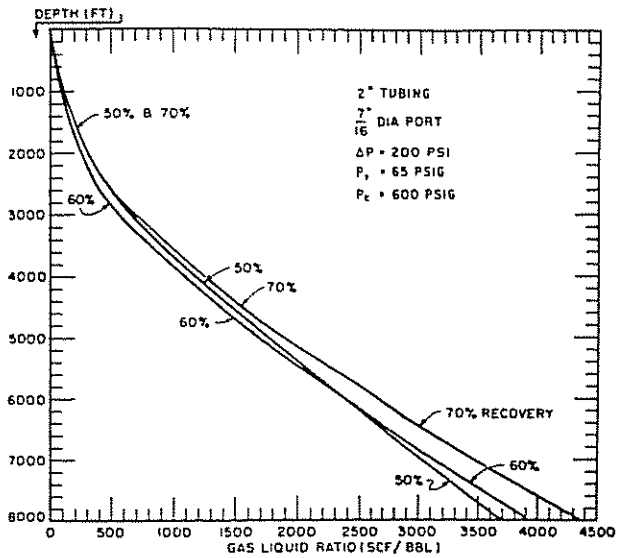


Fig. 3.4133-B GLRs

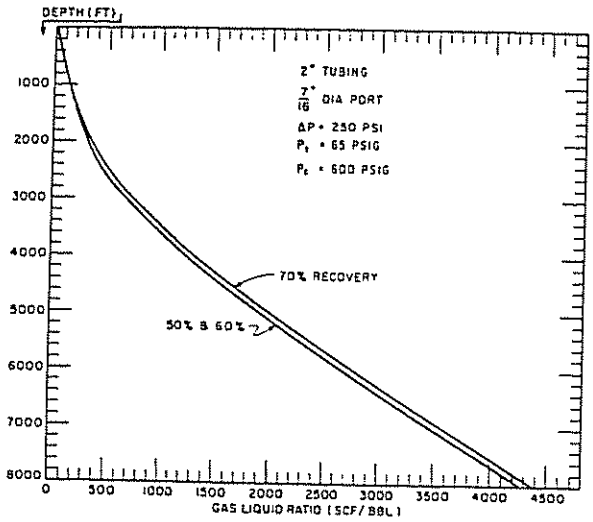


Fig. 3.4133-C GLRs

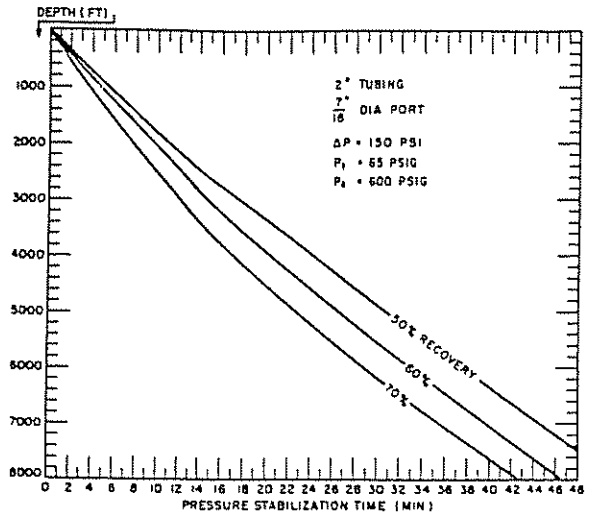


Fig. 3.4134 Pressure Stabilization Time vs. Depth

Some are cycled so fast that they approach a continuous flow condition.

For 2" tubing a good approximation for cycle time in minutes that will produce good clean slugs is 1.5 × [depth/1000]. For example, in an 8000-ft well, this is 1.5 × [8000/1000] = 12 min. (This represents the minimum time.) Although the well may be cycled faster than this, it may also do better on continuous flow. For 2" tubing the velocity of the slug will be approximately 1000 ft/min. Therefore, in the preceding example only 2 or 3 minutes pass after the slug has cleared before the next slug leaves bottom. Certainly in this case all the fallback has not settled down in the tubing string.

The pressure stabilization time may be reduced considerably by lifting lighter slugs (see Figs. 3.4134, 3.4135, and 3.4136).

As the differential between the operating casing pressure and tubing pressure loads increases (denoted by Δps of 150, 200, 250, and 300 psi on the various figures) the pressure stabilization time decreases. Also, as the recovery increases, the pressure stabilization time decreases. Lighter loads are lifted at the higher Δps, giving rise to a greater velocity, hence, a greater recovery and shorter pressure stabilization time.

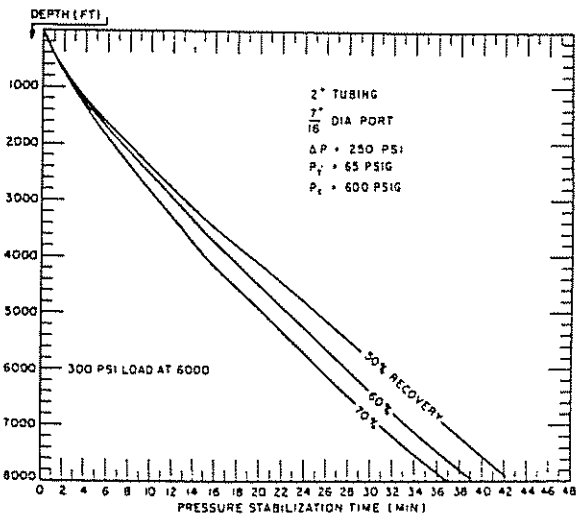


Fig. 3.4135 Pressure Stabilization Time vs. Depth

3.43410 Types of valves for intermittent lift

Various gas lift valves may be used for intermittent lift. These were discussed in Section 3.42 on valve mechanics. In general, for single point injection the valve must have a large port and be fast opening. Depending upon the type of surface gas control, it may or may not need to exhibit spread.

If the gas is controlled entirely from the surface, a low spread valve may be used for intermittent lift. If it is desired to use the annular space for partial gas volume storage and to inject additional gas through an intermitter at the surface then an unbalanced bellows valve is satisfactory. If it is desirable to use 100% choke control of the injection gas, then a pilot-operated valve should be considered. Pilot-operated valves may be used for all three applications. Refer to Section 3.42 for a more detailed discussion of these valves.

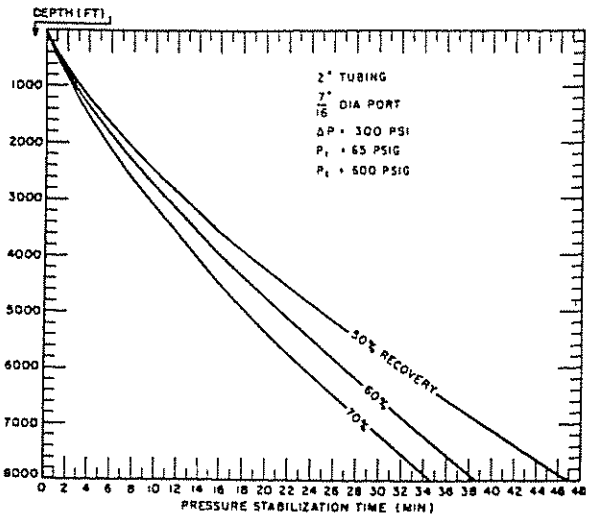


Fig. 3.4136 Pressure Stabilization Time vs. Depth

Unbalanced valves may be set at either the same surface closing pressure, or a 10 psi or more drop per valve in surface closing pressure, at the same surface opening pressure, or at a drop in surface opening pressure down the hole. Normally, a slight drop (10-25 psi per valve) in surface closing pressure should be taken, particularly if sufficient gas pressure is available.

3.43411 Single point vs. multipoint injection for intermittent lift

Although most installations have been designed for the single point injection of gas, there are numerous multipoint installations in the field today. Some controversy still exists whether multipoint injection can be successfully applied, and certainly it presents a more difficult design procedure. Multipoint injection (to the extent of opening one to three more valves above the operating valve) is used in many installations that were designed originally for single point injection. There appears to be no detrimental effect and may be beneficial.

Neeley et al. conducted numerous field tests on intermittent gas lift and concluded that multipoint and single point injection performed equally as well regarding percent recovery and gas volume requirements.[29] For 29 tests conducted for single point injection, the recovery was 52% and for 22 tests conducted for multipoint injection the recovery was 53%. This gave a loss figure of 9.87% per 1000 ft for single point injection and 9.66% per 1000 ft. for multipoint injection, for an overall average figure of 9.77% loss per 1000 ft.

These figures are surprisingly high based on the results of Brown and field experience. The port sizes were probably too small, or the valves were failing to open completely, that is, to the full port size. This would certainly increase the fall-back of liquids for single point injection, since Brown showed that a large port size was very important in obtaining high recoveries. The effect of port size on multipoint injection is unestablished. Multipoint injection has been used for certain fluid operated valve installations to off-set the effect of small port sizes required in some fluid trip valves. Both pressure-operated and fluid-operated valves are used in the following design examples.

3.43412 Summary of design considerations in intermittent lift

The type of installation should be selected from well productivity information. By knowing the kick-off pressure and operating pressure, the location of the top valve can be determined. From the production rate an unloading gradient can be determined [from Figs. 3F.1 and 3F.2 (Appendix 3F)] to be used in spacing. Gas volumes required can be determined from Figures 3G (Appendix 3G). For single point injection a minimum of a ½" port diam should be used in 2" tubing, and the equivalent port/tubing area ratio should be maintained for other tubing sizes. A design differential of 150-300 psi between operating injection pressure at the valve and tubing pressure load should be used where possible. The percent recovery can be approximated by a loss of 7% per 1000 feet of travel. Generally the final cycle frequency is determined by trial and error settings in the field.

3.435 Design and spacing procedures for intermittent gas lift installations

3.4351 Introduction

Several procedures are available for the spacing of gas lift valves for intermittent installations. These vary, depending upon the manufacturer's recommendations and upon the type of gas lift valves. Normally, pressure operated valves can be spaced further apart than fluid operated valves. Any valve, such as a combination fluid opened-pressure closed valve, must be spaced according to the fluid operated section. If it is to be a relatively low productivity well and the operating valve is to be the bottom valve, then the upper valves are used only for unloading down to the bottom and can be relatively far apart. However, it is bad practice to place them too far apart so unloading cannot take place successfully.

If a well has unknown capabilities, another type of spacing should be used so valves will be sufficiently close to one another in the possible operating range. This type of spacing is described in Section 3.4358 and is one whereby the deepest point of operation is automatically achieved by the correct design. The surface control also has an influence on the design of intermittent installations. The design is more critical if a surface choke is being used for gas injection compared to time cycle control.

For surface choke control, use an operating valve with the correct spread to supply the gas volume requirements per cycle. If a time cycle controller is to be used, then the spread is not so critical, except it should not be too large since the gas volume required per cycle can be supplied from the surface.

Multipoint injection is a special case, and extra care must be taken in spacing and design to accomplish this method of lift to prevent opening a valve into the slug of liquid.

The following example problem is to be worked by these spacing and design procedures.

Given data:
- 2⅜" O.D. tubing (1.995" I.D.)
- 7" casing
- P_{so} = 1000 psi
  (Available pressure = 1050 psi)
- G_s = 0.40 psi/ ft = kill fluid gradient
- Depth 7600 ft
- γ_g = 0.7
- P_{wh} = 80 psi
- P_R = 1500 psi
- Temp bottom – 167°F.
- Temp surface – 109°F.
- API = 35
- Average PI = 0.2
  Based on 50% drawdown
- One well test shows 150 B/d at P_{wf} = 750 psi
- Bubble point pressure greater than 1500 psi

To best illustrate every variable involved in intermittent design, the first design will be for one using

standard pressure operated valves so they are in a balanced condition (just ready to open as the slug passes by each valve on its journey to the surface). It could be considered a multi-point injection design if the slug load is determined correctly, or it may be considered as a design to prevent multi-pointing if the slug load is slightly less than anticipated.

The design is explained in the following Section 3.4352 and incorporates in detail many of the principles used in following sections on intermittent design. Section 3.4352 should be studied in detail so that a good understanding of all aspects of intermittent lift will be obtained.

3.4352 Design procedure for intermittent gas lift well. For either single-point or multi-point injection, standard pressure operated valves, time cycle control, or choke control at the surface.

A new procedure is offered for the design and spacing of valves for a multi-point intermittent gas lift installation, or it may be used to prevent multi-pointing. An example problem will be worked in explaining the procedure.

EXAMPLE PROBLEM

Data:
- 2" tubing
- 7" casing
- Depth– 7600 ft
- $\gamma_g = 0.7$
- $P_{so} = 1000$ psi (available pressure = 1050 psi)
- $P_R = 1500$ psi
- $P_{wh} = 80$ psi
- Well killed with produced crude ($G_s = .40$ psi/ft)
- Average PI = 0.2 based on 50% drawdown
- Well test shows 150 B/D at $P_{wf} = 750$ psi
- Bubble point press greater than 1500 psi
- Valves set at constant surface closing pressure

Refer to Fig. 3.4137 for a graphical representation of this procedure.

(1) Prepare a pressure depth scale on a sheet of rectangular coordinate graph paper (Fig. 3.4137).
(2) Mark $P_{so} = 1000$ psi at the top and draw in the gas gradient line from this point. Obtain this from Appendix 3A.1. It is 26.2 psi/1000 ft.
(3) Tentatively select a constant surface closing pressure of 100 psi less than P_{so} (select 900 psi). Mark this point at the top and draw in the gas gradient line which is 23.6 psi/1000 ft, as found from Figure 3A.1 in the Appendix.
(4) Select an appropriate unloading gradient. Refer to Fig. 3F.2 in the Appendix. These curves show minimum gradients possible by continuous flow. Common practice is to choose an unloading gradient based on the final producing rate. For a solution gas drive well and based on the work of Vogel, $q_{o(max)}$ for $P_{wf} = 0 = 214$ B/D. Base the unloading gradient on 200 B/D and find $G_u = .064$ psi/ft from Figure 3F.2. Starting from $P_{wh} = 80$ psi, draw in the unloading gradient line, extending to well depth of 7600 ft.
(5) Determine the gas volume necessary to lift one cycle of liquid.

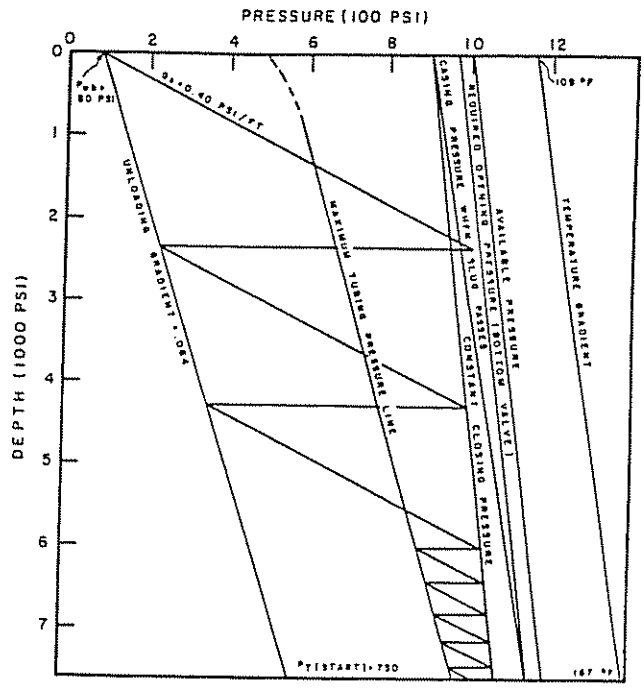


Fig. 3.4137 Design to Explain Both Multipoint and Single Point Injection (Pressure Operated Valves)

Valve closing pressure at 7600 ft = 1080 psi. (From Fig. 3.4137) Assume valve opening pressure = 1080 + 75 = 1155 psi and determine gas volume necessary per cycle. Gas volume from Fig. 3G.8 in the Appendix = 7400 scf/cycle.

(6) Determine the casing pressure change required to furnish 7400 scf of gas from Fig. 3H.6 in the Appendix $\Delta P = 78$ psi.
(7) Select the maximum tubing load to be lifted. As an approximation use $P_{tubing} = \frac{1}{2} P_R = \frac{1}{2}(1500) = 750$ psi = total pressure in tubing at operating valve. If fluid feed-in characteristics are available, use this information to select the tubing load.
(8) Select the valve control port size. Assume choke control at the surface. From Fig. 3G.8 select a $^7/_{16}$" control port size for $(P_{vo} - P_t) = 1158 - 750 = 408$ psi where true $P_{vo} = 1080 + 78 = 1158$ psi. Use a pilot valve with a $^5/_8$" main port size. For the control port solve for

$$R = \frac{P_{vo} - P_{vc}}{P_{vo} - P_t} = \frac{1158 - 1080}{1158 - 750} = .191 = \frac{A_p}{A_b} \quad (3.47)$$

The nearest port size is $^7/_{16}$". R values may be selected from Table 3.488 and will differ slightly depending upon the manufacturer.

This amount of spread is suitable for choke control at the surface. However, if the designer wants to use time cycle control, he would select $\frac{1}{2}(\Delta P) = \frac{1}{2}(78) = 39$ psi spread and find a port size more on the order of $^5/_{16}$" for this type of control. We will use choke control for this well.

(9) Construct the maximum tubing pressure line from bottom to top. Tests show that the slug accelerates rapidly and reaches a more or less con-

TABLE 3.488
"R VALUES"
BELLOWS AREA AND SEAT AREA RELATIONSHIPS FOR A_b = 0.77 IN² FOR 1½" VALVE AND .29 IN² FOR 1" VALVE

Diameter of spread control seat — in.	For 1" O.D. Valves R	For 1" O.D. Valves 1 – R	For 1½" O.D. Valves R	For 1½" O.D. Valves 1 – R
3/16	.0863	.9137	.0359	.9641
¼	.1534	.8466	.0638	.9362
9/32	.1942	.8058	—	—
5/16	.2397	.7603	.0996	.9004
11/32	.2900	.7100	—	—
⅜	.3450	.6550	.1434	.8566
7/16	.4697	.5303	.1952	.8048
½	—	—	.2562	.7438
9/16	—	—	.3227	.6773

stant velocity before reaching the next uppermost valve, and it appears to accelerate again near the surface. With resisting force equal to pushing force, the pressure at the bottom of the slug is equal to the gas pressure underneath the slug. Depending on port size and differential across the valve, the pressure measured inside the tubing opposite the operating gas lift valve can be approximated as follows: $P_{slug} = P_{vo} - (\%)(P_{vo})$ The following table may be used to obtain percent of P_{vo}.

TABLE 3.489
VALUES TO USE FOR FINDING MAX SLUG PRESSURE

Port	Static ΔP ($P_{vo} - P_l$)	% P_{vo} for Dynamic ΔP
3/16 - ¼	200	20
	300	25
	400	30
5/16 - ⅜	200	15
	300	20
	400	25
7/16 - 9/16	200	10
	300	15
	400	20
⅝	200	8
	300	10
	400	15

For this example the pressure underneath the slug and the gas pressure placed in the tubing will be $P_{vo} - .15\ P_{vo}$, based on ΔP Static = 1155 – 750 = 405 psi and a ⅝" main port size. Press at bottom of slug = 1155 – .15(1155) = 982.
This checks reasonably close with frictional resistance offered by the slug moving at 800 ft/min. These calculations are also shown.

Table 3.490 shows the approximate friction loss gradients for velocities ranging from 500 to 1000 ft/min.
Total moving tubing load = liquid load + gas pressure on top of slug + friction due to slug + acceleration (for a constant velocity, acceleration = 0)

TABLE 3.490
FRICTION GRADIENTS

Velocity (ft/min)	Velocity (ft/sec)	Gradient (psi/ft)
1000	16.67	.184
900	1.5	.173
800	13.33	.1186
700	11.66	.0906
600	10.00	.0659
500	8.33	.0461

(a) Liquid + gas = 750 psi
P_{wh} = 80 and gas column = 15 psi; therefore, gas pressure on top of slug = 80 + 15 = 95 psi.
(b) Liquid load = 750 – 95 = 655 psi and represents 1638 ft of .40 psi/ft gradient liquid.
(c) For a velocity of 800 ft/min the frictional loss = (1638)(.1186) = 191 psi.
(d) Dynamic tubing load = 750 + 191 = 941 psi compared to 982 psi as shown by experimental tests. Use the maximum value of 982. Initially, until the slug reaches a constant velocity, there is more pressure underneath the slug; but the slug accelerates very rapidly and some pressure drop occurs across the valve.
(e) The pressure at the bottom of the slug just as it begins its exit from the tubing string can be approximated as follows:
Assume 7% liquid loss/1000 ft of travel
Total loss = (7)(7.6) = 53.2% loss
Length of liquid slug at top = (1638)(1 – .532) = 767 ft
Frictional Resistance for 767 ft with a velocity
of 1000 to 1100 ft/min near the top = (.20)(767) = 153 psi.
The exit velocity of the slug is greater than the normal velocity and can generally be taken as 1000 ft/min or greater.
Total dynamic pressure at 767 ft down the string = pressure exerted by slug + tubing pressure + frictional resistance = 307 + 80 + 153 = 540 psi. (Add 10-15 psi more back pressure due to gas moving out in front = 540 + 15 = 555.) Draw a straight line between 982 psi at the bottom and 555 psi at 767 ft and extend to the top. (zero depth).
(10) Construct a straight line from top to bottom, representing casing pressure at the instant the operating gas lift valve opens. P_{vo} at 7600 as determined in step 8 = 1158 psi. Pressure available at depth = 1200 psi. Extend 1158 psi to the surface deducting gas column weight. This represents 965 psi at the surface. Draw a straight line between 965 psi at the top and 1158 psi at 7600 ft.
(11) Construct a line representing casing pressure at the instant the slug passes each gas lift valve. This is constructed by drawing a straight line between the opening pressure at bottom (1158) and the surface closing pressure (900). Tests

have shown that the operating valve closes at approximately slug arrival time at the surface.

(12) Space the gas lift valves

(a) For Valve 1, space in the conventional manner by constructing the "kill fluid" gradient line of .40 psi/ft from P_{wh} = 80 psi to the available pressure line minus 50 psi permitting 50 psi safety factor to start the unloading procedure (Refer to Fig. 3.4137).

(b) Space two or three more valves in the same manner, utilizing the unloading gradient of 0.064 psi/ft and the constant valve closing pressure line. This locates Valve 3 at 6000 ft.

(c) Space the rest of the valves based on the maximum tubing pressure line and the constant closing pressure line. This locates Valves 4, 5, 6, 7 and 8 at 6420, 6800, 7120, 7450, and 7600 ft. Although this appears to be close spacing, it assures reaching bottom and, in turn, assures multi-point injection as the slug passes each valve. Also, in the event the well is better than anticipated, it insures having valves sufficiently close in the operating range.

Other spacing options could be used here, such as spacing between the maximum tubing pressure line and the gas injection opening pressure line. However, to insure compatibility in duals and a balanced condition just as the slug passes, spacing between the maximum tubing pressure and valve closing pressure line should be used.

(13) Draw in the temperature gradient line beginning with 167°F. at the bottom and connecting with a straight line to 109°F. at the top.

(14) Prepare a summary tabulation as noted in Table 3.491.

Explanation of Various Columns

(1) Valve no.—self explanatory

(2) Depth—from graph—(Fig. 3.4137)

(3) Temp.—from graph—(Fig. 3.4137)

(4) Given as constant value = 900 psi

(5) Tubing pressure is read from the graph only (Fig. 3.4137) on the top two valves which are spaced using the unloading gradient.

(6) $P_{t(max)}$—from graph (Fig. 3.4137)

(7) $P_d = P_{vc}$—from graph = closing pressure at depth. (Fig. 3.4137).

(8) $P_{vo(max)}$ = casing pressure opposite each valve just as bottom valve opens and is read from the graph (Fig. 3.4137).

(9) P_{vo} as slug passes opposite each valve = casing pressure just as the slug passes by (from Fig. 3.4137)

(10) Determine the correct port sizes to give multi-point injection for all but the top two valves. Having these top valves come open is pointless since maximum slug velocity will have been reached and most of the gas will have been injected by this time.

(a) Determine port size for top two valves using the equation:

$$R = \frac{P_{vo} - P_{vc}}{P_{vo} - P_t}$$

where P_t = unloading tubing pressure. Since these are unloading valves only, use maximum casing pressure at each valve and unloading tubing pressure.

For example, for Valve 1:

(a.1) $$R = \frac{1025 - 955}{1025 - 230} = \frac{70}{795} = .088$$

$A_p = R(A_b)$

For 1½" valve and A_b = 0.77 in.²

(a.2) $A_p = (0.77)(0.088) = 0.06776$ in.²

(a.3) Find equivalent circle diameter

Nearest standard valve size = 5/16" port.

(11) Determine port sizes for the lower six valves which will be valves used for multi-point injection, that is such that each valve will open as the bottom of

TABLE 3.491

SUMMARY TABULATION — MULTI-POINT OR SINGLE POINT INJECTION PRESSURE OPERATED VALVES — CHOKE CONTROL

Valve	Depth	Temp	P_{sc}	P_t Unloading	Max. Tubing press P_t	Press casing (Max)	P_{vo} Slug pass	P_{so}	$P_d = P_{vc}$	Calculated R	A_p	Control port diam. (Nearest)	True R values	P_d 60°F.	P_{tro}	P_{tro} Adjusted
1	2350	128	900	230	670	1025	978	922	955	.088*	.06776	5/16	.0996	830	922	920
2	4275	142	900	354	785	1074	1045	937	1001	.1014*	.0804	5/16	.0996	850	944	945
3	6000	155	900	—	887	1117	1104	955	1042	.286	.22	½	.2562	860	1156	1155
4	6420	158	900	—	912	1128	1118	958	1052	.32	.246	9/16	.3227	860	1270	1270
5	6800	161	900	—	934	1138	1130	960	1060	.357	.275	9/16	.3227	868	1282	1280
6	7120	164	900	—	954	1146	1142	963	1068	.394	.303	5/8	.3990	870	1448	1450
7	7450	166	900	—	973	1154	1153	966	1076	.425	.327	5/8	.3990	870	1448	1450
8	7600	167	900	750 (to open)	982	1158	1158	966	1080	—	—	7/16	.1952	875	1087	1085

*Based on maximum casing pressure of 1025 and 1074, respectively

All other valves set on P_{vo} (slug pass)

$$R = \frac{P_{vo} - P_{vc}}{P_{vo} - P_t}$$

the slug passes on its way to the surface.

The same equation is used, that is:

$$R = \frac{P_{vo} - P_{vc}}{P_{vo} - P_t} \qquad (3.447)$$

However, for these valves we use the value of P_{vo} just as the bottom of the slug passes and maximum tubing pressure for calculating the control port size needed.

For example, for Valve 6:

$P_{vo} = 1142$, $P_{t(max)} = 954$ and, $P_{vc} = 1068$.

Substituting:

$$R = \frac{1142 - 1068}{1142 - 954} = .394$$

$A_p = (.77)(.394) = .303$ and the nearest port size is 5/8"

(12) The bottom valve still utilizes a 7/16" control port since it is opened initially before all other valves and is opened from a static condition. Make it a pilot valve incorporating a 5/8" or 3/4" port as the main seat size.

Summary

Large port sizes are needed to open the valves as the slug passes. At its best, an efficient operation cannot be expected because the port size selections depend upon accurate determinations of pressures in the tubing string as the slug passes. However, since the operating valve at bottom definitely opens first, the opening of the upper valves is not that critical for such a large port (5/8") on bottom. The designer certainly does not want to open a valve into any portion of the liquid slug. This tends to break up the slug and reduces overall efficiency. This design shows some of the difficulties in designing for multipoint injection or in preventing multipoint injection.

For most standard intermittent installations where the unloading gradient has been used to space all the way to bottom, there is a good possibility of injection through one or two valves above the operating valve. The valves are set to open on the unloading pressure in the tubing and the true pressure is much higher as the slug passes by. This additional pressure, depending upon seat size, will probably be enough to open one or more valves above on most installations.

If sufficient gas pressure is available and it is desired that all gas pass through the bottom valve, then the pressure on the bottom valve can be dropped 50-100 psi more. This is commonly done in the field to assure the operator when he has reached the bottom operating valve, as well as assuring single point injection.

Multipoint operations for pressure operated valves become very dependent upon good well information and correct determination of tubing and casing pressure as the slug passes. With large port sizes available in pilot valves, we do not try to use multipoint injection, although Section 3.4352 has illustrated this design.

For "fluid operated valves" which by design must use smaller seat sizes, the ability to use multipoint injection correctly should improve the overall efficiency of the installation. In fact, it almost becomes a necessity in that standard 3/16" or 1/4" ports commonly used on fluid operated valves are not large enough for good intermittent operations by single point injection. Port sizes smaller than 5/16" permit high losses in good pressure across the valve itself and, hence, loss in efficiency.

3.4353 Graphical procedure—pressure operated valves (time cycle control at the surface)

In this case pressure operated unbalanced valves will be used with a 25 psi drop in surface operating pressure between valves.

(1) Prepare a pressure-depth plot on a sheet of standard graph paper as noted in Figure 3.4138.

(2) Mark $P_{to} = 1000$ psi at the top and extend a gas gradient line downwards from this point until reaching the depth of 7600 ft. From Figure 3A.1 in the Appendix, this is 26.2 psi/1000 ft.

(3) Mark the tubing wellhead pressure of 80 psi at the top.

(4) Estimate maximum rate from this well to be 214 B/D by method of Vogel for a solution gas drive well. This rate of 214 B/D is for a flowing pressure equal to zero, which is impossible. The maximum production rate will be less than 200 B/D and probably between 150 B/d (when $P_{wf} = 750$ psi) and 200 B/d. (see Volume I Chap I of this series).

(5) Select an unloading gradient for spacing the valves based on the production rate from the well. The unloading gradients increase as the production rate increases. A safe unloading gradient of 0.064 psi/ft is selected from Fig. 3F.2 in the Appendix based on a final production rate of 200 B/D. Construct this unloading gradient line

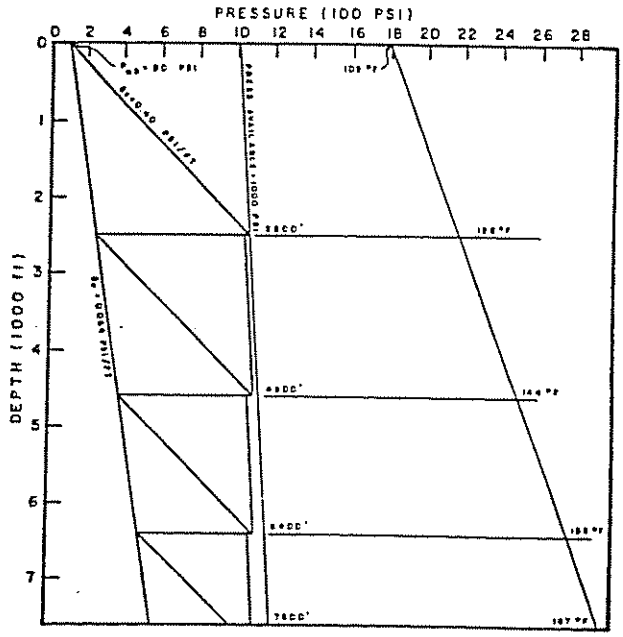


Fig. 3.4138 Design for Pressure Operated Valves (25 PSI Drop in Surface Opening Pressure between Valves)

from the wellhead pressure of 80 psi to the depth of 7600 ft.

(6) Draw in the kill fluid gradient line of 0.40 psi/ft starting from the wellhead pressure of 80 psi and extending it downward until intersecting the 1000 psi operating gas pressure line. This locates Valve 1 at 2500 ft with 1050 psi available. This allows 50 psi safety factor.

(7) Take a 25 psi drop in pressure at the surface and draw in several gas pressure lines downward from the surface for surface operating pressures of 975, 950, 925, and 900 psi. Obtain gas column weights from Fig. 3A.1, Appendix 3A. These values are 25.5, 25, 24.2, and 23.6 psi/1000 ft, respectively.

(8) From the depth of 2500 ft obtained in Step 6, draw a horizontal line to the left until intersecting the unloading gradient line of 0.064 psi/ft.

(9) From the point obtained in Step 8, extend the 0.40 psi/ft kill fluid line downwards until intersecting the 975 psi operating gas line. This locates Valve 2 at 4600 ft.

(10) Repeat the procedure locating Valve 3 at 6400 ft and finding valve 4 to come out lower than the total depth of 7600 ft. Locate Valve 4 at 7600 ft. We now have four valves with surface operating pressures and depths located as follows:

Valve no.	P_{so}	Depth
1	1000	2500
2	975	4600
3	950	6400
4	925 (change to 900)	7600

Since time cycle control is at the surface and single point injection is desired, adjust Valve 4 to open at 900 psi at the surface; re-adjust the pressure settings to 1000, 975, 950 and 900 psi. Lower Valve #4 another 25 psi to prevent valve interference. When the casing pressure is much higher than the tubing pressure, a lower-operating valve pressure could be selected such as P_t + 200 = 750 + 200 = 950 psi at the valve. This eliminates excessive differential across the operating valve. Differentials greater than 300-400 psi may shorten the life of the valve seat, depending upon how clean the gas is. Also ΔP's greater than 400 psi are unnecessary and could induce excessive gas breakthrough.

(11) Determine the gas volume necessary to lift one cycle of liquid. For 900 psi at the surface, the valve opening pressure at 7600 ft is 1080 psi. Gas volume required from Fig. 3G.8, Appendix 3G = 7200 scf/cycle. (Series 3G.1 through 3G.16 are available in the Appendix.)

(12) Determine the casing pressure change necessary to furnish 7200 scf. From Fig. 3H.6 in Appendix 3H this is 74 psi.

(13) Select the tubing load to be lifted per cycle. Without more detailed information, use $P_t = \frac{1}{2} \overline{P}_R = \frac{1}{2}(1500) = 750$ psi.

(14) Select the valve control port size for time cycle control at the surface. As an approximation, place ½ the required pressure change of 74 psi in the valve itself; that is, a valve that has 74/2 = 37 psi spread insures that gas control will be from the surface. For choke control at the surface, all 74 psi change would be placed in the valve. Since the valve opening pressure at depth is 1080 psi, set the valve closing pressure to be 1080 − 37 = 1043 psi. Round this value off to 1040 psi. Since the valve opening pressure is 1080, the valve closing pressure is 1040 psi, and the tubing load is 750 psi, solve for the valve port size.

Use the equation:

$$R = \frac{P_{vo} - P_{vc}}{P_{vo} - P_t} = \frac{1080 - 1040}{1080 - 750} = .1212$$

For a 1½" diameter valve with a bellows area of 0.77 in², this corresponds to a port diameter of between ⅜ and 5/16. Select the lowermost size because it is safer to have less spread than too much for time cycle control. This value can also be obtained from Fig. 3I.1 in Appendix 3I; it falls between 5/16 and ⅜. Again select the 5/16" control port. This represents the control port of the operating gas lift valve. In conjunction, select a large main seat size (½", ⅝", or ¾"). Use a ⅝" main seat size in this case. The 5/16" port has an R value of .0996 and a 1-R value of .9004. If the surface opening pressure is set at 900 psi, which gives a corresponding valve opening pressure of 1080 psi and a tubing load of 750 psi, solve for the exact closing pressure from the equation:

$$P_d = P_{vc} = P_{vo}(1 - R) + P_t R \qquad (3.448)$$
$$= 1080(.9004) + 750(.0996)$$
$$= 1047 \text{ psi}$$

Valve spread = 1080 − 1047 = 33 psi

This is fine since time cycle control is used at the surface.

(15) Select the control port sizes and type of valves for all other valves. Since the top two valves will be used for unloading only, use 5/16" ported valves and another ⅝" main seated pilot valve with a 5/16" control port for Valve 3. In the event the well is better than anticipated, Valve 3 could serve as the operating valve; if so, it should have a large port size.

(16) Construct in the temperature line between 109°F. at the surface and 167°F. at the bottom on Figure 3.4138. Find the temperature at each valve.

(17) Prepare a summary tabulation as noted in Table 3.492 and determine the shop calibration for the valves (test rack opening pressures).

An explanation of the various columns is as follows:

(1) Valve no.
(2) Depth—from Fig. 3.4138
(3) Temp—from Fig. 3.4138
(4) P_{so}—given
(5) P_{vo}—from Fig. 3.4138
(6) For Valves 1, 2, and 3 taken from Fig. 3.4138. For Valve 4 selected as 750 psi. (See step #13)

(7) See Step 14
(8) See Step 15
(9) $P_d = P_{vc} = P_{vo}(1 - R) + P_t R$ (3.448)
(10) $P_{sc} = P_{vc}$—Gas column weight
(11) Convert P_d at temp to P_d at 60°F. (See Fig. 3D.1 or Table 3D.1 in Appendix D)
(12) Solve for $P_{tro} = \frac{P_d}{1 - R}$ (3.449)
(13) Final calibration—round off values in column 12 to nearest 5.

(18) Analyze the final valve settings and check valve interference.

Check to see if any of the upper valves will open as the slug passes by. The maximum pressure underneath the slug can be approximated from Table 3.489 as:

$$P_{t\ max} = P_{vo} - .10\ P_{vo} = 1080 - .10(1080) = 972 \text{ psi}$$

Assume this tubing load is at Valve 3 when the slug passes, and solve for the pressure to open Valve 3.

$$P_{vo} = \frac{P_{vc} - P_t R}{1 - R} = \frac{1048 - 972(.0996)}{.9004} = 1056 \quad (3.450)$$

1056 psi is needed at Valve 3 to open it when the slug passes. If 1080 psi is at Valve 4 when it opens, 1051 psi is opposite Valve 3, which is not insufficient pressure to open it. The only unaccounted factor is the casing pressure over-build to furnish the necessary gas. Recall that a 74 psi change is needed and only 33 psi is obtained from building from P_{vc} = to P_{vo} = 1080.

The casing pressure needed to furnish the gas is $P_{vc} + 74 = 1047 + 74 = 1121$, or a 41 psi over-build by the time cycle controller to furnish sufficient gas. When the bottom of the slug passes Valve 3, the system has used $(74)\left(\frac{1200}{7600}\right) = 12$ psi and the casing pressure will be 1121 − 12 = 1109 psi at Valve 4 and 1080 at Valve 3. This shows that Valve 3 will open near the bottom of the slug since 1056 is sufficient pressure and 1080 psi exists. This further assumes that all gas enters the casing in about one minute. It is possible to either restrict gas entry to a longer period or lower the pressure of Valve 4 another 25 psi. Since friction due to gas has been neglected and will serve to reduce the casing pressure, leave the setting as is. Valves 1 and 2 need not be checked since the casing pressure will have dropped sufficiently so that they cannot open when the slug passes.

The reasoning in taking a 50 psi drop in surface opening pressure between Valves 3 and 4 is evident and a decision to take a 75 psi drop could now be made. If the standard 25 psi drop was taken, then Valve 3 would definitely have opened into the slug and this can be detrimental. There is no damage in opening the valve after the slug passes, but opening into the slug can cause more fallback.

3.4354 Analytical procedure—pressure operated valves (time-cycle control at the surface) (25 psi drop in pressure between valves)

The analytical procedure for spacing gas lift valves is the same as the graphical procedure except the weight of the gas column is generally neglected. By trial and error solution, the gas column weight could be included and then would give the same results as the graphical procedure. Common practice is to neglect the gas column weight; this procedure is favorable because of the additional safety factor.

Procedure:

(1) Estimate the maximum rate possible to be 214 B/D for ($P_{wf} = 0$) by method of Vogel. From Fig 3F.2, Appendix based on a maximum production rate of 200 B/D, select an unloading gradient of .064 psi/ft.
(2) Allow a 25 psi drop in pressure between valves and neglect the gas column weight.
(3) Set the valves for surface opening pressures of 1000, 975, 950, 925, and 900 psi, this is also assumed to be the pressure at each valve by neglecting gas column weight.
(4) Space Valve 1 assuming the well is loaded with "kill fluid" up to the surface.

$$D_{v_1} = \frac{P_{so} - P_{wh}}{G_s} = \frac{1000 - 80}{.40} = 2300 \text{ ft} \quad (3.451)$$

This compares to 2500 ft when the gas column pressure is accounted for.

(5) Space Valve 2 by the following formula:

$$D_{v_2} = D_{v_1} + \left[\frac{P_{vo_1} - P_{wh} - G_g(D_{v_1})}{G_s}\right] \quad (3.452)$$

Note the use of P_{vo}, of 1000 psi here instead of

TABLE 3.492 SUMMARY TABULATION—PRESSURE CHARGED VALVES—TIME CYCLE CONTROL

Valve no.	Depth	Temp	P_{so}	P_{vo}	P_t	Control port size	Main seat size	$P_d = P_{vc}$	P_{sc}	P_d 60°F	P_{tro} 60°F	Adj. P_{tro}
1	2500	128	1000	1066	240	5/16	5/16	984	923	855	950	950
2	4600	144	975	1092	375	5/16	5/16	1021	910	852	946	945
3	6400	158	950	1110	490	5/16	5/8	1048	898	860	955	955
4	7600	167	900	1080	750 (selected)	5/16	5/8	1047	872	850	944	945

$P_d = P_{vc} = P_{vo}(1 - R) + P_t R$
$R = .0996$
$1 - R = .9004$

P_{vo_2} of 975 psi because the 1000 psi is available until reaching Valve 2.

$$D_{v_2} = 2300 + \left[\frac{1000 - 80 - (0.064)(2300)}{0.40}\right]$$
$$= 2300 + 1932 = 4232 \text{ ft}$$

(6) Valve 3 is spaced in the same manner except P_{vo_2} is used.

$$D_{v_3} = D_{v_2} + \left[\frac{P_{vo_2} - P_{wh} - G_u(D_{v_3})}{G_s}\right] \quad (3.453)$$
$$= 4232 + \left[\frac{975 - 80 - 0.064(4232)}{0.40}\right]$$
$$= 4232 + 1560 = 5792 \text{ ft}$$

(7) Find depth to Valve 4

$$D_{v_4} = 5792 + \left[\frac{950 - 80 - 0.064(5792)}{0.40}\right] \quad (3.4534)$$
$$= 5792 + 1248 = 7040 \text{ ft}$$

(8) $$D_{v_5} = 7040 + \left[\frac{925 - 80 - 0.064(7040)}{0.40}\right] \quad (3.455)$$
$$= 7040 + [986]$$
$$= 8026$$

(9) Prepare a tabulation showing pressures and depths of the five valves and make any necessary adjustments.

Valve no.	Pressure	Depth
1	1000	2300
2	975	4232
3	950	5792
4	925	7040
5	900	8026

Adjust Valve 5 to the depth of the well, which is 7600 ft. Leave Valve 4 at 7040 ft in the event it should be necessary for operation. As in the graphical procedure, take an additional 25 psi drop on the last valve, setting it for a pressure of 875 psi.

The adjusted valve settings and depths will appear as follows:

Valve No.	Pressure	Depth
1	1000	2300
2	975	4232
3	950	5792
4	925	7040
5	875	7600

Depending upon any safety factor desired, the pressures can now be considered as pressures at the valve and the lower surface operating pressures calculated. Or they may be considered as surface opening pressures.

(10) Determine the temperature at each gas lift valve.

Temp top = 109°F.
Temp 7600 ft. = 167°F.

$$\text{Temp gradient} = \frac{167 - 109}{7600} = .00763°\text{F./ft}$$

Temp Valve 1 = 109 + (.00763)(2300)
= 109 + 18 = 127°F.

In the same manner, the temperatures at Valves 2, 3, 4, and 5 are 141, 153, 163 and 167°F.

(11) Prepare a summary tabulation as in Table 3.493, considering the valve pressures to be surface opening pressures. Also, see steps 11 through 15 in Section 3.4353 to determine the gas volume to be approximately 7200 scf/cycle, requiring a 74 psi change in casing pressure. Select a control port size of 5/16" which has a spread of approximately 35 psi. Select a tubing load for the last valve in the same manner as Section 3.4353 to be ½ $\overline{P_R}$ = ½(1500) = 750 psi.

(12) Explanation of columns in Table 3.493.
(1) Valve numbers
(2) Depth—from Step 9 in procedure
(3) Temp—Step 10 in procedure
(4) P_{so}—given
(5) P_{vo}—calculated from gas weight
(6) P_t—calculated from unloading gradient
(7) Control port size—same as previous procedure
(8) Main seat size—use large size for operating valve
(9) $P_d = P_{vc} = P_m(1 - R) + P_tR$
(10) P_{sc}—calculated using gas column pressure Fig. 3A.1 (Appendix)
(11) P_d at 60°F—from Fig. 3D.1
(12) $P_{tro} = \frac{P_d}{1 - R}$
(13) Adjust to nearest value of 5.

3.4355 Graphical procedure—pressure-operated valves (choke control at the surface) (25 psi drop in surface opening pressure between valves)

The procedure for spacing is the same as for time cycle control at the surface. The only difference in the design procedure is the operating valve must have sufficient spread to furnish the gas volume required per cycle. The control of gas will be through a choke at the surface; hence, we have no time cycle controller that opens and furnishes the necessary gas in a short period of time. The casing is used as a storage chamber for gas so when the operating valve opens, the casing pressure drops to the closing pressure of the valve which, in turn, furnishes sufficient gas to lift out the accumulated slug of liquid in the tubing.

Procedure:

Prepare a figure the same as Figure 3.4138 for time-cycle control.

(1) Steps 1 through 10 are the same as in Section 3.4353 and will have the following valve settings and depths:

Valve No.	P_{so}	Depth
1	1000	2500
2	975	4600
3	950	6400
4	900	7600

(2) Steps 11, 12, and 13 are the same as in Section 3.4353, finding a gas volume of 7200 scf/cycle requiring a casing pressure change of 74 psi and lifting a tubing load of 750 psi.

When a spread at bottom of 74 psi is selected,

TABLE 3.493
SUMMARY TABULATION—PRESSURE OPERATED VALVES—ANALYTICAL PROCEDURE

Valve no.	Depth	Temp	P_{so}	P_{vo}	P_t	Control port size	Main seat size	$P_d = P_{vc}$	P_{sc}	P_d 60°F	P_{tro} 60°F	P_{tro} final calib 60°F
1	2300	127	1000	1060	227	5/16	5/16	977	922	855	950	950
2	4232	141	975	1083	351	5/16	5/16	1010	910	860	955	955
3	5792	153	950	1095	451	5/16	5/16	1031	895	857	952	950
4	7040	163	925	1095	531	5/16	5/8	1039	879	849	943	945
5	7600	167	875	1050	750	5/16	5/8	1020	850	830	922	920

$P_d = P_{vc} = P_{vo}(1 - R) + P_tR$
$R = 0.0996$
$1 - R = 0.9004$ for 1½" standard valve and 5/16" port

casing pressure needs to change 74 psi at bottom conditions. Due to changes in gas column pressure, the spread at the surface will be less than the spread at the bottom. For example, for 74 psi change at 7600 ft, there is approximately 62 psi change at the top of the well. For the average intermittent well at depths of 6000-8000 ft, subtract 10 psi from the bottom hole spread to obtain the surface spread between opening and closing pressures. For this example expect a surface spread of approximately 74 − 10 = 64 psi.

To obtain an average spread in the casing to give the correct gas volume would require 74 + 5 = 79 psi at the bottom giving 69 psi at the top for an average of $\frac{79 + 69}{2} = 74$ psi, which is the required spread to furnish the necessary gas. For choke control, safely select a control port that will give the bottom-hole calculated spread or even greater. However, previous examples have used a port size to give a spread equal to or less than the calculated bottom spread. This assures safety by not using too much gas.

(3) Choke control requires a valve that will incorporate the spread to give a 74 psi change in casing pressure which furnishes 7200 scf of gas. The valve opening pressure is 900 + gas column weight = 1080 psi. The valve closing pressure should be 1080 − 74 = 1006 psi, and the tubing load is 750 psi.

(4) Select the valve control port size that will satisfy the conditions of $P_{vo} = 1080$, $P_{vc} = 1006$, and $P_t = 750$. Use the equation:

$$R = \frac{P_{vo} - P_{vc}}{P_{vo} - P_t} = \frac{1080 - 1006}{1080 - 750} = .2242$$

A 1½" diameter standard valve with a bellows area of 0.77 in² corresponds to a port diameter of 15/32 where $R = \frac{A_p}{A_b}$ or $A_p = (0.2242)(0.77) = 0.1726$ in². This value can be obtained from Fig. 3I.1 in the Appendix, which shows the port size to fall between 7/16" and ½" standard sizes. Standard procedure is to select the next smallest size, 7/16" in this case, and check the spread on it.

(5) Check the spread on the 7/16" diameter valve. From Table 3.88 the R value is 0.1952 and 1 − R is 0.8058. For $P_{vo} = 1080$ and $P_t = 750$

$$P_{vc} = P_{vo}(1 - R) + P_t(R) = 1080(.8058) + 750(.1952) = 870 + 146 = 1017$$

This 7/16" control ported valve has a spread of 1080 − 1017 = 63 psi compared to 74 psi required. The 63 psi spread will supply (840)(7.6) = 6384 scf instead of the 7200 scf needed. Use the 7/16" port size since the gas requirement volume is based on water and oil is being lifted. Also, the surface injection choke is passing gas into the casing during the lift cycle and, in reality, the system will use more than the 6384 scf indicated.

If additional gas volume is needed, add the additional gas on a time cycle controller by-pass at the surface when the valve opens, since the amount is relatively small. It is also possible to request the manufacturer to make the exact size of 15/32 to give the exact spread of 74 psi. It would also be possible to go to the next largest standard size of ½" but that would use too much gas each cycle.

Further, the designer can allow the casing pressure to build to 1017 + 74 = 1091 and permit the valve to open on a lower tubing load of 706 psi instead of waiting for 750 psi. In all likelihood, the designer will not find the optimum condition at a tubing load of 750 psi. We can adjust down on P_t by upping P_{vo}, but we cannot adjust upwards on P_t unless supplying more gas by time cycle control. Again, use a pilot valve with a ⅝" to ¾" main port size. Use 5/16" ported standard valves for the three unloading valves.

(6) Determine the temperature at each valve as described noted in Fig. 3.4138.

(7) Prepare a summary tabulation as in Table 3.494.

If a lighter tubing load must be lifted, the gas lift valve will have more spread and may use excessive gas.

The operating valve for "choke control" should be retrievable by wireline because of difficulty determining the optimum tubing load even with good well information. For example, in this case we selected ½ $\overline{P}_R$. There should be a well fill-in rate curve. In many instances, $\overline{P}_R$ is unknown. This is another reason why

TABLE 3.494
SUMMARY TABULATION GRAPHICAL PROCEDURE—PRESSURE OPERATED VALVES—CHOKE CONTROL

Valve no.	Depth	Temp	P_{so}	P_{vo}	P_t	Control port size	Main seat size	$P_d = P_{vc}$	P_{sc}	P_d 60°F	P_{tro} 60°F	Adjusted P_{tro}'
1	2500	128	1000	1066	240	5/16	5/16	984	923	855	950	950
2	4600	144	975	1092	375	5/16	5/16	1021	910	852	946	945
3	6400	158	950	1110	490	5/16	5/16	1048	898	860	955	955
4	7600	167	900	1080	750	7/16	5/8	1017	848	830	1030	1030

$P_d = P_{vc} = P_{vo}(1 - R) + P_tR$

the majority of wells are placed on time cycle control.

Another procedure is to test the well initially on time cycle control with a low spread valve and then switch to choke control.

Some control is possible by over-building on the casing pressure allowing the valve to open on a lighter tubing load and, hence, supply sufficient gas.

3.4356 Pressure operated valves—graphical procedure; constant valve closing pressure (time cycle control and choke control)

A widely-used procedure of spacing for intermittent lift is to set all valves for the same surface closing pressure. This holds true for standard pressure operated valves and pilot valves, which have constant closing pressures. This design can be used for either choke or time cycle control at the surface, making certain that a sufficient pressure is available to open the valve. This setting also works very well for dual installations. For example, if one side of a dual is on continuous flow and the other on intermittent flow, the closing pressures on the valves for the intermittent side can be set equal to or slightly above the opening pressure for the continuous flow valves. This permits gas to pass continuously through the continuous flow valves, and the intermittent valves can be opened by raising the casing pressure. The intermittent operating valves can be opened and closed without interfering with the continuous flow operations provided that a differential or fluid opened section is run with the continuous flow valves.

Also, where more than 3-4 valves are needed for unloading the well, this method makes it possible to reach the operating valve with more pressure.

If this design is used on single installations, it is possible to purposely drop the operating valve an additional 25 to 100 psi lower in surface closing pressure to determine when this valve is reached (normally the bottom valve and normally at the bottom of the well).

If the available operating pressure and unloading pressure is the same, then set the surface closing pressures 100 psi less than the operating pressure for 7" casing and 150 psi less for 5½" casing for both 2⅜" O.D. and 2⅞" O.D. tubings. Then check the casing pressure build-up necessary to furnish the gas volume needed and make any adjustments necessary on the original choice of surface closing pressures.

Procedure:

Given—same data as example problem of Section 3.4352.

(1) Prepare a pressure-depth plot on a sheet of standard graph paper as in Fig. 3.4139.
(2) Mark P_{so} = 1000 psi at the top and extend a gas gradient line downwards from this point until reaching the depth of 7600 ft. From Fig. 3A.1 in the appendix, this is 26.2 psi/1000 ft.
(3) Mark the tubing wellhead pressure of 80 psi at the top.
(4) Estimate maximum rate from this well to be 214 B/D by method of Vogel for a solution gas drive well. This rate of 214 B/D is for a flowing pressure equal to zero, which is impossible. Maximum production rate will be less than 200 B/D and probably between 150 B/D (when P_{wf} = 750 psi) and 200 B/D.
(5) Select an unloading gradient for spacing the valves. Select an unloading gradient based on the production rate from the well. The unloading gradients increase as the production rate increases. Select a safe unloading gradient of .064 psi/ft from Fig. 3F.2 in Appendix 3F based on a final production rate of 200 B/D. Construct this unloading gradient line from the

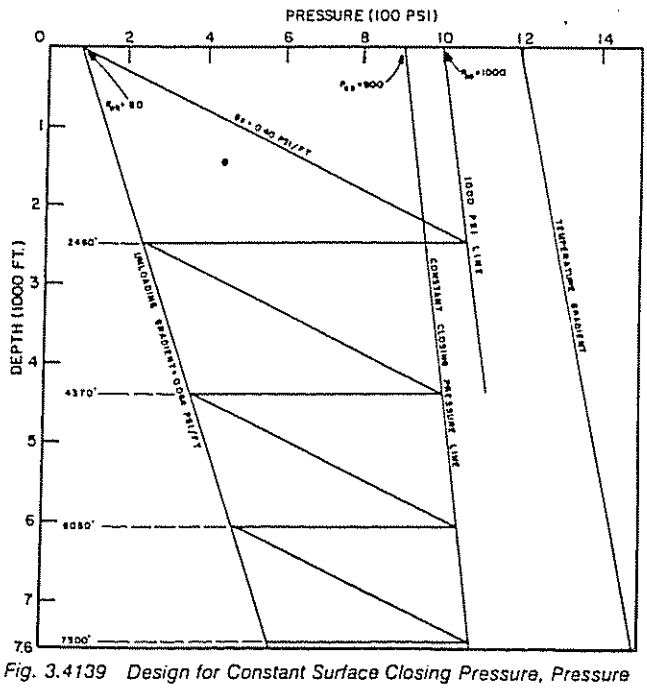


Fig. 3.4139 Design for Constant Surface Closing Pressure, Pressure Operated Valves

wellhead pressure of 80 psi to the depth of 7600 ft.

(6) Assume a constant surface closing pressure of 900 psi (Available pressure minus 100 psi for 7" casing) and mark this pressure at the top. Extend this pressure downwards, including the gas column weight to the bottom of the well. (23.6 psi/ft).

(7) Draw the kill fluid gradient line of 0.40 psi/ft, starting from the wellhead pressure of 80 psi and extending it downwards until intersecting the 1000 psi operating gas pressure line. This locates valve #1 at 2460 ft.

(8) From the depth of 2460 ft obtained in Step 7, draw a horizontal line to the left until intersecting the unloading gradient line of .064 psi/ft.

(9) From the point obtained in Step 8, extend the 0.40 psi/ft kill fluid line downward until intersecting the 900 psi closing operating gas line. This locates Valve 2 at 4370 ft.

(10) Locate all other valves by the same procedure spacing between the .064 psi/ft unloading gradient line and the 900 psi surface closing pressure line. This locates additional valves at 6050 and 7500 ft.

(11) Adjust valve spacings. Depending upon bottom hole conditions, that is, the position of the packer in regard to the center of the perforations, move Valve 4 either upward or, as is the case here, slightly downwards. Exercise care in moving it downward. Since this will be a fairly good intermittent well, adjust Valve 4 to 7200 ft and place another Valve 5 at 7600 ft (400 ft apart).

The adjusted spacing will then be as follows:

Valve no.	Depth
1	2460
2	4370
3	6050
4	7200
5	7600

(12) Adjust the surface closing pressure of the last valve (#5) downward to eliminate valve interference. Reduce its surface closing pressure to 850 psi (50 psi less than all other valves). Note that for duals this would not be recommended; but for a single completion I would recommend this adjustment. The adjusted surface closing pressures would then be:

Valve No.	P_{sc}
1	900
2	900
3	900
4	900
5	850

(13) Determine the gas volume needed per cycle. To use Fig. 3G. 8 (Appendix) the operating valve opening pressure is needed. Assume a surface opening pressure of 900 psi and we find the opening pressure at 7600 ft to be 1080 psi. Gas volume required = 7200 sct/cycle from Fig. 3G.8.

(14) Determine the casing pressure change necessary to supply 7200 scf to be 74 psi from Fig. 3H.6 (Appendix.)

(15) Depending upon whether time cycle control or choke control is used, set the opening pressure of the valve. The valve closing pressure at depth is 850 psi at the surface plus (22.2) (7.6) = 1019 psi at 7600 ft.

For choke control $P_{vo} = 1019 + 74 = 1093$ psi
For time cycle control $P_{vo} = 1019 + 74/2 =$ 1056 psi

(16) Determine port sizes. Select the tubing load to ½ P_R = ½ (1500) = 750 psi

(a) Select the control port size for choke control

$$R = \frac{P_{vo} - P_{vc}}{P_{vo} - P_t} = \frac{1093 - 1019}{1093 - 750} = .2157$$

For a 1½" O.D. valve, the port size nearest to this = 7/16", which has an R value of 0.1952 and 1 − R = 0.8048. For a closing pressure of 1019 psi the opening pressure for a 7/16" port is found as follows:

$$P_{vo} = \frac{P_{vc}}{1-R} - P_t\left(\frac{R}{1-R}\right) \qquad (3.450)$$
$$= \frac{1019}{.8048} - 750\left(\frac{.1952}{.8048}\right) = 1266 - 182$$
$$= 1084 \text{ psi}$$

This gives a ΔP of 1084 − 1019 = 65 psi instead of the required 74 psi. This can be accounted for by increasing the opening pressure to 1093 psi and allowing the valve to open on a tubing load of 714 psi instead of 750 psi. Find this as follows:

$$P_t = \frac{P_{vc}}{R} - P_{vo}\left(\frac{1-R}{R}\right) \qquad (3.456)$$
$$= \frac{1019}{0.1952} - 1093\left(\frac{0.8048}{0.1952}\right)$$
$$= 5520 - 4506 = 714 \text{ psi}$$

In addition, some gas is passing when the valve is open. Use a main seat size of 5/8". By allowing the casing pressure to increase to 1093 psi at the valve, the tubing pressure required to open the valve is reduced to 714 psi for a 7/16" x 1½" valve.

(b) Select the control port size for time cycle control.

$$R = \frac{P_{vo} - P_{vc}}{P_{vo} - P_t} = \frac{1056 - 1019}{1056 - 750} = .1209$$

For a 1½" valve the nearest control port size is a 5/16" port having an R value of .0996 and 1 − R = .9004. Find the true opening pressure for P_t = 750 psi to be 1049 psi in the following manner:

$$P_{vo} = \frac{P_{vc}}{1-R} - P_t\left(\frac{R}{1-R}\right)$$
$$= \frac{1019}{.9004} - 750\left(\frac{.0996}{.9004}\right)$$
$$= 1132 - 83 = 1049$$

TABLE 3.495
SUMMARY TABULATION CONSTANT VALVE CLOSING PRESSURE—CHOKE CONTROL AND TIME CYCLE CONTROL

Valve no.	Depth	Temp	P_{sc}	$P_{vc} = P_d$	P_t	Control port	Main seat	P_{vo}	P_{so}	P_d 60°F	P_{TRO} 60°F	Final calib	Remarks
1	2460	128	900	958	238	5/16	5/16	1037	975	830	922	920	For Both Time Cycle & Choke Control
2	4370	142	900	1003	360	5/16	5/16	1074	965	850	944	945	For Both Time Cycle & Choke Control
3	6050	155	900	1043	467	5/16	5/16	1107	955	860	955	955	For Both Time Cycle & Choke Control
4	7200	164	900	1070	541	5/16	5/16	1128	942	870	966	965	For Both Time Cycle & Choke Control
5	7600	167	850	1019	750	7/16	5/8	1084	905	825	1025	1025	Settings for choke control
*5	7600	167	850	1019	750	5/16	5/8	1049	875	825	916	915	Settings for time cycle control only

*For time cycle control

$$P_{vo} = \frac{P_{vc}}{1-R} - P_t\left(\frac{R}{1-R}\right)$$

For 5/16" Port
R = .0996
1 − R = .9004
TEF = .1106

For 7/16" Port
R = .1952
1 − R = .8048
TEF = .2425

This gives a $\Delta P = 1049 - 1019 = 30$ psi, which is less than the recommended maximum value of 74/2 = 37 psi. This is satisfactory for time cycle control in that the gas is supplied from surface storage through a time cycle controller. Use a main seat size of 5/8".

(17) Select the control port sizes for all other valves.
  (a) For choke control, select the top four unloading valves with 5/16" ports and use standard valves. The last valve has a 7/16" control port x 5/8" main seat size and is a pilot valve.
  (b) For time cycle control, select the top unloading valves with 5/16" ports using a standard valve. The bottom valve is a pilot valve and has a 5/16" control port by 5/8" main seat size. This requires a 1½" O.D. valve.

(18) Construct the temperature gradient line and find the temperature at each valve.

(19) Prepare a summary tabulation as noted in Table 3.495 for choke control and time cycle control. An explanation of the columns in the table is given as follows:

Explanation of Table 3.495—Choke control explanation

Column No.

(1) Valve Numbers
(2) Depth From Figure 3.4139
(3) Temp From Figure 3.4139
(4) P_{sc} Given as 900 psi for four valves and 850 psi for last valve
(5) $P_d = P_{vc} = P_{sc}$ + gas column weight
(6) P_t = taken from Figure 3.4139 except for last valve where $P_t = \frac{1}{2} P_R = 750$ psi
(7) Control port selection. See steps 16 and 17.
(8) Main seat size. See steps 16 and 17
(9) P_{vo} Calculate from $P_{vo} = \frac{P_d}{1-R} - P_t(TEF)$
(10) $P_{so} = P_{vo}$ − Gas Column weight
(11) P_d 60°F From Figure 3D.1 (Appendix)
(12) $P_{tro} = P_d 60°F/1 - R$
(13) Final calibration Round off P_{tro} values to nearest 5

Note that the valve settings are the same for both time cycle and choke control except for Valve 5. For Valve 5 the spread is different for choke control (7/16" control port) compared to time cycle control (5/16" control port).

3.4357 Design procedure fluid operated valves for multipoint injection

Work the same example problem using fluid operated valves for multipoint injection. The fluid valve is commonly used for multipoint injection because of the restricted entry port required. In Chapter 2's discussion on fluid valves, note the maximum port size commonly used is ¼" and, at the most, 5/16". This assures closure of the valve once the slug has been produced and the tubing pressure reduced.

To prevent using too many valves for unloading purposes, two to three standard pressure operated valves on top simplify the unloading process and probably save three or four valves. The important valves for multipoint injection are the lowermost valves. Refer to various steps used in Section 3.4352, which was for the

design of pressure operated valves for multipoint injection. An example will be worked for both dome charged and spring charged valves.

3.43571 Nitrogen charged fluid valve

Procedure:

(1) Prepare a pressure depth plot as in Fig. 3.4140.
(2) Mark P_{so} = 1000 psi at top and draw in the gas gradient line (26.2 psi/1000') (See Figure 3A.1 Appendix.)
(3) Select the operating casing pressure to be 1000 − 100 = 900 psi and draw in the gas gradient line (23.6 psi/1000 ft. from Fig. 3A.1 Appendix).

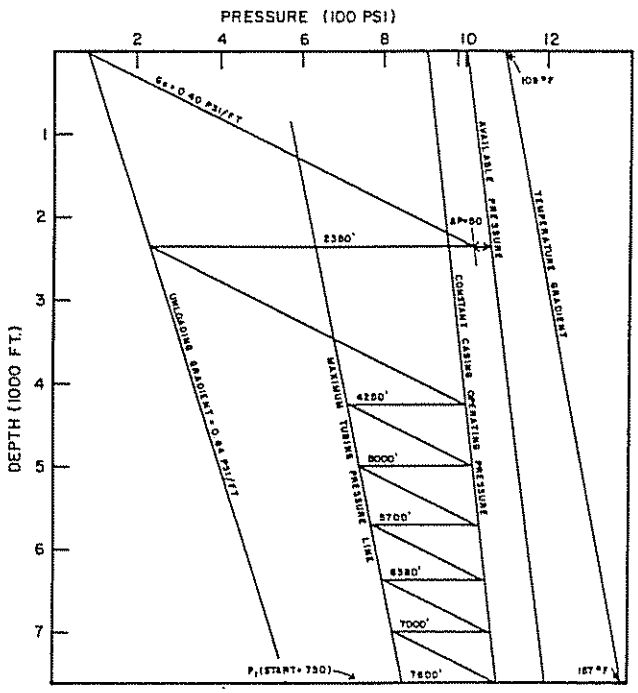


Fig. 3.4140 Design for Multipoint Injection Nitrogen Charged Fluid Operated Valves

(4) Select an unloading gradient of .064 psi/ft and draw in from 80 psi at the top.
(5) Determine gas volume per cycle = 7200 scf/cycle.
(6) Determine casing pressure change to be 74 psi to furnish the necessary gas volume.
(7) Select tubing load to be lifted off bottom as ½(1500) = 750 psi.
(8) Select valve port sizes to be ¼" which are typical for fluid operated valves. For 2" tubing and depending upon spacing distance, we may use smaller choke sizes such as 8, 9, or 10/64 or a standard 3/16" port.
(9) Construct the maximum tubing pressure line.

$$\Delta P \text{ Static} = 1080 - 750 = 330 \text{ psi}$$

(See Table 3.489, Section 3.4352, Step 9.)

The gas pressure placed underneath the slug is estimated to be $P_{vo} - .25\ P_{vo} = 810$ psi.

The reason for the low pressure is the small port size. To make certain the valve does not open into the slug, a minimum of 100 psi greater than the tubing pressure is recommended which would be 750 + 100 = 850 psi. This is equivalent to a velocity of between 500-600 ft/min, which is to be expected for the smaller port sizes. Assume 7% liquid loss per 1000 ft of travel, given a total loss of (7)(7.6) = 53.2% loss and the length of slug at the top is 767 ft. See Step 9, Section 3.4352, to find total press at 767 ft = 565 psi. Draw a straight line between 565 psi at 767 ft and 850 psi at the bottom, and extend it to the surface.

(10) Space the gas lift valves as follows:
(a) For Valve 1, space (use pressure charged valve) in the conventional manner by constructing the "kill fluid" gradient line of .40 psi/ft from P_{wh} = 80 psi to the available pressure line minus 50 psi to allow for safety factor. Refer to Fig. 3.4140. Set this valve for 980 psi surface opening pressure. (Valve 1 only will be a pressure operated valve.)
(b) Space one more valve using the .064 psi/ft gradient line and the gas pressure line of 900 psi at the surface. Valve 2 will be a fluid operated valve.

TABLE 3.496
SUMMARY TABULATION
MULTI-POINT INJECTION — FLUID OPERATED VALVES

(1) Valve no.	(2) Depth	(3) Temp of	(4) P_t Max	(5) P_{casing} surface	(6) P_{csg} Valve	(7) P_t Unload	(8) Port size	(9) P_{to} (Tubing)	(10) $P_d = P_{vc}$	(11) P_c 60°F	(12) P_{TRO}	(13) Final calib.	
1*	2350	128	631	980	1040	230	¼	230	988	860	918	920	Press oper. valve
2	4250	141	710	900	1000	--	¼	710	728	625	667	665	Fluid valve
3	5000	147	742	900	1018	--	¼	742	760	640	684	685	Fluid valve
4	5700	153	771	900	1035	--	¼	771	788	660	705	705	Fluid valve
5	6380	158	800	900	1051	--	¼	800	816	675	721	720	Fluid valve
6	7000	162	825	900	1065	--	¼	825	840	690	737	735	Fluid valve
7	7600	167	850	900	1080	750 (to open)	¼	750	771	627	670	670	Fluid valve

$$P_{to} = \frac{P_d}{1-R} - P_{casing}\frac{R}{1-R}$$

$$P_d = P_{vc} = P_{to}(1-R) + P_{casing}(R)$$

P_{to} Tubing press to open
For a ¼" × 1½" valve
R = 0.0638
1 − R = 0.9362

*Valve 1 only is a pressure operated valve to permit wider spacing.

(c) Space all other valves based on the maximum tubing pressure line and the casing operating pressure line. (All these valves will be fluid operated valves.)

(11) Draw in the temp gradient line and select the temperature at each valve.

(12) Prepare a summary tabulation as in Table 3.496.

Explanation of Columns in Table 3.496

Column no.

(1) Valve numbers

(2) Depth—from Figure 3.4140

(3) Temp—from Figure 3.4140

(4) $P_{t(max)}$—from Figure 3.4140
Determine tubing pressure for first valve from the unloading gradient line. All others are determined from max tubing pressure line except operating Valve 7.

(5) Surface casing pressure = 900 psi for all valves except #1, which is a pressure operated unloading valve to permit wider spacing.

(6) $P_{casing\ at\ valve} = P_{casing\ surface} + gas\ column\ weight$

(7) P_t unload—used on first valve only and P_t of 750 psi is selected for last valve.

(8) Port Size—selected as ¼" for all valves.

(9) P_{to} = tubing opening press for multipoint injection = max tubing press as slug passes for all but #1 valve.

(10) $P_d = P_{vc}$
Solve for Valve 1 (press valve) as $P_d = P_{vo}(1 - R) + P_tR$
For all other valves (fluid valves), use

$$P_d = P_{to}(1 - R) + P_{csR}R \qquad (3.457)$$

(11) $P_{d_{60°F}}$ - See Figure 3D.1 (Appendix) or Table 3D.1 (Appendix)

(12) $P_{TRO} = \dfrac{P_d}{1 - R}$

(13) Final calibration = value rounded off to nearest multiple of 5.

3.43572 Design example for multipoint intermittent lift—spring charged valve

Multipoint intermittent gas lift is accomplished with the same type fluid valve used for the continuous flow design. As in continuous flow, the valve port must be small enough to drop pressure across the valve resulting in the valve sensing only tubing pressure when in the open position.

The operating valve opens after the formation has fed in sufficient liquid to trip this valve. At this time the pressure placed under the slug propels it upward toward the next valve. This is a critical period in multipoint intermittent lift. The port size and differential of the operating valve must be designed to pass enough gas to move the slug to the next valve at a sufficient average velocity. Although a minimum average velocity of about 1000 ft/min should be attained for the entire tubing length, results from field installations indicate average velocities as low as 300 ft/min are sufficient over the first few hundred feet of lift, where depths are relatively deep and the distance between valves is small. The volume of gas required to lift the slug from the operating valve to the next valve up is calculated with Equation 3.460.

$$PV = ZnRT \qquad (3.458)$$

$$n = PV/ZRT \qquad (3.459)$$

$$\text{Volume of gas required} = n \cdot 379\ \text{SCF/lb-mole} \qquad (3.460)$$

where:

P = Average pressure under the slug when it reaches the next valve up from the operating valve, psi.
T = Average temperature, °R
R = Gas constant = 10.73 ft³ psi/lb-mole · °R
Z = Compressibility factor
V = Capacity of tubing between operating valve and the next valve up, cu-ft
One lb-Mole = 379 SCF@14.7 psi and 60°F.

Equations 3.459 and 3.460 show that increasing the tubing capacity, V, by increasing the tubing size or the distance between valves results in a proportional increase of the required gas volume to maintain the same pressure conditions. Tubing size, distance between valves, and casing pressure at the operating valve will affect the selection of the correct port size and differential to be used in the design. As tubing size or distance between the valves increases, the port size or differential must be increased proportionally to ensure sufficient gas passage from the operating valve.

The pressure just beneath the liquid load remains higher than pressure at the top of the slug until the slug reaches a point near the surface. Brown showed this in an actual well test. A pressure of 620 psi was placed under a 350 psi load at 4072 ft with a 7/16 inch ported pressure valve. This is a 270 psi differential. The maximum pressure under the slug remained above the load weight of 350 psi until the slug reached a depth of about 1000 feet.

This high pressure under the slug causes multipoint injection in large ported pressure valve installations also, especially when the valves are closely spaced. This is desirable since it decreases the time required for a specific annular drawdown and the gas is input below the slug. Since the slug immediately begins to lose mass in the form of fallback, a cutting of the slug will not occur if the valve trip pressures are set on a constant differential.

The uppermost valves could be set at higher differentials than the lower valves since the fluid load decreases due to fallback as the slug moves up the hole. This is generally done, since it will reduce the number of valves required to unload and lift a low BHP intermittent installation.

The same example problem will be worked for the spring charged valves except that a 0.468 psi/ft kill fluid gradient will be used and a tubing pressure of 120 psi is used for unloading.

Given Data:

Depth = 7600 ft (mid perf)
$\overline{P}_R$ = 1500 psi
q_L = Max (all oil)

Tubing size = 2⅜ in. O.D.
1.995 in. I.D.
P_{wh} = 120 psi (during unloading)
J = 0.2 B/D/psi
Oil gravity = 41 °API
Gas gravity = 0.7
BHT = 167 °F @ 7600 ft
Separator pressure = 80 psi
P_{ko} = 1000 psi
Pso = 900 psi
Well kill fluid = 9 #/gal
Well loading = All the way to top
Well unloading = To be unloaded to 80 psi separator

Spacing Procedure: (Figure 3.4141)

(1) Plot the gas gradient lines for kickoff and operating pressures.

(2) Construct a 300 psi differential line. Do this by extending a line parallel to the 900 psi operating pressure line from 600 psi at 0 ft to the perforation depth.

(3) Construct the kill fluid gradient line of 0.468 psi/ft. Do this by extending a line from zero to 468 psi at 1000 ft.

(4) Extend the 0.468 psi/ft line from 120 psi at 0 ft until it intersects the 1000 psi gradient line. This locates the first valve at 2000 ft.

(5) Select the tubing or fluid opening pressure (P_{vo_T}) and the port size for the first valve. This P_{vo_T} should be high enough to ensure that the top valve will not open prior to the passage of the entire slug. Therefore, the anticipated slug weight must be calculated considering fallback from the operating valve to the top valve. Do this as follows:
Assume the slug will lose 7% per 1000 ft of lift.

Extend a 0.355 psi/ft (for 40 °API oil) line from the SBHP at perf depth until it intersects the 0 psi axis. Assume that the intersection of this line with the 300 psi differential line is the depth of lift. This is approximately 5300 ft. Use 5500 ft as the depth of lift from the operating valve to the surface.

Feet of lift to the top valve = 5500 − 2000 = 3500
Loss factor = LF = 0.07(3.5) = 24.5%
P_{vo_T} = (Pc@5500′ − Diff − Psep) − (Pc@5500′ − Diff − Psep)LF + Psep
(neglecting dry gas gradient above the slug)

P_{vo_T} = (985 − 300 − 80) − (985 − 300 − 80)(0.245) + 80 = 537 psi

Set the P_{vo_T} of the first valve at 537 psi. This will be used as the transfer point to the next valve. All valves of this design will use ⅛₄th inch ports.

(6) Draw a horizontal line from the point located in step 4 to the point located in step 5.

(7) Draw a line parallel to the 0.468 psi/ft line from the transfer point of step 5 until it intersects the 900 psi surface operating pressure line. This locates valve 2 at 2840 ft.

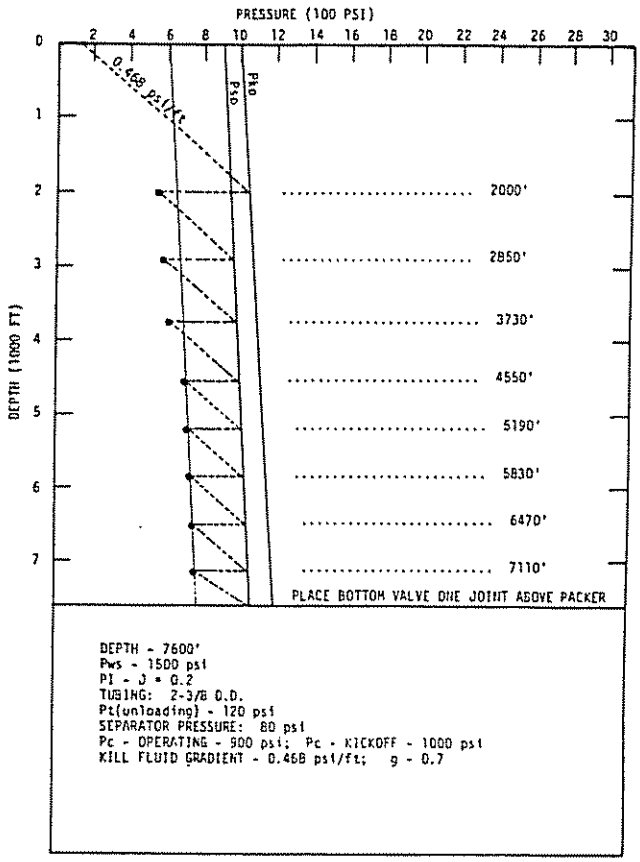


Fig. 3.4141 Intermittent Spacing for Fluid Operated Valves

(8) Calculate the P_{vo_T} of the valve 2 as in step 5.

Feet of lift to the second valve = 5500 − 2850 = 2650 ft
Loss Factor = LF = 0.07(2.65) = 18.55%
OP = (985 − 300 − 80) − (985 − 300 − 80)(0.1855) + 80 = 573 psi

Set P_{vo_T} of the second valve at 573 psi.

(9) Locate valve 3 in the same manner as valves 1 & 2. Space the remaining valves from the 300 psi differential line as shown in Figure 3.4141. All valves could be spaced in the same manner as the top three. However, using differential spacing for the lower valves assures additional safety in the design. The depth of the last two valves may be adjusted somewhat depending on the depth of production packer placement during completion.

(10) Check to see if the P_{vo_T} of the bottom valve is greater than a 0.065 psi/ft gradient plus separator pressure. Do this to ensure proper closure of the bottom valve.

Depth of bottom valve = 7300 ft
0.065(7300) + 80 = 554 psi
P_{vo_T} = 720 psi OK!

Experience shows that 0.065 psi/ft plus separator pressure is a safe criteria to use in determining minimum P_{vo_T}'s of fluid valves in 2 inch tubing under field conditions. Successful installa-

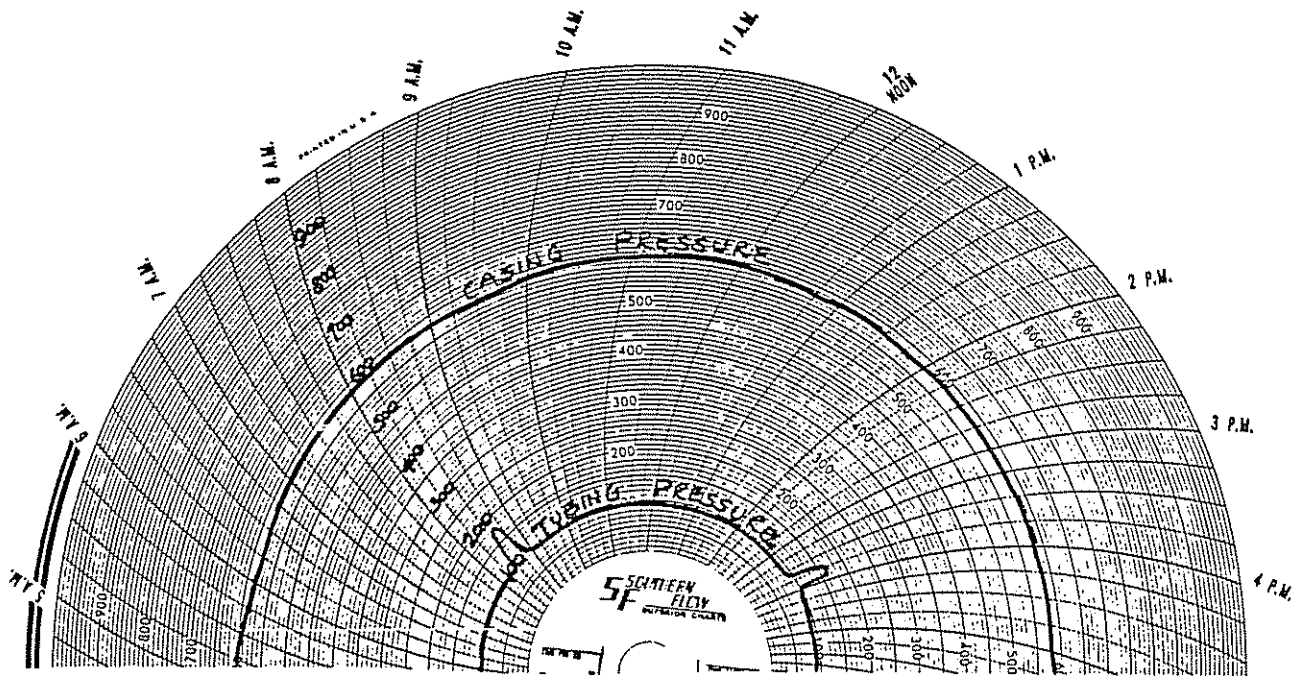


Fig. 3.4142 Two Pen Recorder Chart of Intermittent Multipoint Injection Gas Lift Installation (Low $\bar{P}_R$)

tions have been designed with P_{vo_T}'s as low as 0.05 psi/ft. If the SBHP or operating casing pressure of a well is so low that it is necessary to set an OP less than 0.05 psi/ft plus separator pressure, the operating valve should be a pressure valve or a combination fluid open, pressure closed valve.

Figure 3.4142 is a two-pen pressure recording of a low BHP, multipoint fluid valve intermittent installation. The operating valve was an $^{8}/_{64}$ ported fluid valve set at approximately 6500 ft with the P_{vo_T} set on a 0.04 psi/ft gradient plus separator pressure. Notice the fallback gradient held the valve open for about 20 minutes after the slug was produced each cycle. Obviously, the P_{vo_T} was set to low. This valve should be replaced by a pressure valve or a combination valve.

(11) Prepare a summary tabulation (Table 3.497) showing final calibration of all valves (multipoint - spring charged).

3.4358 Intermittent opti-flow design procedure

A design procedure was proposed by Axelson, Inc., U.S. I, and the procedure was patented for intermittent gas lift operations for either choke or time cycle control.[21] It is particularly suited for intermittent wells when neither the depth of the operating valve nor the productive capacity of the well is known. It is, therefore, a spacing procedure that will permit the well to seek the lowest operating point automatically.

It is based on the premises that if Valve 4 is the operating valve, then Valve 5 should be the operating valve, and after reaching Valve 5, then Valve 6 should be the operating valve. This concept continues until reaching a point whereby the feed-in of well fluids is sufficiently rapid to prevent operation from the next

TABLE 3.497—SUMMARY TABULATION
FLUID VALVE—INTERMITTENT DESIGN

Valve no.	Depth (ft)	Casing pressure at depth (psig)	P_{vc} (valve closing pressure) (psig)	Tubing pressure transfer point (psig)	Port size (64th)	1 – R	R	P_{vo_T} (psig)	P_{tro}
1	2000	934	554	537	8	.9583	0.0417	537	578
2	2850	948	589	573	8	.9583	0.0417	573	615
3	3730	963	604	588	8	.9583	0.0417	588	630
4	4550	977	689	676	8	.9583	0.0417	676	719
5	5190	988	700	687	8	.9583	0.0417	687	730
6	5830	998	711	698	8	.9583	0.0417	698	742
7	6470	1009	722	709	8	.9583	0.0417	709	753
8	7110	1020	733	720	8	.9583	0.0417	720	665

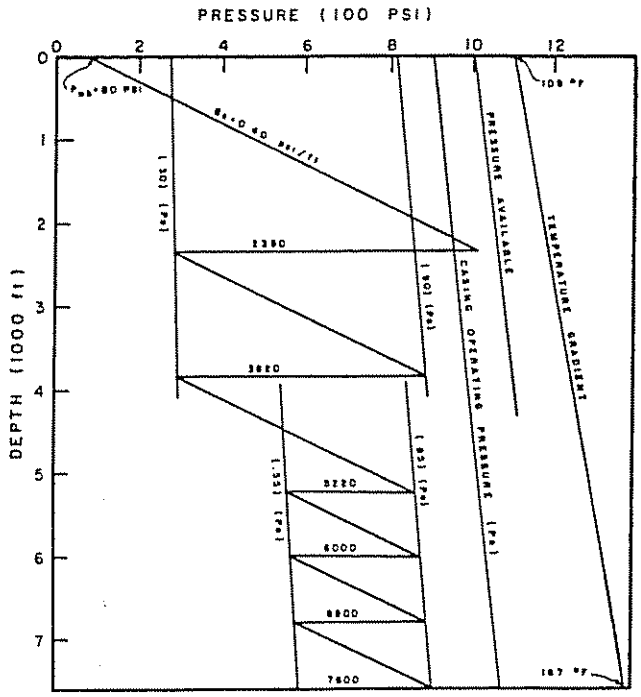


Fig. 3.4143 Opti-Flow Intermittent Design Installations

lowermost valve. Spacing is such to permit this to occur automatically.

Spacing is based on a tubing load taken as a percentage of the casing pressure, and the original patent called for a 90% – 30% unloading spacing and an 85% – 55% spacing in the operating range. For example, if the casing pressure is 1000 psi at Valve 5, the valve opens on a tubing load of (.55)(1000) = 550 psi and Valve 6 has (.85)(1000) = 850 psi tubing load (neglecting gas weight). Therefore, with a tubing load of 850 psi and casing pressure of 1000 psi, Valve 6 should begin being the operating valve and will do so depending upon well feed-in characteristics.

Although the procedure shows a slight increase in spacing distance moving down the well, this is off-set by additional gas column pressure. As a suggestion, it is possible to deviate from the original procedure and place all valves the same distance apart in the operating range, or even slightly closer together, which is consistent with increased unloading gradients from deeper depths.

Procedure: (For either choke control or time cycle control)

The same problem will be worked by this procedure.

(1) Prepare an appropriate scale as noted on Fig. 3.4143.
(2) Mark 1000 psi casing pressure at the top and extend downwards accounting for gas column weight. (Fig. 3A.1 Appendix).
(3) Mark 1000 – 100 = 900 psi operating casing pressure at the top and extend downwards accounting for gas column weight.
(4) Check the static fluid level if $\overline{P}_R$ is known. For this problem this is $7600 - \frac{1500}{.40} = 3850$ feet from the top. The reason for checking this is that we can use a wider spacing for this unloading distance of 3850 ft before the well starts any feed-in of fluids.
(5) Use a percent loading for the unloading valve spacing of 90% – 30%.

(.90) (900) = 810 psi
(.30) (900) = 270 psi

Mark both 810 psi and 270 psi at the surface and extend downward to approximately 4000 ft (3850 static fluid level) taking into account their respective gas column pressures. If desired, we may construct the 270 psi line parallel to the 810 psi line, which prevents increased spacing down the well.

(6) Mark 80 psi wellhead pressure at zero depth and from this point extend the kill fluid gradient line of 0.40 psi/ft until reaching the 1000 psi casing pressure line minus 50 psi for safety in unloading. This locates valve #1 at 2350 ft.
(7) Extend a line from this depth of 2350 ft horizontally to the left until intersecting the (0.30) P_c line.
(8) From the point found in step 7, extend the .40 psi/ft kill fluid line until intersecting the (0.90) (P_c) line. This locates valve #2 at 3820 ft.
(9) Check this against the static fluid level of 3850 feet and change to the 85% - 55% spacing for valves in the possible operating range.
(10) Use an 85% - 55% loading for valves in the operating range.

(.85) (900) = 765 psi
(.55) (900) = 495 psi

Mark both 765 and 495 psi at the surface and extend downwards, including respective gas column weights until reaching the bottom of the well. Again, it is possible to exercise the option of drawing the 495 psi line parallel to the 765 psi line eliminating increased spacing down the well.

(11) From the point located in step 8 (3820 ft) extend the kill fluid gradient line downwards until intersecting the (0.85)P_c line locating valve #3 at 5220 ft.
(12) Continue spacing between the 0.85 P_c line and the 0.55 P_c line until reaching 7600 ft (bottom), locating valves at 6000, 6800, and 7600 ft.
(13) Construct the temperature gradient line with 109°F at the surface and 167°F at 7600 ft. Determine the temp at each valve.
(14) Determine the gas volume needed per cycle. For a surface operating pressure of 900 psi, the opening pressure at 7600 ft is 1080 psi and the gas volume required per cycle is 7200 scf (Fig. 36.8 Appendix 3G).
(15) Determine the casing pressure change necessary to furnish 7200 scf to be 74 psi (Fig. 3H.6 Appendix 3H).
(16) Note the minimum tubing load on the last valve to be 600 psi.
(17) Solve for the control port size needed for choke control.

TABLE 3.498
SUMMARY TABULATION—CHOKE CONTROL
INTERMITTENT OPTI-FLOW DESIGN PROCEDURE

Valve No.	Depth	Temp	Control port	P_{so}	P_{vo}	$P_{t(min)}$	$P_{t(max)}$	$P_d = P_{vc}$	P_{sc}	P_d 60°F	P_{po}	Main port	ΔP Valve	ΔP Surface
1	2350	127	3/8	975	1035	285	--	927	875	810	945	3/8	108	100
2	3820	138	3/8	950	1046	300	--	939	855	805	940	3/8	107	95
3	5220	149	3/8	900	1023	565	865	957	842	810	945	5/8	66	58
4	6000	155	3/8	900	1042	575	880	975	842	810	945	5/8	67	58
5	6800	161	3/8	900	1060	585	900	992	842	810	945	5/8	68	58
6	7600	167	3/8	900	1080	600	915	1011	842	815	951	5/8	69	58

All valves are set to open on minimum tubing pressures and surface operating casing pressures as indicated

$P_d = P_{vo}(1 - R) + P_t R$

3/8" Control port on pilot valve—5/8 main seat
R = .1434
1 – R = .8566

$P_{vo} = 1080 \qquad P_{vc} = 1080 - 74 = 1006$
$P_t = 600$ (minimum tubing load)

$$R = \frac{P_{vo} - P_{vc}}{P_{vo} - P_t} = \frac{1080 - 1006}{1080 - 600} = .154$$

Select a 3/8" control port with a 5/8" main seat size on a pilot valve.

For a 3/8" port:

R = .1434
1 – R = .8566

This is the nearest standard size to a calculated R value of 0.154. These values are for a 1½" O.D. valve having a bellows area of 0.77 in.^{2} For time cycle control select a valve with a spread = ½(74) = 37 psi. Solving for R:

$$R = \frac{1080 - 1043}{1080 - 600} = .07708$$

Use a 5/16" control port with a 5/8" main seat size on a pilot valve. For a 5/16" port (nearest standard size to R = .07708)

R = .0996
1 – R = .9004

These values are for a 1½" O.D. valve having a bellows area of 0.77 in^2.

(18) Prepare a summary tabulation as in Table 3.498. Valves #1 and #2 are set to open on the lower minimum tubing loads of .30(P_c). To prevent opening when the slug passes, raise their surface opening pressures to 975 and 950 psi, respectively. Since they are unloading valves only, this will offer no problem.

All other valves are set to open on the minimum tubing load of 55 P_c and the surface casing pressure of 900 psi.

Summary and Discussion

Note the difference between ΔP at the valves and ΔP at the surface. For good choke control, the spread needed to furnish sufficient gas may be slightly low. For example, Valve 6 shows a spread at the valve of 69 psi and a surface spread of 58 psi, giving an average spread of 63.5 psi. Recall that 74 psi was needed to supply the necessary gas volume. If the choke at the surface does not furnish the additional gas needed during that time when the valve is open, overbuild the casing pressure to 910 psi at the surface and permit the valve to open on a slightly lower tubing load. Therefore, there should be no problem in the final regulation of the well.

Note also that the valve closing pressure on the last four valves is a constant value at the surface, which makes this method somewhat analogous to setting valves at a constant surface closing pressures. The main difference is the closer spacing, which permits an automatic seeking of the next lowermost valve from the operating valve. Column 5 also shows a constant surface opening pressure on the last four valves.

Time cycle control could be used with this installation as is. Or it would be possible to change to the 5/16" control ported pilot valves, which give a lower spread, but offer no problem since the additional gas would be furnished from the surface. In fact, the opti-flow design is often used in conjunction with a time cycle controller.

3.4359 Design for combination fluid-opened pressure-closed valves

3.43591 Introduction

In reality, this is a fluid operated valve. However, its unusual combination of having a pressure operated valve on top to close the differential section permits use of a large ported valve for what, in fact, is a fluid opened valve. It incorporates a spring in the fluid section and may incorporate a charged bellows or spring charge in the upper section. This valve is particularly suited for duals since it offers the additional safety feature of incorporating two valves in one.

One of the first decisions for this type of valve is whether to use "choke control" or time cycle control. A change in casing pressure of 74 psi is needed to furnish the necessary gas of 7400 scf/cycle. This type of valve permits more flexibility in adjusting this change in casing pressure to supply the correct gas volume.

For example, the bottom valve should open on a tubing load of 750 psi. This is accomplished by setting the spring differential between casing and tubing pressure. Assuming choke control, the bellows section of the valve can be set for the correct spread of 74 psi, but the

spring section will not open until there is 750 psi in the tubing even though the upper section of the valve is open.

To assure more flexibility, set the bellows section for ½(74) = 37 psi spread, then permit over-build on the casing up to 74 psi change thereby supplying sufficient gas. The limiting port size is the passage through the differential section, which is a maximum of ½ inch.

Therefore, there is no need to use a port size greater than ½" on the upper section for either a standard or pilot valve. To control spread, run a 5/16" control port and ½" main seat size on the upper section which allows flexibility for time cycle control. Choke control can still be used by overbuilding the operating pressure before the differential section opens.

It may be desirable to run a standard pressure operated or pilot valve on bottom instead of the combination valve. The pilot valve can be opened against variable tubing loads. Once the combination valve is placed on bottom, a predetermined tubing load is needed to open it, depending upon the spring differential. This particular type of valve is limited to a spring setting of about 400-450 psi in the fluid section for good operations.

3.43592 Design procedure I for the combination pressure-closed tubing pressure-opened valve (choke control or time cycle control)

(1) Prepare a pressure depth scale as in Fig. 3.4144.
(2) Mark P_{so} = 1000 at top and draw in the gas gradient line. (Fig. 3A.1 - Appendix)
(3) Mark P_{sc} = 1000 − 100 = 900 psi at the top = constant valve closing pressure and draw in the gas gradient line. (Fig. 3A.1)
(4) Estimate the casing change in pressure to supply the needed gas volume of 7400 scf/cycle to be 74 psi (use 70 psi).
(5) Valve opening pressure at bottom = 1080 + 70 = 1150 psi. Draw in the gas gradient line giving 960 psi at the top. This is the surface casing pressure to supply sufficient gas for a valve closing pressure at bottom of 1080 psi and is also used in spacing the valves.
(6) Draw in the maximum tubing pressure line as described previously using $P_{t(max)}$ at bottom = P_{vo} − .20P_{vo} for ⅜" port and $\Delta P \cong 300$ psi. $P_{t(max)}$ = 1150 − .20(1150) = 920. (Refer to Table 3.489) The pressure at 767 ft is 565 psi as found previously. Construct a line between 920 psi at bottom and 565 psi at 767 ft and extend on to the surface.
(7) From 80 psi at the wellhead extend downwards the kill fluid gradient of .40 psi/ft locating Valve 1 at 2350 ft at the intersection of the casing pressure line to open the valves.
(8) Space the valves between the maximum tubing pressure line and the casing pressure to open the valves.
(9) Set the bottom valve. There is an option here of using a standard pressure operated valve or a combination valve. For this example, use a combination valve which incorporates a spring and use the opening tubing pressure of 750 psi and the necessary casing pressure to supply the needed gas.

$$\text{Spring} = P_{vo} - P_{\text{tubing load}} = 1150 - 750 = 400 \text{ psi}$$

This assures that the differential section of Valve 8 (bottom valve) will open first but cannot pass gas until the bellows section opens. Set the bellows section to open at 1115 psi (1080 closing + ½ of spread), which gives some flexibility. Time cycle control can be used and the casing can be overbuilt to 1150, which will not allow the spring section to open until reaching 750 psi in the tubing. Remember that the spring section opens on differential between casing and tubing pressure. For example, if the spring is set at 400 psi, the spring section opens whenever the differential between casing and tubing pressure is 400 psi. At 1115 psi casing pressure, the differential section opens at 1115 − 400 = 715 psi tubing pressure.
(10) Set the springs on all other valves to be [$P_{\text{bellows open}} - P_{t(max)}$].

For example, on Valve 5 located at 5900 ft.:
Pressure to open bellows section = 1075 psi
Maximum tubing press = 835 psi
Spring = [$P_{\text{bellows open}} - P_{t(max)}$] = 1075 − 835 = 240 psi
(11) Construct the temperature line and find the temperature at each valve.
(12) Set the bellows section on each valve for $P_{t(max)}$ and the line labeled casing pressure to open valves.
(13) Solve for correct port sizes.
(14) Prepare a summary tabulation as noted in Table 3.499.

Note: Since the flexibility exists to overbuild on the casing pressure and wait for the tubing load to open the valve, it is unnecessary to place the correct spread in the bellows section of the valve. Therefore, this same design will work satisfactorily for both time cycle control and choke control at

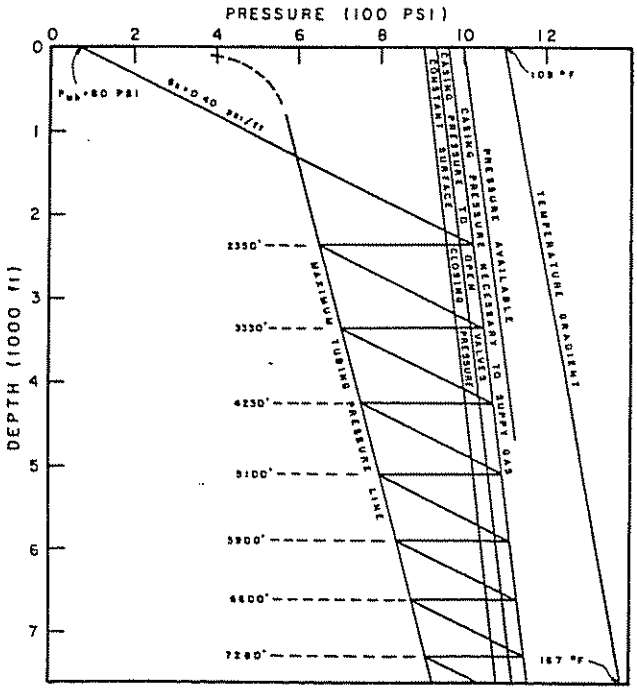


Fig. 3.4144 Design for Combination Fluid Opened Pressure Closed Valves

TABLE 3.499
SUMMARY TABULATION COMBINATION PRESSURE CLOSED—FLUID OPENED VALVE
CHOKE CONTROL OR TIME CYCLE CONTROL

Valve no.	Depth	Temp	P_{sc}	$P_d = P_{vc}$	P_{vo}*	P_{so}	P_{casing} Max (at Valve)	P_{surf} Casing Max	P_{tubing} Max	P_{tubing} Start	Spring diff**	P_d 60°F	P_{tro}	Control port	Main port
1	2350	127	900	955	986	930	1015	956	651	—	365	830	922	5/16	5/16
2	3330	134	900	979	1011	930	1041	957	701	—	340	850	944	5/16	5/16
3	4230	141	900	1000	1033	930	1065	959	747	—	320	848	942	5/16	5/16
4	5100	148	900	1020	1053	930	1087	959	792	—	295	852	946	5/16	5/16
5	5900	154	900	1039	1073	930	1107	959	833	—	275	862	957	5/16	5/16
6	6600	159	900	1056	1090	930	1125	959	869	—	255	864	960	5/16	½
7	7280	165	900	1072	1107	930	1141	959	904	—	235	870	966	5/16	½
8	7600	167	900	1080	1115	930	1150	960	920	750	400	875	972	5/16	½

*P_{vo} set at P_{vc} + ½ spread
**Spring values rounded off to nearest (5)

5/16" control port
R = 0.0996
1 − R = 0.9004

the surface, that is, placing approximately ½ of the spread in the bellows section of the valve.

(15) Explanation of columns in Table 3.499.
- (1) Valve numbers—self explanatory
- (2) Depth—from Fig. 3.4144
- (3) Temp—from Fig. 3.4144
- (4) P_{sc} = valve set from given data
- (5) $P_{vc} = P_d = P_{sc}$ + gas column pressure
- (6) P_{vo}—valve set to be P_{vc} + ½ of spread at each position, taking 70 psi spread at bottom and 60 psi spread at top
- (7) $P_{so} = P_{vo}$ − gas column pressure
- (8) $P_{casing\ maximum}$ = required casing pressure to furnish gas volumes needed = P_{sc} + (full spread)
- (9) $P_{surface\ casing\ maximum} = P_{casing\ max}$ − gas column press
- (10) $P_{tubing\ max}$ − from Fig. 3.4144
- (11) $P_{tubing\ max\ start}$ = for operating valve only = 750 psi
- (12) Spring = $P_{casing\ max}$ (step 8) − $P_{t(max)}$
- (13) P_d 60°F—see Fig. 3D.1, Appendix 3D
- (14) $P_{tro} = \dfrac{P_d}{1 - R}$

3.43593 Design procedure II for combination pressure closed tubing pressure-opened valves

Another procedure used in the field is described in a publication by Davis, Thrash and Canalizo; their design example is presented here:[23]

Given data:

Depth of well = 5100 ft
Packer depth = 5050 ft
Normal available gas pressure = 700 psi
Temporary kick-off pressure = 800 psi
Low productivity well (less than 100 B/D)
Well loaded to surface with kill fluid of 0.50 psi/ft
Formation temperature = 130°F.
Surface operating temperature = 80°F.
Gas gravity = 0.65
Valves to be set at 80°F. in the shop
Tubing size = 2⅜" O.D.
Casing size = 5½" − 17#/ft
Separator pressure = 50 psig
Unload to separator for all valves

Procedure: (Refer to Figure 3.4145)

(1) Plot depth on vertical scale (5100 ft) vs. pressure on horizontal scale (1000 psi).
(2) Plot (P_{ko} − 50) at surface (800 − 50 = 750 psi).
(3) Obtain gas column weight from Fig. 3A.1 (Appendix 3A) of 18 psi/1000 ft and extend 750 psi down the hole.
(4) Plot a design casing pressure of 50 psi less than surface operating pressure (700−50 = 650 psi). Extend this pressure down the hole (Refer to Fig. 3A.1).
(5) Select spring pressures to be 300 psi on each valve except 350 psi for the bottom valve. This allows the bottom operating valve to lift a lighter load with less chance of the slug opening upper valves as it passes. Subtract the spring pressures of 300 psi from the pressure in Step 4 (650−300 = 350 psi) and extend this down the hole parallel to the 650 psi line.
(6) Plot the tubing pressure of 50 psi at 0 depth.
(7) Plot the kill-fluid gradient of .50 psi/ft from the tubing pressure of 50 psi.
(8) Extend this line until it intersects the (P_{ko} − 50 = 750 psi) line. This locates Valve 1 at 1450 ft.
(9) Extend a horizontal line back to the left until it intersects the 350 psi line.
(10) From this point, draw a line parallel to the 0.50 psi/ft line until it intersects the design casing pressure line of 650 psi.
(11) Continue this procedure until all valves have been located at 1450, 2050, 2650, 3250, 3850, 4450, and 5050 ft.
(12) Draw in the geothermal gradient line from 130°F. at 5100 ft to 80°F. at the surface.
(13) Tabulate the temperature at each valve to be 94, 100, 106, 112, 118, 123, and 129°F.
(14) Determine the casing pressure at depth for each

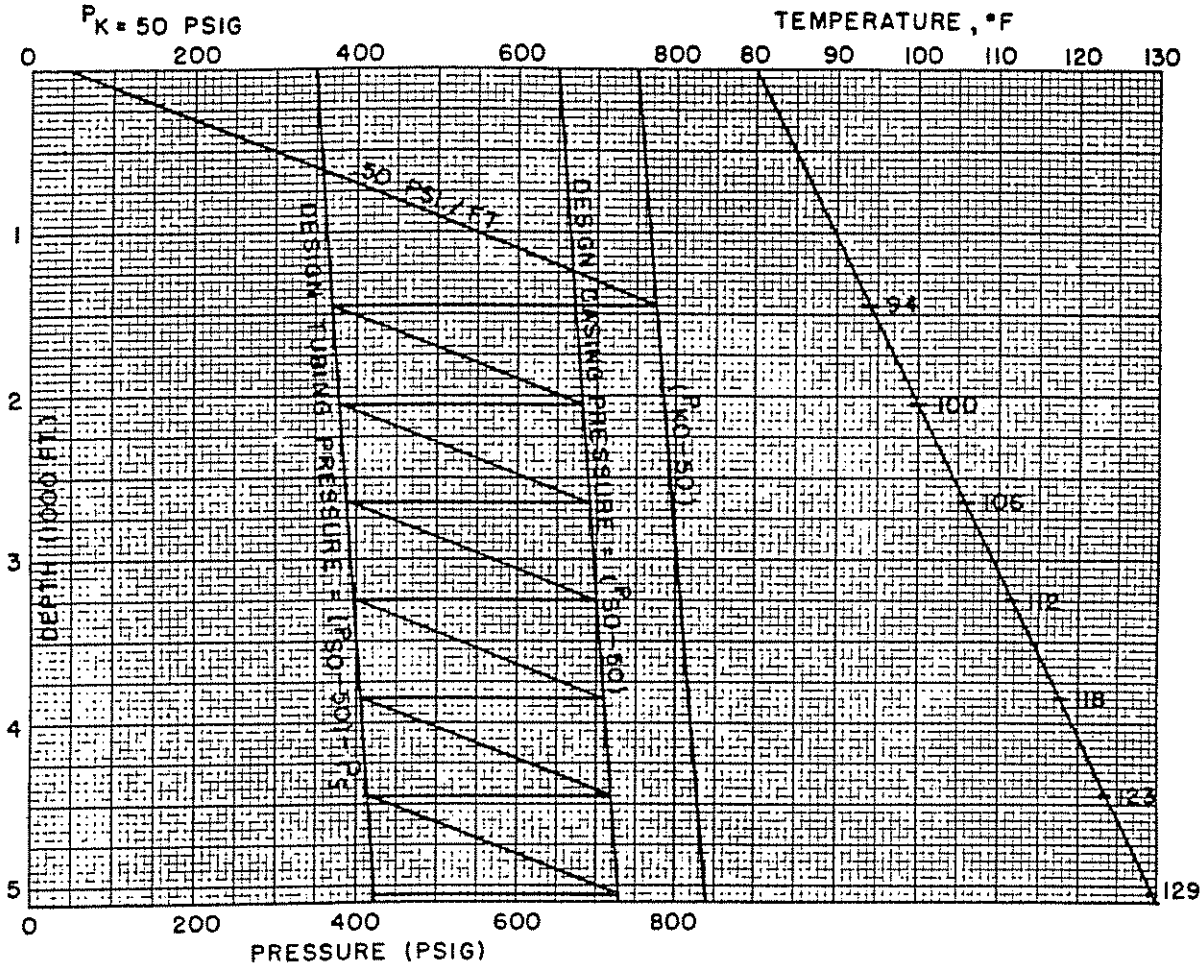


Fig. 3.4145 Graphical Intermittent Flow Installation for Combination Pressure Closed Fluid Opened Valves (Courtesy Otis Engineering Corp.)

valve with 650 psi surface pressure. These values are 672, 682, 691, 700, 710, 719, and 728 psig, respectively.

(15) Subtract 50 psi from each valve opening pressure at depth and obtain values of 622, 632, 641, 650, 660, 669, and 678 psi, respectively.

(16) Determine the opening tubing pressures of each valve by subtracting the spring pressures of each valve from the pressures of Step 14. These values are found to be 372, 382, 391, 400, 410, 419, and 378 psi, respectively.

(17) From Fig. Series Number 3G in the Appendix, determine the gas volume necessary per cycle to be 3250 scf.

(18) From the Fig. Series Number 3H, determine the annular reduction necessary to be 94 psi.

(19) Allow a 50 psi over-build in the valve giving 94 − 50 = 44 psi spread for the valve.

(20) Determine the valve closing pressures of the bellows section of the operating valve by deducting 44 psi from (P_{vo} − 50) as found in Step 15. This is 678 − 44 = 634 psi. Use the same procedure on all other valves and note closing pressures of 578, 588, 597, 606, 616, 625, and 634 psig, respectively. These pressures also represent the dome charge at operating temperatures.

(21) Determine the dome pressure at 80°F by referring to Fig. Series Number 3D. The dome pressure for each valve is found to be 560 psi.

(22) Determine the valve control seat size to take care of the 44 psi spread since casing overbuild will take care of the rest of the total of 94 psi.

(23) Refer to Fig. Series Number 3I and note that a 3/8" control port is needed. Select a 9/16" main seat size.

(24) Determine the test rack opening pressures of all valves from Fig. Series Number 3E. All are noted to be 655 psi at 80°F.

(25) Tabulate a summary of results as noted in Table 3.4100

Simplified procedure for previous example problem

As in the previous example, the test rack opening pressures are all 655 psi at 80°F. The design of this example can be simplified as follows:

Assume that all valves have been located and the temperature at each valve determined as set out in Steps 1 through 13 in the previous example.

(1) From 650 psi surface operating pressure, determine the operating pressure at the bottom valve to be 728 psi from Fig. 3A.1. (Appendix 3).

TABLE 3.4100
SUMMARY TABULATION—COMBINATION VALVES

Valve no.	Depth	Temp.	P_s Spring rating	P_t to Open	P_{so}	P_{vo} Pilot section opens	$P_{vo} - 50$ Bellows section opens	P_{vc}	P_{sc}	P_d 80°F	P_{tro} 80°F
1	1450	94	300	372	650	672	622	578	560	560	655
2	2050	100	300	382	650	682	632	588	650	560	655
3	2650	106	300	391	650	691	641	597	561	560	655
4	3250	112	300	400	650	700	650	606	563	560	655
5	3850	118	300	410	650	710	660	616	562	560	655
6	4450	123	300	419	650	719	669	625	563	560	655
7	5050	129	350	378	650	728	678	634	565	560	655

(2) From Fig. Series Number 3G, determine the gas volume necessary per cycle to be 3250 scf.
(3) From Fig. Series Number 3H, determine the annular reduction to be 94 psi.
(4) Subtract 94 psi from this value (728 – 94 = 634 psi). This will be the dome pressure and closing pressure of the bottom valve.
(5) Select the valve control seat size, using a low R value from 0.1 to .15. Use a 3/8" control seat size having an R value of .14. Use a large main seat size of 9/16".
(6) From Fig. Series Number 3D, determine the dome pressure at 80°F to be approximately 560 psi.
(7) Determine P_{tro} from $P_{tro} = \frac{P_d}{1-R} = \frac{560}{1.0-.14} = 651$ psi
(8) Set all valves at a test rack opening pressure of 650 psi. This simplified design works very well. A check on surface opening and surface closing pressures can be made at this time if desired.

3.43510 Design procedure for completely balanced valves

For valves that are completely balanced (no tubing pressure effect), always take a drop in surface casing pressure between valves. The following procedure and design example is given for this type of valve.[23] This flexible sleeve valve has been described in Chapter 2. It must be charged with gas and is set in the shop at 80°F.

3.435101 Graphical procedure for designing a low productivity intermittent installation for balanced valves

The following procedure serves as a guide in this design. Plot on rectangular coordinate paper (refer to Fig. 3.4146).
(1) Plot pressure on abscissa (horizontal) and depth on ordinate (vertical).
(2) Plot kick-off pressure less 50 psi at surface (zero depth).
(3) Obtain gas column weight from Fig. 3D.1 (Appendix D) and extend the pressure in Step 2 down the well.
(4) Plot surface operating pressure at surface (zero depth) and extend this pressure down the well by accounting for gas column weight from Fig. 3D.1.
(5) Plot the tubing pressure at the surface (for intermittent lift this is equivalent to the separator pressure).
(6) From either Fig. 3F.1 or 3F.2 (Appendix F), depending upon tubing size, select an unloading gradient for the design unloading rate. Always use a design rate slightly higher than the anticipated maximum production rate. This design rate is for spacing purposes only. Use a maximum of 0.04 psi/ft for all tubing sizes.
(7) Plot the unloading gradient from Step 6 down the well from the separator pressure or surface wellhead unloading pressure.

Spacing the valves:
(8) Determine at this point if the well is to be loaded with "kill" fluid all the way to the surface or if the well was pulled without adding "kill" fluid. The first valve may be placed at the static fluid level if the well has not been killed. For wells loaded to the surface, construct the "kill" fluid gradient from surface tubing pressure (for wells unloaded into system) down the well. For wells not loaded with "kill" fluid, make a point on the

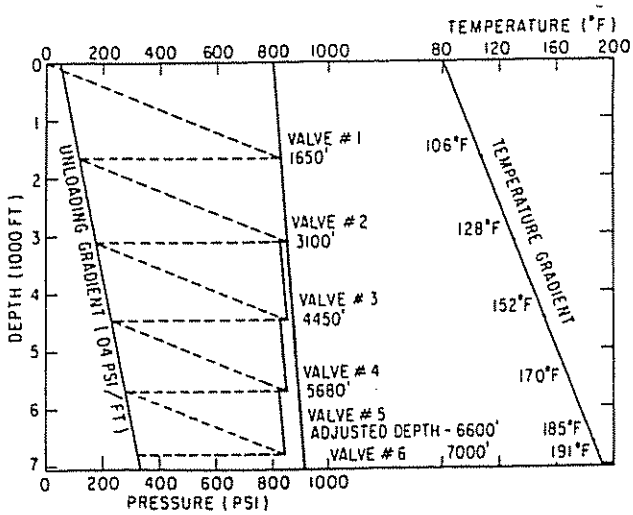


Fig. 3.4146 Intermittent Flow—Graphical Design for Low PI Well Using Balanced Valves

operating casing pressure line at the static fluid level for location of Valve 1.
(9) Extend the line down the well until it intersects (P_{ko} – 50): this locates Valve 1.
(10) From this point draw a horizontal line to the left until it intersects the unloading gradient line.
(11) From this point draw a line parallel to the "kill" fluid gradient line until it intersects the operating casing pressure line. If kick-off pressure is available for unloading all the way to bottom, this line may be brought to (P_{ko} – 75). For balanced valves take a 25 psi drop per valve down the hole. If kick-off pressure = operating pressure, use a 25 psi drop per valve from operating pressure. This locates Valve 2. In spacing use the pressure of the valve above less 25 psi to locate the next lower valve.
(12) Construct a series of lines from the surface that drop 25 psi from the kick-off and operating pressure. These serve as intersecting lines for spacing.
(13) Continue the spacing procedure between the unloading gradient line and the operating pressure lines until reaching the bottom of the well.
(14) Adjust and tabulate the depth of each valve.
(15) On the same graph paper, draw the geothermal gradient line from the BHT to the surface temp.
(16) Tabulate the temp at each valve.
(17) Tabulate the surface opening pressures of each valve.
(18) Make a complete summary tabulation of valve depth, temperature, surface opening pressure, settings at 60°F or 80°F, and pressures at the valves. Determine valve settings at 80°F and/or 60°F from Appendix D.
(19) Make any necessary adjustments in valve settings.

EXAMPLE PROBLEM—LOW PRODUCTIVITY WELL USING BALANCED VALVES (Fig. 3.4146)

Given data:
Tubing size = 2" nominal
Casing size = 5½" (17 lb/ft)
Packer depth = 7030 ft
Perforation depth = 7050-7060 ft
Separator pressure = 50 psig
Kick-off pressure = 850 psig
Operating pressure = 800 psig
Gas sp. gr. = 0.60
Oil gravity = 35° API
Formation temp = 192°F.
Flowing surface temp = 80°F.
Well loaded all the way to surface with a "kill" fluid having a gradient of 0.50 psi/ft
Static BHP — unknown
PI — unknown
Similar wells in field make form 20-100 B/D (all oil—35° API)
Can unload to zero pressure for first valve only

Procedure: (see Fig. 3.4146)
(1) Plot 7050 ft depth scale and 1000 psi pressure scale
(2) Plot (850 – 50) as (P_{ko} – 50) = 800 psi
(3) Obtain gas weight of 17 psi/1000 ft for a 0.60 sp gr gas and 800 psi from Fig. 3A.1 (Appendix A) and extend down the well.
(4) Plot surface operating pressure of 800 psi which is the same as (P_{ko} – 50) in this problem and extend down the well as in Step 3.
(5) Plot the tubing pressure of 50 psig which is also separator pressure for intermittent lift.
(6) From Fig. 3F.2 (Appendix 3F) unloading gradient is 0.04 psi/ft for 100 B/D design production rate and 2 in. tubing..
(7) Plot 0.04 psi/ft down the well from the tubing pressure of 50 psi.
(8) Construct the "kill" fluid gradient line of 0.50 psi/ft from zero pressure.
(9) Extend this 0.50 psi/ft line until it intersects the 800 psi line. This locates Valve 1 at 1650 ft.
(10) Draw a horizontal line left from 1650 ft until it intersects the 0.04 psi/ft line.
(11) Draw a line from Step (10) parallel to the 0.50 psi/ft line until it intersects the 800 psi line. Although these valves will be dropped 25 psi per valve, it is unnecessary to take this drop in spacing until Valve 2. In this problem there is 800 psi surface operating pressure to uncover valve 2 although it is set for a surface operating pressure of 775 psi. The reason for this is that the drop in pressure cannot occur until the valve is uncovered. Therefore, the pressure of the valve above is maintained until the next lower valve is reached. Although a drop in pressure will occur at this time, the pressure of the valve above has been utilized in reaching this point. Therefore, the pressure of the valve above is used to space the next valve down the well.
(12) Construct gas lines from the surface having operating pressures of 775, 750, 725, and 700 psi.
(13) Continue the spacing procedure down the well with Valve 3 intersecting the 775 psi line, Valve 4, the 750 psi line, etc.
(14) Valve depths are 1650, 3100, 4450, 5680, 6800, and 7000 ft. Since valves 5 and 6 are very close together, it would be better to adjust the spacing for the last two valves. Since valve 5 could conceivably operate part of the time, place it at 6600 ft, leaving only 400 ft to the bottom valve. The adjusted valve depths are 1650, 3100, 4450, 5620, 6600 and 7000 ft.
(15) Construct the temperature scale by drawing a straight line from 192°F at 7050 ft to 80°F at the top.
(16) Tabulate the temp at each valve to be 106, 128, 152, 170, 185 and 191°F.
(17) The surface opening pressures of each valve are 800, 775, 750, 725, 700, and 675 psi.
(18) Prepare a summary tabulation of valve settings at 80°F (Table 3.4101) selected from Fig. 3D.9 (Appendix).
(19) There is no need to make adjustments to this valve string. Since the valves are designed for a 25 psi drop in surface operating pressure there will be no valve interference. Notice that the valve settings at 80°F also drop in pressure. Al-

TABLE 3.4101
SUMMARY TABULATION—BALANCED VALVES

Valve no.	Depth (ft)	Temp (°F)	Surface opening pressure (psig)	Pressure at depth (psig)	P_{TRO} Setting at 80°F (psig)
1	1650	106	800	830	780
2	3100	128	775	835	745
3	4450	152	750	835	710
4	5680	170	725	830	670
5	6600	185	700	815	650
6	7000	191	675	795	630

though the valve opening pressures at depth do not show much change, there is no cause for alarm. The gas column weight takes care of this, leaving a 25 psi drop in surface opening pressures. There is also no need to worry about the tubing pressure load, that is, well feed in characteristics, because the tubing pressure has no effect in opening this valve. Therefore, the valve can be opened under any tubing load.

3.43511 Design example in which the static fluid level is low in the well and the well has not been loaded with "kill" fluid—balanced valves

This problem is the same as in the previous section except the static fluid level is at 4000 ft and the well is standing with its own oil with a gradient of 0.368 psi/ft. Given data:

Tubing size = 2"
Casing size = 5½" and 17 lb/ft
Packer depth = 7030 ft
Perforation depth = 7050 ft
Separator pressure = 50 psig
P_{ko} = 850 psig
P_{so} = 800 psig
γ_g = 0.60
Oil gravity = 35° API
Formation temp = 192°F.
Flowing surface temp = 80°F.
Design rate = 100 B/D

The analytical design procedure for this well would be as follows:

(1) D_{v1} = depth to static fluid level = 4000 ft

Although the valve could be placed slightly lower than 4000 ft, it is unwise to do so; when the tubing and packer are run, enough fluid may be displaced to cause the casing fluid level to rise.

(2) Reference to Fig. 3F.2 in Appendix F shows an unloading gradient of 0.04 psi/ft for 2" tubing and 100 B/D.

(3) $$D_{v2} = D_{v1} + \frac{(P_{so})_{v1} - G_u D_{v1} - P_{wh}}{G_s}$$

where:

G_u = unloading gradient = 0.04 psi/ft
$(P_{so})_{v1}$ = surface pressure of valve (1) = 800 psi
G_s = static gradient = 0.368 psi/ft

Since the well is not loaded

D_{v1} = depth to valve (1) = 4000 ft
P_{wh} = tubing back pressure = 50 psi
$D_{v2} = 4000 + [800 - (0.04)(4000) - 50]/0.368$
$= 4000 + 1600 = 5600$ ft

(4) $D_{v3} = D_{v2} + [775 - (0.04)(5600) - 50]/0.368$
$= 5600 + 1360 = 6960$ ft

Place this valve at 7000 feet which is one tubing joint above the packer.

As noted from Section 3.43510, this installation required a minimum of six valves when the well was loaded to the surface with a 0.50 psi/ft "kill" fluid. With the fluid level at 4000 ft and for a gradient of 0.368 psi/ft, it is possible to reach bottom with three valves.

This installation could also be designed graphically, starting with Valve 1 at 4000 feet.

The same procedure would be used for unbalanced valves except that Valves 2 and 3 could be spaced on a constant surface pressure of 700 psi.

The selection of valves at their shop set temp and the completion of the design would be conducted as set out in previous examples.

3.43512 Designing chamber gas lift installations for intermittent lift

3.435121 Introduction

(The several types of chamber installations are discussed in Section 3.43.) In low BHP, high PI wells, production may often increase and lift efficiencies may improve by the installation of a chamber type assembly. Additional fluid entry for a given drawdown can be accomplished and often results in increased production per day or fewer lift cycles per day for the same quantity of fluid. The lift gas savings are evident. Chambers are normally designed so the operating valve is the chamber valve. However, there are cases where the valve immediately above the chamber valve can be the operating valve. For the latter case, a small ported valve is used to unload the chamber and a larger ported valve above opens to supply sufficient gas to move the liquid out the tubing string.

3.435122 Procedure for designing standard chamber installations

Unloading valves for chamber installations are spaced as in previous instructions for conventional installations.

There are some precautions to be taken in chamber lift. The unloading valve next to the chamber may need to be deeper than normally assumed. If unbalanced bellows valves are being used, the bottom chamber valve may need to be lowered more than the normal 25 psi drop in pressure in order to ensure that all gas enters the bottom valve.

Another caution is to remember that the operating valve is set above the chamber and can receive no tubing pressure from the fluid load to help open it. The following procedure will serve as a guide in designing a chamber installation.

(1) Decide from well completion information the type

of chamber best suited for the particular well. For long perforated intervals and/or open hole completions, an insert chamber extending into the perforated interval would be best. For short perforated intervals and small casing sizes, the two-packer chamber is best suited. This assumes ample BHP to feed the fluids into the chamber.

(2) Decide upon the type of unloading valves.

(3) From available operating pressure and kick-off pressure, space the unloading valves to near bottom.

(4) Select the pressure of the operating valve. The pressure setting on the operating valve and the length of the chamber will be influenced by two limitations: the available gas injection pressure and the fluid feed-in rate into the well bore.

If information is available from a pressure buildup curve, it can give an idea of the size of chamber needed. Probably all that will be known will be the static BHP. Since chambers are best suited for high PI wells, it is likely that the well will feed in at a fast rate for 50-75% of the static pressure.

If there is a limit to available injection gas pressure, it automatically sets the upper limit on the size of chamber.

From the spacing of the valves in Step 3, the operating pressure available on bottom is known. Obtain this by adding gas column weight to surface operating pressure.

(5) Allow 150-300 psi differential between the total load in the tubing and the opening pressure of the operating gas lift valve. Total tubing load = back pressure plus liquid. (The higher the differential, the greater the percentage recovery).

(6) Convert the tubing liquid load in psi to barrels of liquid and, hence, feet of liquid in the tubing.

(7) Select a chamber size to hold this amount of liquid.

(8) Select a differential valve (spring-loaded) for the bleed valve or port. In cases of high GLR, a special, large opening bleed assembly should be utilized or a reverse flow chamber can be used. The differential on this valve must not be set too high. For low operating pressures set this differential as low as 25 psi. Small holes in the tubing or couplings have been used in some cases.

(9) Be sure the unloading valve immediately above the chamber valve is not spaced too far above the top of the chamber, for a stymied condition could easily occur. Back-check the surface opening pressures of all valves and determine the opening pressure of the operating valve.

(10) Prepare a summary tabulation.

3.4351221 EXAMPLE PROBLEM #1: FOR CHAMBER

Given data: (in Section 3.4351)

2⅜" O.D. tubing
Pressure available = 1000 psi
Desired operating press on casing = 900 psi
G_s = 0.40 psi/ft = kill fluid gradient
Depth = 7600 ft = position of bottom packer for two packer chamber
γ_g = 0.7
P_{wh} = 80 psi
$\overline{P}_R$ = 1500 psi
Well test shows 150 B/D for P_{wf} = 750 psi
Use time cycle control at the surface
7" casing (23 #/ft) (Use 2-packer Chamber)
Temp bottom = 167°F
Temp top = 109°F
PI = 0.2 (Avg.)

Various types of valves and/or spacing procedures could be used. Select pressure-operated unbalanced valves and time cycle control at the surface. Space the unloading valves as in Section 3.4353. A portion of Table 3.492 is reproduced here to show the summary tabulation for standard intermittent lift. For the spacing procedure, refer to Figure 3.4138 and Section 3.4353. Four valves are located as follows:

Valve no.	Depth	P_{so}
1	2500	1000
2	4600	975
3	6400	950
4	7600	900

Since this well is fairly good and should produce between 150-200 B/D, add one more valve for the chamber valve and permit the chamber valve to be the operating valve. Therefore, it should be a pilot valve with a control port of such size to supply one-half the required spread for choke control.

I would suggest the following settings and depths for a chamber installation. Assume the bottom packer is to be set at 7600 feet and the top packer at 7400 feet (also location of chamber valve). Keep Valve 5 very close to the chamber valve (within one or two joints), depending upon casing size and clearance.

Valve no.	Depth	P_{so}
1	2500	1000
2	4600	975
3	6400	950
4	7340	925
5	7400	850

Note that the pressure on Valve 5 is dropped 75 psi instead of the normal 25 psi. This prevents interference when the slug passes by Valve 4. Later, Valve 4 may be the operating valve and will, therefore, open as the slug passes. Also for low operating pressures available, proper design can prevent pressure loss on the bottom valve. The 75 psi drop also permits us to flag the chamber valve, that is, to know for certain when it is reached.

In chamber design remember that the chamber valve opens "dry," that is, it only has tubing pressure acting against it and has no liquid load. For example, in this case for 80 psi at the surface, there is approximately 90 psi in the tubing at the position of the chamber valve. Depending upon cycle frequency there may be more than 90 psi due to liquid spray still in the tubing string. If cycles are less than 30 minutes, stabilization time in the tubing will not be reached (time for all the spray to settle back down) (refer to Figures 3.4132 through 3.4136). For larger port sizes, this stabilization time occurs more quickly.

Chamber Design Procedure:

(1) Space the unloading valves and select the surface operating pressure of the chamber valve as discussed above.

P_{so} of chamber valve = 850 psi

(2) Find the opening pressure at 7400 ft of the chamber valve.

$P_{vo} = P_{so}$ + gas column weight
[Refer to Figure 3A.1 (Appendix)]
$P_{vo} = 850 + (22.2)(7.4) = \underline{1014}$ psi

(3) Select the tubing load to be

$$P_{t(load)} = P_{vo} - 250 = 1014 - 250 = 764 \text{ psi}$$

For lower pressures available, use between 150 to 200 psi differential between valve opening pressure and tubing load.

(4) Determine the tubing pressure opposite the chamber valve when the valve is closed.

$P_{t(valve\ closed)} = P_{surface}$ + wt. of gas and spray

Assume dry tubing

$P_{t(valve\ closed)} = 80 + 10 = 90$ psi

(5) Determine liquid load in psi

Liquid load = total tubing load − 90 = 764 − 90 = 674 psi

(6) Convert the liquid load in psi to feet of liquid in 2" tubing

$$\text{Feet of liquid} = \frac{674}{.40} = 1685 \text{ feet}$$

(7) Determine the ratio of chamber volume to tubing volume.

Ratio of casing volume to tubing volume = 8.8 (Refer to Table 2 Appendix 3J)
Ratio of chamber volume to tubing volume = 8.8 + 1 = 9.8

(8) Determine chamber length

$$= \frac{1685}{9.8} = 172 \text{ feet}$$

Place the chamber valve at 7600 − 172 = 7428 ft
Valve #4 is then located at 7428 − 60 = 7368 ft (2 joints above chamber valve). The chamber length can also be determined by the following formula:

$$L_{CH} = \frac{P_{vo} - \Delta P - (\text{Static } P_{tub})}{G_s(R_{ct} + 1)} \qquad (3.461)$$

where:

L_{CH} = chamber length in feet
ΔP = selected differential between casing pressure and tubing load to be lifted
Static P_{tub} = tubing pressure opposite the operating valve just as valve is ready to be opened
G_s = gradient in psi/ft of liquid to be lifted
R_{ct} = ratio of casing to tubing volume

For this example and using equation 3.461:

$$L_{CH} = \frac{1014 - 250 - 90}{(.40)(8.8 + 1)} = 172 \text{ ft}$$

(9) Determine chamber volume in bbl.

= (1685) (0.00387 B/FT) = 6.52 bbl

(10) Determine gas volume required per cycle (Refer to Figure 3G.8 in Appendix)

$P_{vo} = 1014$
Depth = 7600

Note that the depth of lift is from the bottom packer (perforated nipple slightly above) and not from the physical position of the valve itself. Also for a chamber it is necessary to fill the chamber with gas; in effect, the equivalent length of the chamber in feet must be added to the depth of the top packer. The equivalent length of tubing to fill both the tubing and the chamber annular space is:

Equivalent tubing length = (7600 − 172) + 1685 = 7428 + 1685 = 9113 ft

In using Fig. 3G.8 in the Appendix, we should use a depth of 9113 ft

Gas Vol/cycle = 7800 scf

(11) Determine the change in casing pressure to supply 7800 scf (refer to Fig. 3H.6 in the Appendix). This is 80 psi. Do not use 9113 ft for this value but use only that casing available for storage, which is 7428 ft. Hence, to use the chart, take 7800/7428 = 1050 scf/1000 ft and read vertically upwards to 2" tubing and find 80 psi.
So ΔP_{casing} = 80 psi (change at bottom)

(12) Determine the change in casing pressure for time cycle control. This is ½(80) = 40 psi.

(13) Determine valve closing pressure = P_{vo} − 40 = 1014 − 40 = 964 psi.

(14) Determine control port size

$$R = \frac{P_{vo} - P_{vc}}{P_{vo} - P_t} = \frac{1014 - 964}{1014 - 90} = .0433$$

Select the control port size to be 3/16 or ¼. (Use 3/16)
The R valve for a 1½" O.D. × 3/16 valve is .0359 for A_b = 0.77 in.²

1 − R = .9641

(15) Select the main seat size to be as large as feasible. Use ⅝" diameter pilot valve.

(16) Select the port sizes for all other valves by first determining that port size to just open Valve 4 as the bottom of the slug passes (refer to Table 3.489).

For ΔP = 250 and ⅝" port

Use:

$$P_{t(max)} = P_{vo} - (.09)(P_{vo})$$
$$= 1014 - (.09)(1014) = 923$$

Use:

$\Delta P_{minimum}$ = 100 psi

Therefore:

$$P_{t(max)} = 914 \text{ psi}$$

This is the pressure in the tubing at Valve 4 when the slug passes.

$$R = \frac{P_{vo} - P_{vc}}{P_{vo} - P_t}$$

Use:

$$P_{vc} = P_{vo} - 40$$

$$P_{vo} \text{ Valve 4} = 925 + (24.2)(7.368)$$
$$= 925 + 178 = 1103$$
$$R = \frac{1103 - 1063}{1103 - 914} = \frac{40}{189} = .2116$$

The nearest port size is a 7/16" ported valve with an R value of .1952 and 1 – R = .8048 for a bellows area of 0.77 in. This port size assures that all the slug is above the valve before it can be opened with 1105 psi in the casing.

The chamber valve opens with 1014 in the casing and requires an overbuild of 40 psi = 1054 psi, which is insufficient to open Valve #4 even under maximum tubing load. This assures operation from the chamber valve only.

This means that the opening pressure of the chamber valve could be raised.

The other three unloading valves are set with 5/16" ports for unloading only.

(17) Prepare a summary tabulation showing the setting of all valves (see Table 3.4102).

(18) Check to see if the passing slug will open unloading Valve 3. Since it was designed to open for a surface casing pressure of 950 and a tubing load of 490, it is conceivable that the added tubing pressure as the slug passes could open it. Solve for the tubing pressure to open it when the operating valve is open.

$P_{vo\#5} = 1014$ and $P_{vo\#3}$ for this press. = 933 psi

$$P_{t(req.)} = \frac{P_d}{R} - \frac{P_{vo}(1-R)}{R} = \frac{1048}{.0996} - 993\left(\frac{.9004}{.0996}\right)$$
$$= 10522 - 8977 = 1545 \text{ psi} = \text{required tubing press. to open Valve 3}$$

With less than 900 psi possible as the slug passes, the unloading valves cannot be opened.

Summary discussion

This design is a safe chamber installation for single point injection through the chamber gas lift valve. The valve immediatley above #4 can possibly be opened as the bottom of the slug passes by and will cause no harm if it does. If Valve 4 is the operating valve, run the chamber valve with a small port only and set Valve 4 to open on a slightly lower tubing load, assuring that it would open.

Additional design procedures are illustrated in the following sections.

3.4351222 EXAMPLE PROBLEM #2

Given data:
- Tubing size = 2"
- Casing size = 5½" and 17 lb/ft
- Packer depth = 7030 ft (Use a 2-packer chamber)
- Perforation depth = 7050-7060 ft
- Separator pressure = 50 psi
- Kick-off pressure = 850 psi
- Operating pressure = 800 psi
- Gas sp gr = 0.60
- Oil sp gr = 35° API
- Formation temp = 192°F.
- Flowing surface temp = 80°F.
- Well loaded to surface with "kill" fluid having a gradient of 0.50 psi/ft; can unload to pit for first valve only.
- Estimated production rate = 100 B/D

Procedure for Balanced Valves:
(For Two-Packer Chamber)

(1) From completion information run a two-packer chamber since there is only a 10-ft perforated interval and a packer is already set at 7030 ft. In particular, if this is a permanent-type packer it may be utilized for the bottom packer of the chamber.

(2) Assume balanced pressure operated valves for unloading and for the operating valve.

(3) Space the valves to near bottom. Use an analytical procedure and space the valves to bottom on

TABLE 3.4102
SUMMARY TABULATION—CHAMBER EXAMPLE #1

Valve no.	Depth	Temp	P_{so}	P_{vo}	P_t	P_t Max	Control port	Main port	$P_d = P_{vc}$	P_{sc}	P_d 60°F	P_{tro}	Final calib
1	2500	128	1000	1066	240	-	5/16	5/16	984	923	855	950	950
2	4600	144	975	1092	375	-	5/16	5/16	1021	910	852	946	945
3	6400	158	950	1110	490	-	5/16	5/16	1048	898	860	955	955
4	7368	165	925	1103	-	914	7/16	5/8	1066	895	865	1075	1075
5	7428	166	850	1014	-	90	3/16	5/8	980	820	800	830	830

3/16" R = 0.0359
1 – R = 0.9641

5/16" R = 0.0996
1 – R = 0.9004

7/16" R = 0.1952
1 – R = 0.8048

$P_d = P_v = P_{vo}(1 - R) + P_t R$

a design rate of 100 B/D. The unloading gradient of 0.04 psi/ft is found from Fig. 3F.2 in Appendix.

$$D_{v_1} = \frac{p_{ko} - 50}{0.50} = \frac{850 - 50}{0.50} = 1600 \text{ ft}$$

(use 800 psi valve)

$$D_{v_2} = 1600 + \left[\frac{800 - (0.04)(1600) - 50}{0.50}\right] =$$

2972 ft (use 775 psi valve)

$$D_{v_3} = 2972 + \left[\frac{775 - (0.04)(2972) - 50}{0.50}\right] =$$

4182 ft (use 750 psi valve)

By the same procedure:

D_{v_4} = 5248 ft (use 725 psi valve)
D_{v_5} = 6178 ft (use 700 psi valve)
D_{v_6} = 6984 ft (use 675 psi valve)

The bottom valve at 6984 ft will not be needed since the bottom packer is at 7030 ft. However, it is likely Valve 6 near the top of the chamber will be needed in addition to a 7th valve which will be the chamber operating valve.

(4) Select the pressure of the operating valve on the basis of available pressure at that point. Six unloading valves will be needed and the surface operating pressure of the last unloading valve (6) is 675 psi, so the surface operating pressure of the chamber valve must be less than 675 psi. Since the chamber valve does not pass gas immediately into the tubing, but gas first enters on top of the fluid in the chamber, this gas is likely to cause a pressure buildup on the casing greater than a straight tubing installation. Therefore, I recommend a drop in pressure of 50-75 psi from the last unloading valve to the chamber valve. Set the chamber valve at 600 psi surface operating pressure.

From Figure 3A.1 in Appendix 3A, gas column weight is 14 psi/1000 ft. The top of the chamber is estimated to be at 6800 ft and the pressure at that point is 600 + [(14)(6800)]/1000= 695 psi.

(5) Allow a 150 psi differential between the 695 psi casing pressure and the tubing load. With a surface separator pressure of 50 psi, the estimated pressure in the tubing at 6800 ft is 60 psi. Total tubing load = (695 − 150) = 545 psi. Liquid load = (545 − 60) = 485 psi.

(6) A 485 psi liquid load = 485/0.368 = 1320 ft. A 35° API oil has a gradient of 0.368 psi/ft. (1320)(0.00387 B/ft) = 5.1 barrels of liquid to be lifted.

(7) Table 2 in Appendix 3J shows a casing annulus to tubing volume ratio of 4.61/1. The chamber itself includes the tubing volume. The total chamber (including tubing volume) has a ratio of 5.61/1 to the tubing only. Therefore, the 1320 ft. of fluid load in the tubing will be 1320/5.61 = 235 ft of chamber to hold 5.1 barrels of fluid.

(8) Use a 50 psi spring or less for the differential of the bleed valve.

(9) Check the unloading valve immediately above the chamber. With the fluid in the annulus unloading through the bottom valve, it is possible to support the (695 − 60)/0.50 = 1270 ft of "kill" fluid loading the well. This 1270 ft is balanced from bottom and will be pushed (1270 − 235) = 1035 ft above the chamber valve. Therefore, a valve is needed within 1000 ft of the top of the chamber. The top of the chamber is located at (7030 − 235) = 6795 ft. With Valve 5 at 6178 ft there should be no need for Valve 6. However, Valve 6 should be placed two or three joints above the chamber (6700 ft) as an unloading valve, positioned to serve as an operating valve if necessary.

(10) A summary tabulation follows in Table 3.4103. Although some operating pressure has been sacrificed, this would be a good chamber design. A better design that would conserve pressure would use unbalanced valves for unloading. A large-ported, unbalanced valve with special pilot and control port to control the spread would be a good selection for the chamber valve.

TABLE 3.4103
SUMMARY TABULATION—TWO PACKER CHAMBER
(BALANCED VALVES)

Valve No.	Depth (ft)	Temp (°F)	Surface operating pressure (psig)	Pressure at depth (psig)	Setting at 80°F (psig)
1	1600	106	800	830	780
2	2972	127	775	830	745
3	4182	146	750	830	715
4	5248	164	725	820	680
5	6178	179	700	800	650
6	6700	186	675	772	620
7	6795	188	600	695	555

The design procedure for other valves would be similar.

This same chamber could be completed with unloading valves all set for the same surface closing pressure. A summary tabulation for unbalanced valves is presented in Table 3.4104.

TABLE 3.4104
SUMMARY TABULATION—TWO PACKER CHAMBER
(UNBALANCED VALVES)

Valve no.	Depth (ft)	Temp (°F)	P_{sc} (psig)	P_{vc} (psig)	P_{vo} (psig)	P_{so} (psig)	P_d (psig)	P_{tro} at 60°F (psig)
1	1650	106	700	727	828	795	650	760
2	2900	126	700	748	847	790	640	750
3	4100	146	700	768	860	780	635	740
4	5225	164	700	786	880	780	625	730
5	6280	180	700	804	880	760	620	724
6	6745	188	605	695	880	700	530	620

Procedure for Unbalanced Valves

For unbalanced valves, know the casing pressure reduction required to lift the slug. A 105 psi drop in casing pressure is needed to supply the injection gas volume of 5300 scf/cycle (volume needed to lift the slug).

Since the operating valve will have only separator pressure plus gas weight acting on it before it opens, it will be at a position of maximum spread. Therefore, it should be a pilot controlled valve with a large main port size.

Assume that 100% of the gas is supplied by choke control at the surface. The bottom operating valve must have a 105 psi spread built into it with a back pressure of 60 psi tubing pressure. Again, the opening pressure on this bottom valve should be 50-75 psi less than the valve directly above it. Set this valve at 700 psi surface operating pressure with a gas weight of 102 psi at 6750 feet (approximate top of chamber).

The opening pressure at depth is (700 + 102) = 802 psi (use 800 psi opening pressures at depth).

$$p_d = p_{vo}(1 - R) + p_t R$$

and:

$$R = \frac{p_{vo} - p_d}{p_{vo} - p_t}$$

$$p_{vc} = p_d = 800 - 105 = 695 \text{ psi}$$

and:

$$R = \frac{800 - 695}{800 - 60} = \frac{105}{740} = 0.142$$

Figure 3I.1 in Appendix 3I shows that a ⅜" control port is needed. Use a 9/16" or larger main port with this control port.

To find chamber length:

$$L_{CH} = \frac{P_{vo} - \Delta P - P_{t(static)}}{G_s\,(R_{ct+1})} = \frac{800 - 150 - 60}{(0.368)\,(5.61)} = 285 \text{ ft}$$

Comparing this installation to the previous one with balanced valves results in arriving at the operating valve with more pressure. However, the opening and closing pressures of the balanced valve are essentially the same as the closing pressure of the unbalanced valve. For intermitter control at the surface, the two installations would perform essentially the same.

3.435123 EXAMPLE PROBLEM FOR INSERT CHAMBER

Given data:
Packer = 5030 ft
Long perforated interval = 560 ft
2⅜" O.D. tubing
5½" 17#/ft casing
Pressure available = 600 psi
Rate less than 100 B/d
G_s = 0.40 psi/ft
Temp bottom = 104°F.
Temp top = 80°F.
$P_{separator}$ = 50 psi
Use time cycle control at the surface.
Space valves analytically taking 25 psi drop between valves.
Design an insert chamber for this well using 3½" O.D. tubing and a 1" dip tube and the dip tube is to extend 30' up into the tubing.

Procedure:

(1) Space the gas lift valves. The gradient for unloading is .04 psi/ft as found in Appendix 3F.2. Space valves analytically, assuming a standard intermittent installation.

$$D_{v_1} = \frac{600 - 50}{.40} = 1375 \text{ ft}$$

$$D_{v_2} = \frac{600 - 50 - .04(1375)}{.40} + 1375 = 2612 \text{ ft}$$

The other three valves are spaced accordingly, and the following table 3.4105 shows the results.

(2) For this chamber design place one more valve (no. 6) at 5030 ft and set it for P_{so} = 450 psi to prevent interference. It will be the operating valve and will open against only 60 psi tubing press. For choke control the spread will be 60 psi for a gas volume of 2200 scf/cycle. For time cycle control use spread = ½(60) = 30 psi.

So $P_{vc} = 520 - 30 = 490$ psi for time cycle control.

$$R = \frac{P_{vo} - P_{vc}}{P_{vo} - P_t} = \frac{520 - 490}{520 - 60} = \frac{30}{460} = .0652$$

Use a ¼" control port and a 9/16" or ⅝" main seat size.

R = .0638 for a bellows area of 0.77 in.²
1 − R = .9362

(3) Use ΔP = 150 psi between the tubing load and casing pressure at valve.
Total tubing load = 520 − 150 = 370 psi
Liquid load = 370 − 60 = 310 psi
Feet of liquid in tubing = $\frac{310}{.40}$ = 775'

(4) Solve for chamber size. The 775 ft in the tubing is a combination of 1" tubing inside the chamber and 2" tubing above.
The following volumes are needed:

TABLE 3.4105
SUMMARY TABULATION—INSERT CHAMBER

Valve no.	Port	Depth	P_{so}	Temp	P_{vo}	P_t	P_{vc}	P_{sc}	P_d (60°F)	P_{tro}	Adj. P_{tro}
1	⅜	1375	600	965	620	105	545	530	510	545	595
2	⅜	2612	575	111.3	610	154	545	510	495	580	580
3	⅜	3663	550	124	598	197	540	495	475	555	555
4	⅜	4547	525	134.6	582	232	532	480	460	537	535
5	⅜	5000 (Adj)	500	140	560	250	515	460	440	514	515
6	¼"×⅝"	5030	450	140	520	60	490		420	449	450

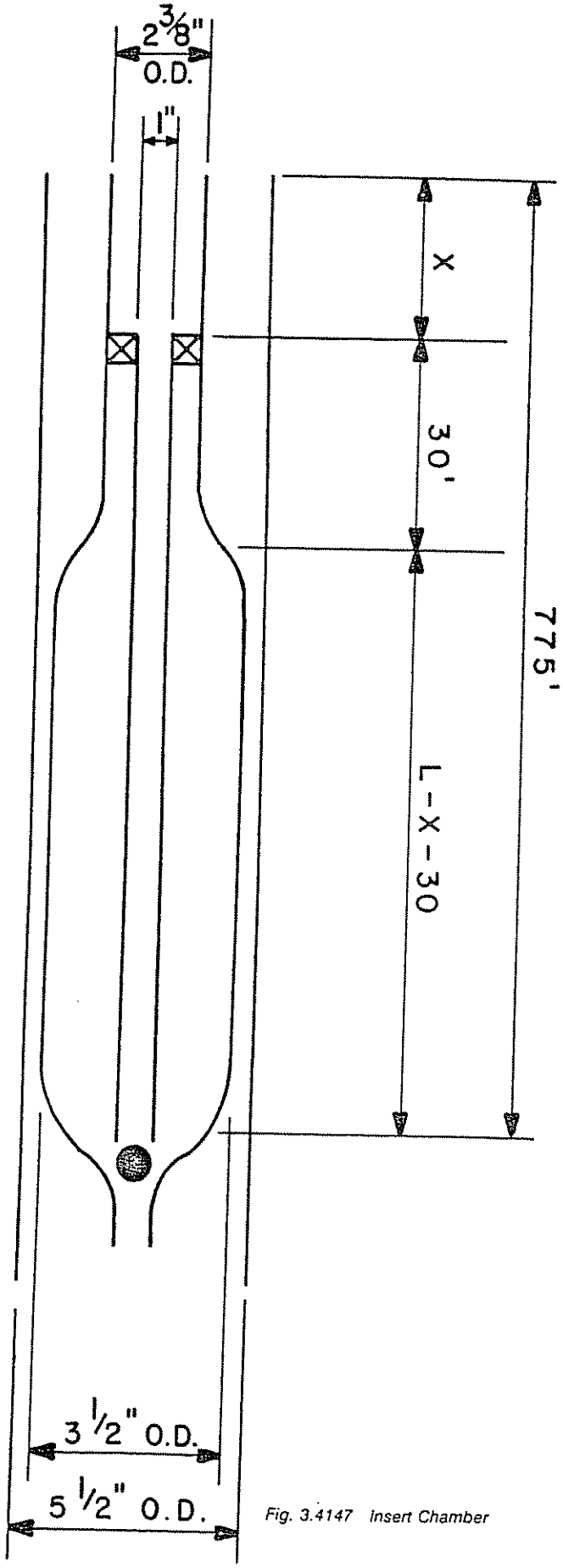


Fig. 3.4147 Insert Chamber

Vol of 3½" O.D. × 3" I.D. pipe = 8.70 B/1000' (9.3#/ft)
Vol of 2" = 3.87 B/1000 ft
Vol of 1" = 1.07 B/1000 ft

Just as the fluid all enters the 1" pipe from the chamber, we want 775 ft above that point (bottom of chamber). We also have 30 ft of 1" extending into the 2" pipe.

Refer to Figure 3.4147 to solve for the length of the chamber.

$$(L - X - 30)(.0087 \text{ BBLS/ft}) + 30(.00387 \text{ B/ft}) = X(.00387) + (L - X - 30 + 30)(.00107 \text{ B/ft}) \quad (3.462)$$

$L = 775$ ft total

$(775 - 30 - X)(.0087) + (30)(.00387) = X(.00387) + (775 - X)(.00107)$

$6.4815 - .0087X + .1161 = .00387X + .82925 - .00107X$

$5.76835 = .0115X$

$X = 501$ ft

Chamber length = 775 − 501 = 274', which includes 30 ft of 1" inside 2", or 274 − 30 = 244 ft of 1" inside of 3½".

To check this solution: For 274 ft of chamber, there is (244)(.0087) = 2.1228 BBLS for 1" inside of 3½". There is 30 ft of chamber with 1" inside of 2" = .00387 B/ft = (30)(.00387) = .1161

Total chamber = 2.1228 + .1161 = 2.238 B

These 2.238 B are to enter 775 total tubing made up of 1" and 2" with 501 ft of 2" and 274 ft of 1" for a total of 775 ft.

(501)(.00387) + 274(.00107) = 1.93887 + .29318 = 2.232 B (check close enough).

Therefore, the chamber will consist of 244' of 3½" O.D. tubing and 30 ft of 2" tubing above and with 1" inside both.

3.435124 Special chamber design for deep wells and low surface gas operating pressure

3.4351241 Introduction

Many cases exist where low surface pressure is available for the operation of a gas lift system, yet the well is fairly deep (8000 − 10,000 ft). Perhaps only 650-700 psi is available to operate the well. The well will produce more than 200 B/D if it is possible to reach bottom and operate from a chamber. In this type of installation, conserve all the pressure possible.

Two types of installations have been designed for wells of this type. The chamber valve can be the operating valve or the valve immediately above the chamber valve can be the operating valve, and possibly 2 or 3 other valves can be opened above for multi-point injection.

Work the following example well, utilizing both types of installations and incorporate a chamber at the bottom.

Given data:

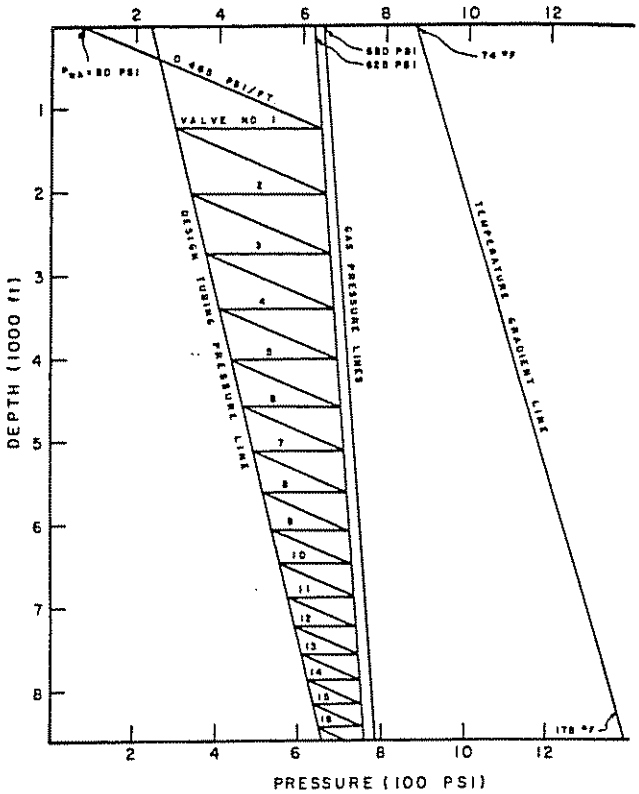


Fig. 3.4148 Special Chamber Design for Low Operating Pressure and Deep Wells

Center perforations = 8650 ft
Bottom packer = 8600 ft
Operating gas pressure = 650 psi
Maximum gas pressure = 650 psi
Minimum gas pressure = 650 psi
Gas gravity = 0.65
Casing size = 7", 23#/ft
Tubing size = 2⅜" O.D.
Est. oil production = 200 B/D
Static BHP = low
PI = unknown
$P_{separator}$ = 80 psi
Temp = 74° Surface
Temp gradient = 1.21°F/100 ft
Well loaded to surface with salt water of gradient = 0.465 psi/ft
Use a two-packer installation
Well produces 100% 35°API oil
Refer to Figure 3.4148

3.4351242 Design procedure—intermitter control (chamber valve to be operating valve)

(1) Prepare scales as noted in Figure 3.4148 for depth and pressure.
(2) Note 650 psi casing pressure at top of well and extend casing pressure downwards including gas column weight (15.3 psi/1000 ft from Figure 3A.1, Appendix).
(3) Allow 25 psi safety factor to open valves and mark 625 psi at top and extend casing pressure downwards to bottom of well (15 psi/1000 ft from Figure 3A.1, Appendix).
(4) Note a design tubing pressure at the surface. This can vary depending upon operating pressure. A good value is separator pressure plus 20 to 25% of casing pressure. For our case this is 80 + .25(625) = 236 psi (use 240). Fewer valves will be used by taking 80 psi + 20% of 625.
(5) Select the differential between tubing load at bottom to be lifted and casing operating pressure at bottom to be between 150 and 250 psi. (Use 200 psi) P_{vo} at 8500 ft = 753 psi. P_{tubing} = 753 − 200 = 553 psi (use 550). Mark this pressure at location of chamber valve estimated to be 100 ft above bottom packer = 8600 − 100 = 8500 ft.
(6) Draw in the design tubing pressure line between 240 psi at top and 550 psi at 8500 ft.
(7) Mark 80 psi surface wellhead pressure.
(8) Start spacing by constructing the kill fluid gradient of 0.465 psi/ft downwards from 80 psi at the surface until intersecting the 625 psi operating pressure line. This locates Valve 1 at 1225 ft. Space all other valves between the 625 psi casing pressure line and the design tubing pressure line.
(9) Draw in the temperature gradient line starting with 74°F at the surface and using a gradient of 1.21°F per 100 ft which gives 178°F at the bottom (8600 ft).
(10) Prepare a tabulation showing valve depths, temperature, valve opening pressures at depth and tubing design pressures opposite each valve.
(11) Select the opening pressure for the chamber valve to be available pressure − 50 = 752 − 50 = 702 psi (use P_{vo} = 700 psi).
(12) Determine the tubing pressure opposite the chamber valve at the instant it is ready to open. Assume 80 + 20 = 100 psi.
(13) Find the gas volume necessary per cycle. See Figure 8 (Appendix 3G). Gas volume = 6000 scf/cycle. Assume chamber volume = 1200 ft in tubing; therefore, total gas volume must fill 8500 + 1200 = 9700 ft of tubing.
(14) Determine change in casing pressure necessary to supply the gas volume of 6000 scf/cycle. See Figure 6 (Appendix 3H). ΔP = 52 psi = valve spread (use 50 psi).
(15) Set ½ the spread in the valve for surface intermitter control. Therefore valve closing pressure = 700 − ½(50) = 675 psi.
(16) Solve for valve control port size

$$R = \frac{P_{vo} - P_{vc}}{P_{vo} - P_t} = \frac{700 - 675}{700 - 100} = \frac{25}{600} = .04167$$

(17) Check for nearest standard port size available. Nearest port size = 3/16 (see Table 3.488). However due to possible plugging use a ¼" port. Use a 1½" valve with a bellows area of 0.77 in². For a ¼" control port and 1½" diameter valve we have:

R = .0638 and 1 − R = 0.9362
TEF = .06815 for A_b = 0.77 in²

For P_d = constant at 675 then $P_{vo} = \frac{P_d}{1-R}$ –

$$TEF(P_T) = \frac{675}{.9362} - 100(.0615)$$

$P_{vo} = 721 - 6 = 715$ psi and we have a spread of $715 - 675 = 40$ psi.

This is still satisfactory for intermitter control at the surface. Use a ⅝" to ¾" main seat size on the chamber valve with the ¼" control port.

(18) Select the chamber size. With $P_{vc} = 675$, build the casing to 725 (50 psi spread) to furnish the necessary gas.
  (a) Use a tubing load of 200 psi less than maximum pressure of 725 psi on casing = 525 psi.
  (b) Liquid load = 525 – 100 = 425 psi in tubing.
  (c) Feet of load = 425/.368 = 1155 ft in tubing.
  (d) Convert to feet in chamber, chamber vol/tubing vol = 9.80 (refer to Table 2, Appendix 3J) length of chamber $= \frac{1155}{9.8} = 118$ ft.

(19) Select port size for unloading valves based on P_T design. For valve immediately above chamber, $P_{vo} = 752$, $P_T = 548$. Try $P_{vc} = 752 - 50 = 702$ psi. $R = \frac{P_{vo} - P_{vc}}{P_{vo} - P_T} = .2745$. Nearest port size that does not exceed this spread = 7/16". $R = 0.1952$; $1 - R = .8048$ for $A_b = .77$ in². Therefore $P_{vc} = P_d = P_{vo}(1 - R) + P_T R = 752(.8048) + 548(.1952)$; $P_{vc} = 712$; spread $= 752 - 712 = 40$ psi.

(20) Run all other unloading valves with 7/16" ports.

(21 Adjust locations of bottom valves.
  (a) Chamber valve and top packer = 8600 – 118 = 8482 ft
  (b) Place next Valve 1 joint above chamber valve = 8450 ft

(22) Solve for closing pressures on all unloading valves which is also the dome pressure at depth for each valve.

(23) Determine dome pressure at 60°F.

(24) Determine P_{TRO}

(25) Complete the summary tabulation as noted in Table 3.4106.

3.4351243 Design procedure—chamber choke control—chamber valve to be operating valve

Same problem as the example in Section 3.435125. Steps 1 through 14 are the same as previous example section 3.4351252.

(15) Set the necessary spread of 50 psi (rounded off) in the chamber operating valve. Therefore valve closing pressure = 700 – 50 = 650 psi.

(16) Solve for valve control port size

$$R = \frac{P_{vo} - P_{vc}}{P_{vo} - P_T} = \frac{700 - 650}{700 - 100} = .08333$$

(17) Check for nearest port size. Nearest port size = 5/16". Although this valve has an R value of .0996 and will give a spread at bottom of 61 psi for an opening pressure of 711 psi, its average spread in the casing will be only about 55 psi, giving slightly too much gas per cycle.

(18) Design the chamber as set out in the previous example using 711 psi as the opening pressure.

(19) Unloading valves are the same as in the previous example.

Steps 20, 21, 22, 23, 24, and 25 are the same as set out in the previous example.

3.4351244 Special chamber design

Design Procedure—Choke control—valve above chamber valve to be the operating valve.

Introduction

In this example, the same problem example of Section 3.4351252 is to be worked but the valve immediately above the chamber valve is to be the operating valve. The port size on the chamber valve is purposely left with a smaller port size than the operating valve. The

TABLE 3.4106
SUMMARY TABULATION—LOW PRESSURE DEEP CHAMBER

Valve No.	Depth	Temp	P_{so}	P_{vo}	P_T Design	P_{vc}	P_d (60°F)	P_{TRO}	Control Port	Main Port	Adjusted P_{TRO}
1	1225	89	625	643	285	573	540	671	7/16	7/16	670
2	2000	98	625	655	313	588	543	675	7/16	7/16	675
3	2750	107	625	666	340	602	547	680	7/16	7/16	680
4	3400	115	625	676	364	615	550	683	7/16	7/16	685
5	4000	122	625	685	386	627	553	687	7/16	7/16	685
6	4550	129	625	693	406	637	555	690	7/16	7/16	690
7	5100	136	625	702	426	648	557	692	7/16	7/16	690
8	5600	142	625	709	444	657	559	695	7/16	7/16	695
9	6075	147	625	716	462	666	561	697	7/16	7/16	695
10	6450	152	625	722	475	674	563	700	7/16	7/16	700
11	6875	157	625	728	491	682	565	702	7/16	7/16	700
12	7200	161	625	733	503	688	565	702	7/16	7/16	700
13	7550	165	625	738	515	694	566	703	7/16	7/16	705
14	7850	169	625	743	526	701	568	706	7/16	7/16	705
15	8150	173	625	747	537	706	568	706	7/16	7/16	705
16	8450	176	625	752	548	712	570	708	7/16	7/16	710
17	8480	177	595	715	100	675	540	578	¼	⅝	580

accumulated liquids are U-tubed out of the chamber at a slower rate due to slower gas entry.

After the accumulated liquids have passed by the operating valve, it opens and, having a larger port size, gives the added gas necessary to move the slug on to the surface. Also an additional two or three valves may open, giving partial multi-point injection to drive the slug on to the surface at an even greater velocity.

This design does not require any additional drop in pressure on the chamber valve; therefore, all valves can be set for the same surface opening pressure of 625 psi. In the final analysis it is possible to adjust the operating valve immediately above the chamber, and it may incorporate a slightly different opening pressure.

EXAMPLE PROBLEM:

Same as example problem of Section 3.4351252. Steps 1 through 12 will be the same as in section 3.4351252.

(13) Find the gas volume per cycle to be 6000 scf and is slightly greater due to P_{vo} = 753 psi (see Figure 8, Appendix 3G).

(14) Change in casing pressure necessary to furnish 6000 scf = 52 psi (see Figure 6, Appendix 3H).

(15) Set the correct spread in the chamber valve (52 psi). Solve for port size

$$R = \frac{P_{vo} - P_{vc}}{P_{vo} - P_T} = \frac{753 - 701}{753 - 100} = \frac{52}{653} = .0796$$

(16) Select nearest standard port size. Nearest port size = 9/32"

where:

R = .08068
1 − R = .91932
TEF = .08776
A_b = 0.77 in² for 1½" valve

Solve for the opening pressure for the 9/32" valve to be:

$$P_{vo} = \frac{P_d}{1 - R} - TEF(P_T) = \frac{701}{.9193} - .08776(100) = 754 \text{ psi (use } P_{vo} \text{ available} = 753 \text{ psi) spread} = 52 \text{ psi.}$$

For this design use a standard pressure operated valve with a 9/32" port (no pilot) as the chamber gas lift valve. In this case this valve is used only to unload the chamber and the valve immediately above is to be the operating valve. Since the liquid from the chamber will not be moving very fast as it leaves the chamber (due to a small 9/32" port) the slug will not offer too much frictional resistance at that time.

(17) Determine the chamber size
(a) Tubing load = 750 − 200 = 550 psi
(b) Liquid load = 550 − 100 = 450 psi in tubing.
(c) Feet of liquid = 450/.368 = 1223 ft
(d) Length of chamber = $\frac{1223}{9.8}$ = 125 ft

(18) Select the port size for the operating valve (valve above chamber) based on a total tubing load of 750 − 200 = 550 psi, and a casing pressure slightly less than 750 psi due to a drop in casing pressure as gas has entered the chamber. To prevent opening the operating valve (valve above chamber) into a portion of the slug, set the operating valve to open on a tubing load of 50 psi greater than the liquid load plus back pressure. Therefore, set it to open on a tubing load of 600 psi. Determine the drop in casing pressure that has occurred in filling the chamber with gas. The total equivalent tubing length that must be filled with gas prior to reaching the operating valve is 1223 ft (125 ft of chamber) and represents a drop in casing pressure of 8 psi. Therefore set the operating valve to open on 750 − 8 = 742 psi casing pressure and 600 psi tubing pressure. Set the valve to close at 700 psi casing pressure (same as the chamber valve). Hence, its spread will be 742 − 700 = 42 psi. Solve for its control port size:

$$R = \frac{P_{vo} - P_{vc}}{P_{vo} - P_T} = \frac{742 - 700}{742 - 600} = \frac{42}{142} = .2958$$

The nearest control port size is ½" so use a ½" standard pressure operated valve. For a ½" valve the R value is .2562, 1 − R = .7438 and TEF = .3444, for A_b = 0.77 in². The closing pressure on this valve is:

$$P_{vc} = P_{vo}(1 - R) + P_T R = 742(.7438) + 600(.2562) = 552 + 154 = 706 \text{ psi}$$

This should give satisfactory operations.

(19) Additional valves up the hole (perhaps the next 2 or 3) can also be set to open as the slug passes, assuring the gas of entering more quickly into the tubing string.

3.4351245 Field example of deep chamber for low operating pressure

The following examples represent field cases in the U.S. of two chambers designed to operate from below 7000 ft with little or no loss in surface operating pressure.

FIELD CASE #1 (USING VALVE ABOVE CHAMBER AS OPERATING VALVE)

Given:
2⅜" tubing by 7" Casing
Maximum operating pressure = 650 psi
Minimum operating pressure = 650 psi
γ_g = 0.65
Rate = 250 B/D all oil (35° API)
P_{wh} = 100 psi
P_R = unknown
PI = unknown
Temp. grad. = 1.21 °F/100 ft
Surface temp. = 74°F.
Depth to top packer = 8470 feet

Refer to Table 3.4107 for the spacing and summary tabulation. Note that a surface opening pressure of 650 psi is maintained constant for all valves until reaching the operating valve, at which time it is dropped 11 psi (volume to fill chamber) to 639 psi. This is a field design operating successfully whereby the chamber valve with a 9/32" port is used to U-tube the fluids from the chamber into the tubing string after which the valve

TABLE 3.4107
FIELD EXAMPLE DEEP CHAMBER LIFT
(Valve Above Chamber = Operating Valve)

Valve no.	Depth	Port	Temp	P_{so}	P_{vo} Depth	P_T Design	$P_d = P_{vc}$ (Depth)	P_{sc}	P_d 60°F	P_{TRO} 60°F	Final calib.	Remarks
1	1200	7/16	94	650	670	270	589	571	546	687	685	7/16 Port, R = .201 1 – R = .799, TEF = .355
2	2050	7/16	104	650	680	305	604	574	552	692	690	7/16 Port, A_p = .1538 A_b = .7650
3	2850	7/16	114	650	695	330	621	579	556	696	695	
4	3600	7/16	123	650	705	360	635	582	559	700	700	
5	4300	7/16	132	650	715	390	649	586	562	703	705	
6	4950	7/16	140	650	730	410	665	592	567	709	710	
7	5550	7/16	146	650	740	430	677	595	571	715	715	
8	6150	7/16	154	650	750	455	690	600	574	718	715	
9	6700	7/16	161	650	760	480	703	604	577	723	720	
10	7200	7/16	166	650	765	490	709	603	577	722	720	
11	7700	7/16	172	650	770	510	717	604	577	723	720	
12	8175	7/16	178	650	780	530	729	608	581	723	725	
13	8440	½	183	639	761	577	713	591	563	763	760	Operating Valve
14	8470	9/32	183	650	775	112	719	594	568	620	620	Chamber Valve

For ½" Port
A_p = 0.2002, R = 0.262, TEF = 0.355, A_b = 0.7650
1 – R = 0.738

For 9/32" Port
A_p = 0.0643, TEF = 0.092, R = 0.084, A_b = 0.7650
1 – R = 0.916

immediately above the chamber opens and drives the slug of liquid on to the surface.

Notice that it opens on a casing pressure at the surface of 639 psi which is 11 psi lower than the normal surface opening pressure of 650 psi. This 11 psi is the necessary casing pressure drop to fill the chamber with gas. It opens on the design tubing load of 577 psi which is at a casing to tubing differential of (761 – 577) = 184 psi which represents a good differential and assumes no frictional resistance at this time. Change the valve setting to open at a higher tubing pressure (say 625 psi) to assure that it will not open into the slug.

This type of installation prevents having to drop the pressure on the chamber valve when it is used as the operating valve. Refer to Table 3.4107 for complete details on this installation.

3.4351246 Field case #2—chamber lift (chamber valve = operating valve)

Same data as Field Case #1 of Section 3.4351255 except P_{wh} = 130 psi. Refer to the summary tabulation of Table 3.4108. A kill fluid gradient of .50 psi/ft was

TABLE 3.4108
SUMMARY TABULATION—CHAMBER INSTALLATION—DEEP WELL WITH LOW OPERATING PRESSURE
(Chamber Valve = Operating Valve)

Valve no.	Depth of valve (ft)	Well temp. at valve depth (°F)	P_{so}	Opening pressure at valve depth (PSIG)	Assumed tbng. load at valve (PSIG)	Closing or dome pressure at valve (PSIG)	Dome pressure at 80°F	Surface closing casing pressure (PSIG)	Calc. test rack opening pressure (PSIG)	Final calib.	Remarks
1	1050	102	640	654	400	618	590	604	688.7	690	1/8" Port
2	1600	109	640	662	410	626	590	604	688.7	690	1/8" Port
3	2140	116	640	669	420	633	590	604	688.7	690	1/8" Port
4	2700	123	640	676	425	640	585	604	682.9	680	1/8" Port
5	3250	130	640	684	430	648	585	604	682.9	680	1/8" Port
6	3750	136	640	691	440	655	585	604	682.9	680	1/8" Port
7	4300	142	640	698	445	662	585	604	682.9	680	1/8" Port
8	4800	148	640	705	450	668	580	603	677	680	1/8" Port
9	5350	155	640	712	460	676	580	604	677	680	1/8" Port
10	5900	162	640	720	470	684	580	604	677	680	1/8" Port
11	6450	168	640	727	480	692	580	605	677	680	1/8" Port
12	7000	175	640	735	485	699	580	604	677	680	1/8" Port
13	7500	182	640	741	490	705	580	604	677	680	1/8" Port
14	8000	188	640	748	500	712	580	604	677	680	1/8" Port
15	8397	192	640	753	505	718	580	605	677	680	1/8" Port
16	8449	194	590	697	200	627	510	520	595	600	1/8 × 9/16 (Pilot Valve)

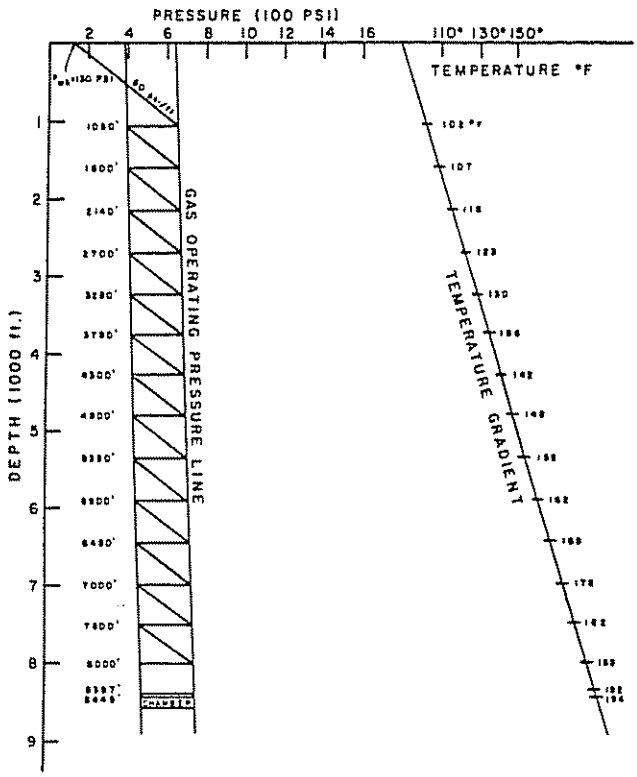


Fig. 3.4149 Chamber Design (Chamber Valve = Operating Valve)

used in this example and all valves were set to open at the surface at 640 psi except the chamber valve, which is the operating valve in this field case and is set at 590 psi. Normally, when the chamber valve is designated the operating valve, it is lowered in pressure to avoid interference from the upper valves. For a large ported chamber valve the fluids reach a high velocity rapidly —causing additional friction in the tubing string— and can open one or two upper valves into the slug if set too high. This can be detrimental in increasing slippage.

Figure 3.4149 shows that a differential spacing was used for this case requiring 16 gas lift valves. The length of the chamber was 104 feet.

All valves were run with a ⅜" port and the chamber operating valve utilized a ⅜" control port in conjunction with a $^{9}/_{16}$" main port size on a pilot valve. Refer to Table 3.4108 for details on valve settings with valves calibrated at 80°F. in the shop (charged with nitrogen).

3.4351247 Summary

Between 12 and 16 valves are needed to reach the bottom of these chamber installations. This is a low system operating pressure for deep wells. Also a thorough analysis of these installations will show that the closing pressures on the upper unloading valves will be lower than the opening pressures on the lower unloading valves, indicating some valve interference. However, since the well should be unloaded intermittently, the upper valves will be given time to close before the next lower valve is opened; therefore, the unloading process should proceed without any trouble.

Chamber installations have been run to 12,000 ft and unloaded successfully. Numerous installations have been run where continuous flow valves were used for unloading and an intermittent valve was run on bottom to operate the chamber. By design the continuous flow valve can close without the pressure having to drop to the dome pressure; therefore, less trouble may be experienced in the unloading process.

3.436 Detailed design for intermittent flow—a method for determining the production rate

3.4361 Introduction

If it is desired to determine the maximum production rate possible from an intermittent well, a method for calculating the average flowing BHP must be used. Such a method was proposed by Brown.[22] In order to use this method, both the static BHP and PI must be known.

The flowing BHP varies from maximum when the gas lift valve is open to minimum when the slug of liquid is removed. These limits will vary somewhat, depending upon whether or not a standing valve has been installed, and will be reduced considerably in a chamber installation compared to a straight tubing installation (see Figs. 3.4150 and 3.4151).

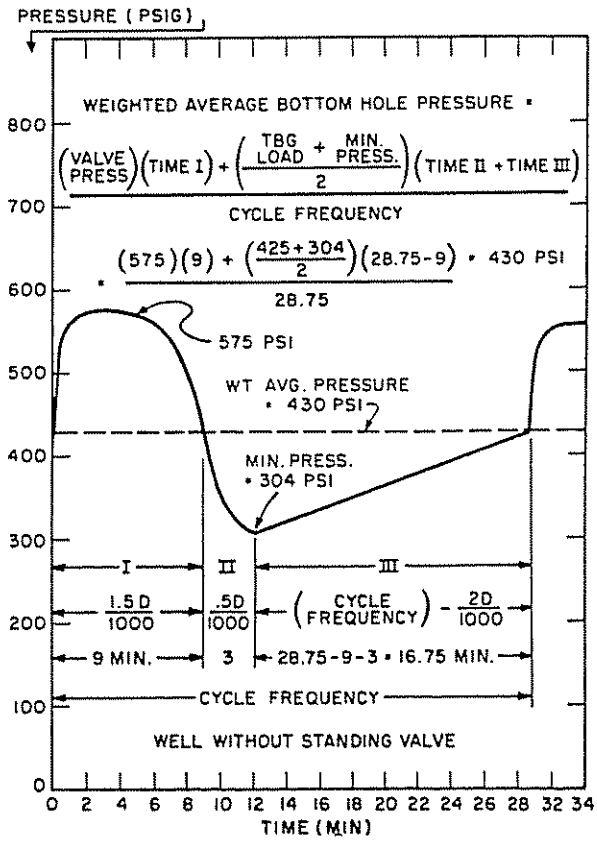


Fig. 3.4150 Time Periods and Weighted Average Flowing BHP for a Typical Intermittent Gas Lift Cycle (after Beadle, et al,[7] Courtesy Jour. Pet. Tech.)

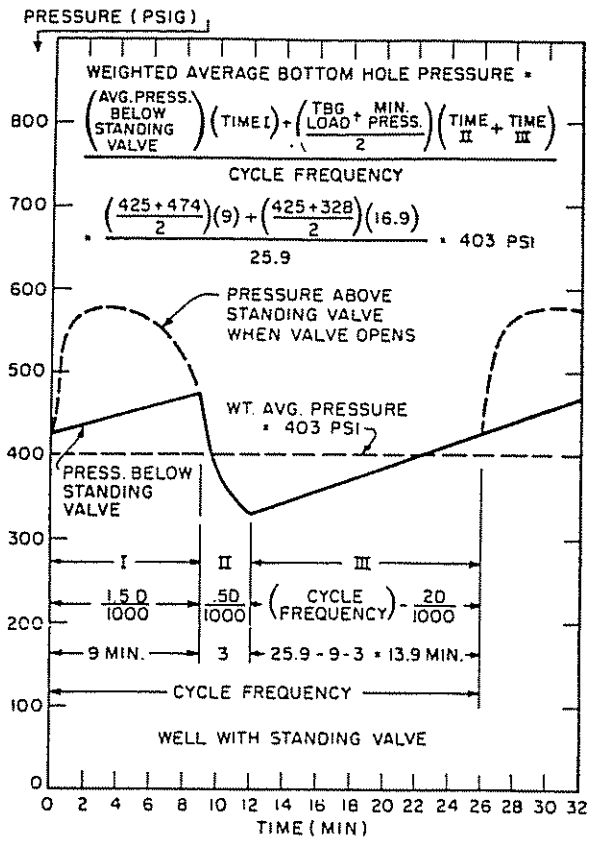


Fig. 3.4151 Weighted Average Flowing BHP with Standing Valve (after Beadle, et al,[7] Courtesy Jour. Pet. Tech.)

3.4362 Calculating the weighted average BHP

This procedure was given by Brown[22] and Beadle, Harlan and Brown.[30] The problem of determining the weighted average BHP in an intermittent well is quite involved. The following factors must be properly accounted for.

(1) Gas pressure at the surface. This is a direct reflection of separator pressure.
(2) Tubing string gas density. This permits a calculation determining the pressure on the top of the liquid slug due to the weight of the gas in the tubing.
(3) Liquid slug size
(4) Tubing size and depth
(5) Gas pressure with which the liquid slug is to be lifted. This includes the gas lift valve opening and closing pressures along with that percentage of the gas pressure that will be placed underneath the slug (not a factor for installations having a standing valve).
(6) Percent recovery expected from the initial liquid slug, and the pressure exerted by this unrecovered liquid.
(7) The input flow characteristics of the well itself

As seen from Figures 3.4150 and 3.4151, there are both a maximum and a minimum pressure created at the sand face. These figures were prepared using the operating gas lift valve as a datum (well lifting from bottom). They were worked for a 6000-ft well with 2" tubing with a 575 psi gas lift valve lifting a total of 425 psi in the tubing. The gradient of the liquid is 0.40 psi/ft and the well has an average PI of 0.3. The tubing back pressure at the surface is 70 psi, giving approximately 90 psi at the top of the slug. Fifty % recovery per cycle has been assumed in this example. Although this example utilizes some simplifying assumptions, this and others were verified by numerous pressure measurements. This method is good enough to permit a reasonably accurate calculation of the weighted average BHP.

One of the more difficult problems is the determination of minimum pressure. It is a function of the back pressure due to the weight of the gas column in the tubing, the pressure exerted by the unrecovered liquids in the tubing string, and the feed-in of fluids from the well. The difficulty of calculating the pressure exerted by the unrecovered liquid as it is scattered up and down the tubing string is complicated because part of this liquid is clinging to the tubing walls and part of it is in the form of a spray in the tubing string.

However, this calculation is necessary since the minimum pressure must occur before all of the liquids have settled back down. The well is still blowing out some spray and gas at the surface at the time the minimum pressure occurs. By making the simplifying asumptions that the gas lift valve opening pressure is placed underneath the slug, that the well fluids continue to feed toward the well bore when the standing valve is closed, that an initial spurt of fluid occurs when the standing valve reopens, and that the unrecovered liquid exerts a pressure at bottom equivalent to the pressure this liquid would exert if it were in a static column, the minimum pressure can be calculated as shown in Figures 3.4150 and 3.4151.

If this procedure of calculating the average BHP gives a pressure which does not allow the desired drawdown in a straight tubing installation, then a chamber installation should be considered for the well. This allows the same amount of liquid to accumulate in the well bore in a much shorter length of pipe. In turn this reduces the pressure exerted by the liquid which reduces the flowing BHP. If a high surface back pressure is to be carried on the well, a chamber should be considered as one possibility of maintaining a lower average flowing BHP.

This average BHP is influenced by either a change in the pressure at the surface or by a restriction incorporated at the surface.

These three examples utilize this procedure.

3.43621 Example problem #1—calculating the minimum BHP that occurs for one cycle of intermittent lift

A typical well with 2" tubing at 5000 ft makes 48 B/D and has 80 psi back pressure at the top of the slug. The minimum GLR occurs at 60% recovery; the liquid gradient is 0.40 psi/ft. A standing valve is installed in this well.

Our problem is to determine the minimum pressure expected at the operating gas lift valve (which is used as a datum level). The operating valve may or may not be installed at bottom, but assume that it is for this example.

The minimum pressure possible for any installation can be approximated in the following manner. Attempting to strive for minimum GLRs, the following stepwise procedure would suffice:

(1) Assume 60% liquid recovery.
(2) Arrive at the fallback of liquid by subtracting the percent of recovery from 100 (100 − 60 = 40%).
(3) Convert the percent of fallback to ft of liquid (assume 1000 ft total) in the tubing string, then convert this height to its resultant pressure by multiplying the fluid gradient by ft of fallback (0.40 × 1000) = 400 ft; (400 × 0.40) = 160 psi.

Although liquid fallback is not complete at the time the minimum pressure occurs, assume that its back pressure is approximately equivalent to the pressure it would exert if in a static column.

(4) Obtain the tubing string gas column weight and select the total weight at the equivalent top of the fallback in static liquid. For example, this is given as 80 psi at the correct depth.
(5) Add the combined pressure of fallback (Step 3) and tubing back pressure (step 4): (80 + 160) = 240 psi. This figure represents the minimum pressure that can be created at the operating valve if the well is not feeding in any liquids. However, this minimum pressure will be somewhat greater, depending upon the well feed-in characteristics.
(6) Determine the added pressure that will be present because of the well feed-in.
   (a) If the well is feeding in at the rate of 48 B/D, the feed-in rate per minute will be 48/1440 = 0.0333 B/min. In 2" tubing this is (0.0333)/(0.00387) = 8.61 ft/min. For a gradient of 0.40 psi/ft this represents a pressure of 3.44 psi/min feed-in rate. This will not be constant, but is assumed to be in this example.
   (b) The minimum pressure in 2" tubing occurs from 1.5 × depth/1000 to 2.25 × depth/1000.
   (c) For a 5000-ft well, taking the minimum pressure occurring at a time of 2 × depth/1000, the pressure due to feed-in rate is:

$$\left(2 \times \frac{5000}{1000}\right)(3.44) = 34.40 \text{ psi}$$

(7) Obtain the total minimum pressure. Step 3 + Step 4 + Step 6(c) = 160 + 80 + 34.4 = 274.4 psi. This represents the minimum pressure that can be obtained for this particular well.

What effect would a chamber installation have on this minimum pressure? Assume that the well is to be placed on chamber lift utilizing a chamber with a 5/1 chamber/tubing volume ratio. Assume 85 psi back pressure at the top of the slug in the chamber.

(1) Convert the ft of fallback to pressure. From Step 3 in Example 1, the pressure due to fallback in the 2" tubing was 160 psi. In the chamber this would be 160/5 = 32 psi.
(2) Add the combined pressure of fallback and tubing back pressure: (85 + 32) = 117 psi, which represents the minimum pressure that can be created at the operating valve with no feed-in of well fluids.
(3) Determine the added pressure that will be present because of Mc well feed-in of fluids. From Step 6 in Example 1 this is 8.61 ft/min in 2" tubing or 3.44 psi/min. In the chamber this would be 3.44/5 = 0.690 psi/min. The pressure due to feed-in is [(2 × 5000/1000)(0.690) = 6.90 psi].
(4) Obtain the total minimum pressure. Add Steps 1, 2, and 3 = 85 + 32 + 6.90 = 124 psi. This compares to a minimum of 274 psi in 2" tubing, showing a difference of 150 psi in minimum pressure created.

3.43622 Example 2—calculating the weighted average BHP for one complete cycle (without standing valve)

A well with 2" tubing and without a standing valve has a gas lift valve located at an operating depth of 6000 ft. The valve opening pressure is 575 psi, and the load to be lifted is 425 psi. The back pressure at the top of the slug is 90 psi. The static BHP is 700 psi. The liquid gradient is 0.40 psi/ft. The average PI is 0.3. Assume 50% recovery.

Determine the weighted average BHP and the production expected (refer to Fig. 3.4150).

(1) Determine the feed-in rate for that period of 1.5 × D/1000 when the slug is traveling to the surface.
   (a) The average BHP for this period = 575 psi
   (b) Drawdown for this period = (700 − 575) = 125 psi
   (c) Production rate for this period = (125)(0.3) = 37.5 B/D
   (d) Production rate = 37.5/1440 = 0.0261 B/min
   (e) Production rate = 0.0261/0.00387 = 6.74 ft/min
   (f) Pressure increase rate = (6.74)(0.40) = 2.696 psi/min
   (g) Total pressure feed-in rate = (2.696)(1.5 × 6000/1000) = 24.3 psi/9 min.
(2) Approximate the minimum BHP as set out in Example 1 to lift a load of 425 psi. The tubing back pressure is 90 psi at the top of the slug.
   (a) Fallback = (100 − 50) = 50% (50% recovery)
   (b) Total ft of liquid in 2" tubing = (425 − 90)/0.40 = 838 ft of liquid
   (c) Ft of fallback = (0.50)(838) = 419 ft
   (d) Pressure exerted by fallback = (419)(0.40) = 168 psi
   (e) Total minimum pressure with no feed-in of fluid is (168 + 90) = 258 psi
   (f) The minimum pressure possible is greater than (258 + 24.3) = 282.3 psi and is approximated to be 305 psi.
   (See step g)
(3) Determine the approximate feed-in rate for the period between the occurrence of minimum pressure at time 2D/1000 min. and 1.5D/1000 min.
   (a) The average pressure for this period will be [(load pressure + minimum pressure)/2] and will cover a time period of 0.5D/1000 minutes. Average pressure = (425 + 305)/2 = 365 psi
   (b) The feed-in rate for this period is (700 − 365) (0.3) = 100.5 B/D = 0.0697B/min = 0.0694/0.00387 = 18 ft/min = 7.2 psi/min

(c) $7.2\left[0.5 \times \frac{6000}{1000}\right] = 21.6$ psi

(4) Determine total minimum pressure: [Back pressure at top of slug + Step 1(g) + Step 2(d) + Step 3(c)] = [90 + 168 + 24.3 + 21.6] = 303.9 psi. Since 305 psi was assumed, this is sufficiently close so that 304 psi can be used as the minimum pressure for 50% recovery.

(5) Determine the time necessary for the pressure to increase from 304 psi to 425 psi due to the average fluid fill-in rate for this period.

(a) Average BHP for this period = (304 + 425)/2 = 364.5 psi

(b) Average fill-in rate = (700 − 364.5)(0.3) = 100.65 B/D = 100.65/1440 = 0.07 B/min = 0.07/0.00387 = 18.1 ft/min = (18.1)(0.40) = 7.22 psi/min

(c) $\frac{425 - 304}{7.22} = \frac{121}{7.22} = 16.75$ min

(6) Determine the total cycle frequency:

$\left[\frac{1.50}{1000} + \frac{0.50}{1000}\right]$ depth + buildup time from minimum pressure to load pressure

$= \left[1.5 \times \frac{6000}{1000}\right] + 16.75 = 9 + 3 + 16.75 =$ 28.75 min

This cycle frequency would, fortunatley, work well on a 30-minute cycle time.

(7) Calculate the weighted average BHP (see Fig. 3.4150)

The weighted average BHP is calculated as follows:

$$\text{Average pressure} = \left[\text{(valve opening pressure)(Time I)} + \left(\frac{\text{tubing load} + \text{minimum pressure}}{2}\right)\text{(Time II)} + \left(\frac{\text{tubing load} + \text{minimum pressure}}{2}\right)\text{(Time III)}\right] \div \text{cycle frequency}$$

$$\text{Avg. pressure} = \left[\text{(valve opening pressure)}\left(1.5 \times \frac{D}{1000}\right) + \left(\frac{\text{tubing load} + \text{minimum pressure}}{2}\right) \times \left(\text{cycle frequency} - 1.5 \times \frac{D}{1000}\right)\right] \div \text{cycle frequency}$$

$$\text{Avg. pressure} = \frac{(575)(9) + [(425 + 304)/2](28.75 - 9)}{28.75} = \frac{5175 + (364.5)(19.75)}{28.75} = 430 \text{ psi}$$

(8) The expected production would be:

$(700 - 430)(0.3) = (270)(0.3) = 81.0$ B/D

3.436221 Reducing the weighted average BHP

How can the weighted average bottom hole pressure be reduced and by how much?

Although the lifting may be rendered somewhat less efficient, the BHP can be lowered by increasing recovery per cycle and possibly be decreasing the load per cycle. Assume that enough gas is utilized to increase the recovery to 70%.

(1) Same as Step 1 in Example 2 (24.3 psi/9 min).

(2) Assume 70% recovery

(a) Fallback = 30%

(b) Same as Step 2(b) in Example 2

(c) Ft of fallback = (0.30)(838) = 251.4 ft

(d) Pressure exerted by fallback = (251.4)(0.46) = 100.56 psi

(e) Total minimum pressure with no feed-in = (100.56 + 90) = 190.56 psi

(f) Minimum pressure = some pressure greater than (190.56 + 24.3) = 214.86 psi. Assume this to be 240 psi.

(3) Determine the approximate feed-in rate for the period between the occurrence of minimum pressure at time 2D/1000 min and 1.5D/1000 min.

(a) Average pressure $= \frac{425 + 240}{2} = \frac{665}{2}$ $= 332.5$ psi

(b) Feed-in rate for this period = (700 − 333)(0.3) = 110 B/D = 0.0763 B/min = 19.7 ft/min = 7.9 psi/min

(c) $(7.9)\left[0.5 \times \frac{6000}{1000}\right] = 23.7$ psi

(4) Determine total minimum pressure:

90 + 100.56 + 24.3 + 23.7 = 238.56 psi

This is close to the assumed value of 240 psi, so use 238 psi.

(5) Determine the time necessary for the pressure to increase from 238 psi to 425 psi.

(a) Average BHP $= \frac{425 + 238}{2} = 331$ psi

(b) Average fill-in rate = (700 − 331)(0.3) = 110.7 B/D

$\frac{110.7}{1440} = 0.077$ B/min;

$\frac{0.077}{0.00387} = 19.90$ ft/min = 7.95 psi/min

(c) $\frac{425 - 238}{7.95} = 23.6$ min

(6) Total cycle frequency = 9 + 3 + 23.6 = 35.6 min

(7) Weighted average BHP $= \frac{(575)(9) + \frac{425 + 233}{2}(35.6) - 9)}{35.6} = \frac{5175 + 8800}{35.6} = 392$ psi

(8) The expected production would be:

(700 − 392)(0.3) = (308)(0.3) = 92.4 B/D

Although the GLR will be greater for this example compared to 2170 scf/B for the example with 50% recovery, there is a total difference in production of (92.4 − 81) = 11.4 B/D. Efficiency may be sacrificed for drawdown to obtain greater recovery. There would be a point of optimum cycle frequency for both maximum production and minimum GLR.

3.43623 Example 3—calculating the weighted average BHP for one complete cycle (with standing valve)

This is the same as Example 2 except for the installation of a standing valve.

Refer to Figure 3.4151 which shows this condition. Again, the designer is concerned with determining the average BHP for any one cycle. However, with the standing valve in place, the full impact of the gas lift valve opening pressure is not reflected against the formation. An approximation is to assume that the proportionate feed-in occurs due to pressure buildup underneath the standing valve; when the standing valve reopens, the total feed-in that would have occurred during the shut-in period occurs suddenly. The feed-in rate can be calculated on this basis. The given conditions again are a 6000-ft well with a gas lift valve opening at 575 psi and a desired load of 425 psi. The static BHP is 700 psi, the gradient is 0.40 psi/ft, and the PI is 0.3.

(1) Estimate the feed-in rate during the period when the standing valve is closed (see Time I in Fig. 3.4151). The pressure at the start of this period is 425 psi and the pressure at the end of 9 minutes is estimated to be 475 psi.
   (a) The average pressure for this period = (425 + 475)/2 = 450 psi.
   (b) Drawdown = (700 − 450) = 250 psi
   (c) Rate = (250)(0.3) = 75 B/D = 0.0521 B/min = 0.0521/0.00387 = 13.5 ft/min = 5.4 psi/min (5.4)(9) = 48.6 psi/9 min. 425 + 48.6 = 473.6 psi
   (d) This is very close to the assumed 475 psi, so use 474 psi as the pressure under the standing valve at the end of 9 minutes.

(2) Approximate the minimum BHP as set out in Example 2. Again the total minimum for no feed-in of fluids would be 258 psi. As set out in steps 2(e) and (f) in Example 2, the minimum pressure possible will then be something greater than (258 + 49) = 307 psi. Assume 330 psi.

(3) Determine the feed-in rate for that period when the minimum pressure occurs between time 2D/1000 minutes and 1.5D/1000 minutes.
   (a) Average pressure for this period = (425 + 330)/2 = 377.5 psi
   (b) The feed-in rate for this period = (700 − 377.5)(0.3) = 96.75 B/D = 0.0672 B/min = 0.0672/0.00387 = 17.35 ft/min = 6.95 psi/min
   (c) $(6.95)\left(0.5 \times \frac{6000}{1000}\right) = 20.85$ psi

(4) Determine that total minimum pressure = (90 + 168 + 49 + 20.85) = 327.85 psi. Since 330 psi was assumed, this 328 psi is close enough and can be used as the minimum pressure for 50% recovery.

(5) Determine the time necessary for the pressure to increase from 328 psi to 425 psi (determined by the average liquid fill-in rate over this period).
   (a) Average BHP = (425 + 328)/2 = 376.5 psi.
   (b) Average fill-in rate = (700 − 376.5)(0.3) = 97 B/D; 97/1440 = 0.0673 B/min = 0.0673/0.00387 = 17.4 ft/min; (17.4)(0.40) = 6.96 psi/min
   (c) $\frac{425 - 328}{6.96} = \frac{97}{6.96} = 13.9$ min

(6) Determine total cycle frequency:
   9 + 3 + 13.9 = 25.9 min
   This compares to 28.75 minutes to feed in the same liquid for the same well without a standing valve.

(7) Calculate the weighted average BHP (see Fig. 3.4151)

$$\frac{\left[\frac{425 + 474}{2}\right]9 + \left[\frac{425 + 328}{2}\right]3 + \left[\frac{328 + 425}{2}\right]\frac{[25.9 - 12]}{25.9}}{25.9}$$

$$= \frac{(4495)(9) + (376.5)(3) + (376.5)(13.9)}{25.9}$$

$$= \frac{4040 + 6370}{25.9} = \frac{10410}{25.9} = 403 \text{ psi}$$

(8) Determine the expected production = (700 − 403)(0.3) = 89.1 B. This is an increase of 9.1 B/D for the same well without a standing valve. This emphasizes the importance of a standing valve.

3.4363 Detailed design of intermittent installation

Given:
Depth = 5000 ft
Tubing size = 2″
Static BHP = 775 psi
PI = 0.30
Liquid gradient = 0.40 psi/ft
Surface tubing pressure = 65 psi
Mean surface temp = 74°F.
Temp gradient = 1°F/100 ft
Temp bottom = 124°F.
Casing size = 5½″; 17 lb/ft
Tubing OD = 2⅜″
Solution GOR = 200 scf/B
Injection pressure = 650 psi
Valve port size = 7/16″
Oil gravity = 39° API
Desired Rate = 65 B/D
Salt water sp gr = 1.07 (37.5%)
Injection gas sp gr = 0.60
Standing valve installed
Well produces 62.5% oil

What oil production can be expected from this well? The well does not have to be killed to pull and run tubing. Plot all information on a sheet of rectangular coordinate paper (see Fig. 3.4152).

(1) Plot the depth scale on the vertical axis with zero depth at the top.
(2) Plot pressure on the horizontal scale.
(3) Plot the static BHP at the correct depth (775 psi at 5000 ft).
(4) Plot the tubing back pressure at zero depth at the top of the page (65 psi).
(5) Determine the static fluid level with no back pressure (775/0.40 = 1940 ft from bottom). Mark this point. Draw a line from the static BHP to the static fluid level on the graph. This is where fluid will be standing to unload the well initial-

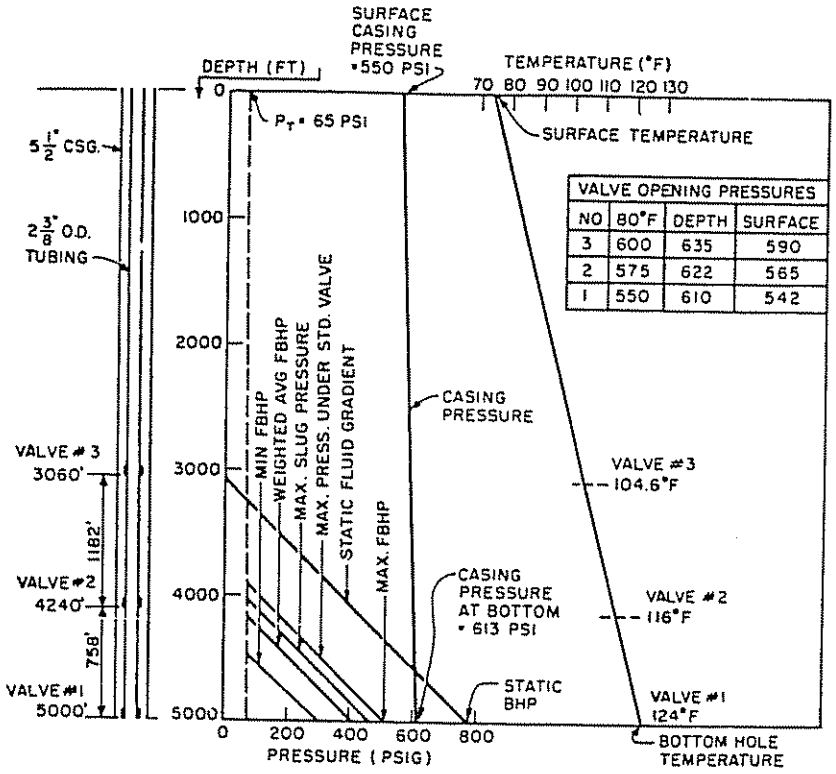


Fig. 3.4152 Intermittent Flow—Detailed Design Example

ly. A valve will be placed here or slightly below with evidence of needing two more valves to unload to the suspected operating point at bottom.

(6) Determine the surface operating pressure using available line pressure less 100 psi as an estimate. This should give ample differential to put the desired gas volume into the well. Plot (650 − 100) = 550 psi at the top of the page.

(7) From Fig. 3A.1 in Appendix 3A, obtain the weight of the gas to be 12 psi/1000 ft or 60 psi/5000 ft. Applying the proper temperature corrections as indicated on Fig. 3A.1, the corrected weight of the gas column is 63 psi. Plot (550 + 63) = 613 psi at bottom and draw a line between 550 psi at top and 613 psi at bottom. Since we know we are to lift near bottom, we can operate there with 610 psi. This still leaves a 100 psi differential at the surface with which to inject gas.

(8) Select a total tubing load of 150 psi less than the gas pressure of 610 psi (valve opening pressure in position). It may or may not be possible to lift this load, depending upon cycle frequency and drawdown. Tubing load = (610 − 150) = 460 psi.

(9) Convert this tubing load to barrels of liquid + tubing back pressure. Mark this pressure as maximum slug pressure on the graph. Extend the gradient line upward from this pressure.

(a) From Fig. 3A.1 select the weight of the gas column in the tubing to be 2.5 psi/1000 = 12.5 psi/5000 ft. Plot this column starting with 65 psi at the surface.

(b) Since where the top of the liquid slug will finally be is unknown, assume 1000 ft from bottom and use a tubing gas column weight of 10 psi. As noted, this will not materially influence the liquid load.

(c) (460 − 75) = 385 psi liquid load

(d) Convert to ft of liquid:

$$\frac{385}{0.40} = 962.5 \text{ ft of liquid}$$

(e) Convert to barrels of liquid:
(962.5)(0.00387) = 3.72 barrels of liquid

(10) Determine the minimum BHP that will be created for a reasonable GLR. For additional production the percent recovery can be increased by increasing the gas volume. See Example 1 for the exact procedure in determining this minimum pressure and Fig. 3.4151 for a typical example.

(a) Estimate the feed-in rate during the time when the standing valve is closed. The initial pressure is 460 psi which is the total tubing load at 5000 ft. Time I is estimated to be (1.5 × D/1000) = 7.5 min. The pressure is estimated to be 510 psi at the end of 7.5 min.

(1) The average pressure for this period = (460 + 510)/2 = 485 psi

(2) Drawdown = (775 − 485) = 290 psi

(3) Production rate = (290 × 0.3) = 87 B/D = 0.0604 B/min = 15.6 ft/min = 6.24 psi/min = 46.8 psi/7.5 min

(4) This is close to the assumed value, so use (460 + 46.8) = 507 psi as the pressure under the standing valve at the end of 7.5

min. Locate this pressure on the graph in Fig. 3.4152.

(b) Approximate the minimum BHP. Assume 60 percent recovery.
  (1) Fallback = (100 − 60) = 40%
  (2) Ft of liquid = 962.5 ft
  (3) Ft of fallback = (0.40)(962.5) = 386 ft
  (4) Pressure exerted by fallback = (0.40)(506) = 154 psi
  (5) Total minimum pressure without any feed-in = (154 + 75) = 229 psi
  (6) The minimum pressure is somewhat greater than (229 + 47) = 276 psi and is approximated to be 295 psi.

(c) Determine the approximate feed-in rate for the period between the occurrence of minimum pressure at 2D/1000 min and 1.5D/1000 min.
  (1) The average pressure = (460 + 295)/2 = 377.5 psi
  (2) The feed-in rate for this period = (775 − 377.5)(0.3) = 119.75 B/D = 0.083 B/min = 21.4 ft/min = 8.57 psi/min = (8.57)(2.5) = 21.4 psi

(d) The total minimum pressure = (75 + 154 + 47 + 21.4) = 297.4 psi. This checks with the assumed value of 295, so use 297 psi (plot on Fig. 3.4152).

(e) Determine the time for the pressure to increase from 297 psi to 460 psi and determine the average fill-in rate for this period. Average BHP = (297 + 460)/2 = 378.5 psi. The average fill-in rate = 8.45 psi/min, so fill-in time = (460 − 297)/8.45 = 163/45 = 19.4 minutes.

(f) Total cycle frequency = (7.5 + 2.5 + 19.4) = 29.4 min (use 30-minute cycles). This would be the cycle frequency to lift 460 psi loads, which can be lifted efficiently with 610 psi at that depth.

(11) Determine the weighted average BHP using operating valve depth as a datum:

$$\text{Avg pressure} = \frac{[(460 + 507/2]7.5 + [(460 + 297)/2] \times 2.5 + [(297 + 470/2][29.4 - 10]}{29.4}$$

$$= \frac{3620 + 8300}{29.4} = \frac{11920}{29.4} = 406 \text{ psi}$$

Refer to Fig. 3.4151 for a typical example.

(12) Determine the expected productin: (775 − 406)(0.3) = 110.7 B/D. This would allow a production of (0.625)(110.7) = 69.4 B/D of oil and would handle the 65 B/D desired production. If the production were not in the desired range, a chamber installation would be considered to create more drawdown, thereby increasing production. Increased production could also be obtained by greater gas injection per cycle or more cycles/day.

(13) Determine the total gas volume necessary. Refer to Fig. 3G.1 in Appendix 3G.

(14) Check to see if sufficient stabilization time is available for the 30-minute cycle frequency. (See Figures 3.4132 through 3.4136).

(a) The position of the top of the slug = (5000 − 962.5) = 4037.5 ft.

(b) The stabilization time for 60% recovery: ΔP = 150 psi, and with a $^7/_{16}$ in. port = 21 min. This is sufficient time for good 30-minute cycles.

(15) Select the position of the operating gas lift valve to give the desired drawdown.
  (a) The weighted average flowing BHP was 406 psi.
  (b) The weighted average BHP from valve location is also 406 psi, since the valve must be located on bottom to create this flowing pressure.
  (c) The valve, of course, would be placed on bottom in this case for the well to produce 111 B/D.

(16) Make operating valve selection.
  (a) Draw in temp gradient line 74°F. at top and 124°F. at bottom. See Fig. 3.4152.
  (b) Select temp opposite valve depth (124°F for this example).
  (c) From Fig. 3D.9 in Appendix 3D, select a valve that will operate at 610 psi opposite the valve at 5000 ft and at 124°F. Valve setting at 80°F. = 550 psi (balanced valve).
  (d) Surface opening pressure = 542 psi.
  Since the intention is to select a valve to operate at 550 psi at the surface, our selection is close enough (this is valve 1).

(17) Space the gas lift valves. (Use balanced valves set at 80°F).
  (a) Place the top valve at the static liquid level with no back pressure, or (5000 − 1940) = 3060 ft.
  (b) Using a 0.04 psi/ft unloading gradient for a $^7/_{16}$" ported valve, space additional valves as necessary.
  (c) Increase each valve up the hole some 25 psi for spacing purposes.
  (d) Using a pressure of 660 psi at 3060 ft, the next lower valve is set at 630 psi with the lowermost valve opening at 610 psi. Using the spacing formula, [660 − (3060)(0.04) − 65]/0.40 = 473/0.40 = 1182 ft. Second valve down will be at (3060 + 1182) = 4242 ft. Checking the bottom valve, [630 − (4242)(0.04) − 65]/0.40 = 396/0.40 = 990 ft. Since the required distance is only 758 ft, this is satisfactory. The valve depths will be 3060 ft, and 5000 ft.
  (e) Check the opening pressure opposite the second valve for a 25 psi change in pressure setting. The temperature at this point is 116°F. For a valve set at 575 psi at 80°F, the opening pressure opposite the valve is 622 psi, equivalent to a surface opening pressure of 565 psi. This will be satisfactory for a large port at bottom since it will pass the gas without any valve interference. Valve 3 would be selected in a similar manner and would be set at 600 psi at 80°F. The temperature opposite the valve at 3060 ft is 104.6°F. The opening pressure opposite the valve is 635 psi and the surface opening pressure is 590 psi. For a smaller port a greater difference should be taken

to eliminate any valve interference when gas is injected (note the complete design in Fig. 3.4152).

Discussion

The previous design example was for a 7/16" ported valve. If a different size port is to be used, the design should be changed to account for this difference. For example, if a 5/8" port can be utilized on bottom, the GLR can be decreased and, if a well requires it, additional injection cycles are possible since a reduction in pressure stabilization time can be realized for the same amount of gas used for the smaller port. If a smaller port such as 5/16" is used, the GLR will increase and the pressure stabilization time will increase for the same gas volume used with the 7/16" port. Unbalanced valves can be used in this example and would perform equally as well.

This example is long and tedious for longhand calculations but lends itself readily to computer application.

3.437 Effect of variables in intermittent lift

3.4371 Detailed design for intermittent flow—effect of variables.

In order to look at the effect of such variables as port size, differential between casing pressure, tubing load, and PI, the following problem was worked by Deschner and Brown.[31]

Given data:
- Tubing sizes = 1¼" and 2⅞" OD
- Casing size = 5½" (17 lb/ft)
- P_{wh} = 70 psig
- Surface temp = 74°F.
- Temp gradient = 1.6°F./100 ft
- R_s = 400 scf/B
- Surface operating pressure = 800 psig
- γ_g = 0.65
- Well depth = 6000 ft
- $\overline{P}_R$ = 1000 psi
- Producing 100% oil
- Standing valve installed

3.43711 Effect of differential between valve pressure and tubing load

The summary tabulation of Table 3.4109 shows the effect of changing the differential across the valve from 150 to 300 psi. By removing the liquid slugs with a δp of 300, it is possible to maintain a much lower average BHP, thereby obtaining greater production. However, the GLR increases about 700 scf/B in each case. The profit on the additional oil production will more than offset the cost of the additional gas.

The best method for lifting this well is not where the minimum GLR occurs but at a point of optimum gas injection. For wells where a greater production rate is desired, a chamber installation should be considered to create an even lower average flowing pressure.

TABLE 3.4109
EFFECT OF DIFFERENTIAL ACROSS VALVE
J = 0.1; PORT SIZE = ½"

Δp	Minimum pressure at valve	Weighted average pressure	Production rate (B/D)	Cycle frequency (minutes)	Cycles per day	GLR (scf/B)
150	447	615	38.5	139	10	1570
300	271	458	54.2	109	13	2270
		J = 0.4; port size = ½ in.				
150	501	670	132	41	35	1610
300	358	535	186	32	45	2260

3.43712 Effect on PI

Refer to Table 3.4110 to note the effect of PI. Several things are noted as the PI changes. The production rate is most important and increases from 38.5 B/D for a PI of 0.1 to 132 B/D for a PI of 0.4. The weighted average BHP increases from 615 to 670 psi because of the faster feed-in rate of fluids. However, the higher PI increases the production rate. As noted, a well with a PI of 0.1 needs to be cycled only 10 times per day, while the well with a PI of 0.4 must be cycled 35 times per day.

TABLE 3.4110
EFFECT OF PI
Δp = 150; port size = ½"

PI	Minimum pressure at valve (psig)	Weighted average pressure (psig)	Production rate (B/D)	Cycle frequency (minutes)	Cycles per day
0.1	447	615	38.5	139	10
0.2	480	642	71.4	74	20
0.4	501	670	132.0	41	35

3.43713 Effect of valve port size

The port size multiplying factors of Table 3.4111 may be used to change the gas volumes and pressure stabilization times for port sizes other than 7/16".

TABLE 3.4111

Port size (in.)	Factor
7/16	1.00
5/16	1.10
½	0.975
1	0.75

In conclusion, as large a port size as possible should be used in intermittent lift operations for single point injection; the well cycle frequency should be changed until the maximum recovery is obtained. If gas use becomes so excessive that its cost exceeds the additional recovery value, the cycle time should be increased.

3.4372 Computer solution to the problem

Deschner and Brown presented a computer solution to the intermittent flow problem to determine the effect

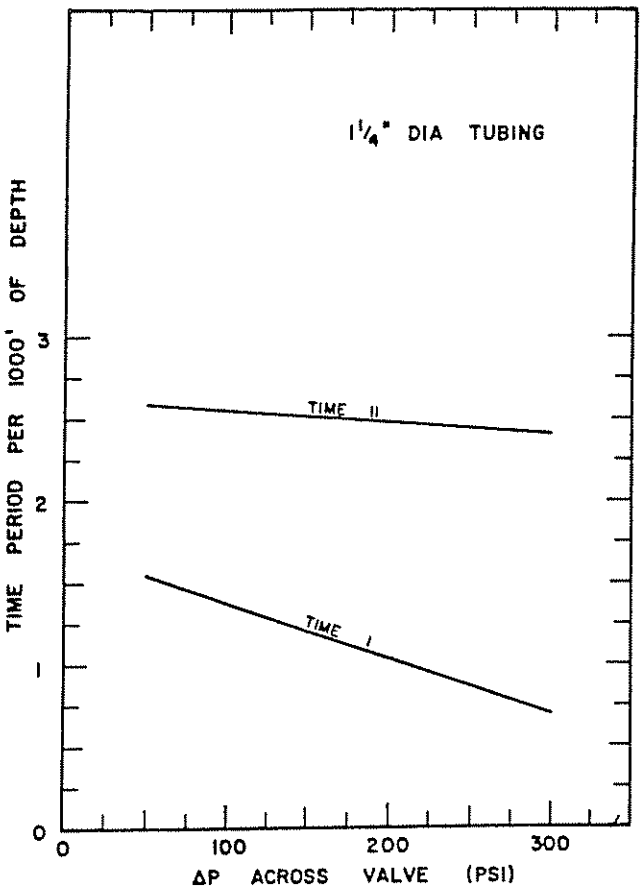


Fig. 3.4153 Time Period Per Thousand ft. of Depth vs. ΔP Across Valve for 1¼-in. Tubing (after Deschner & Brown,[9] Courtesy API)

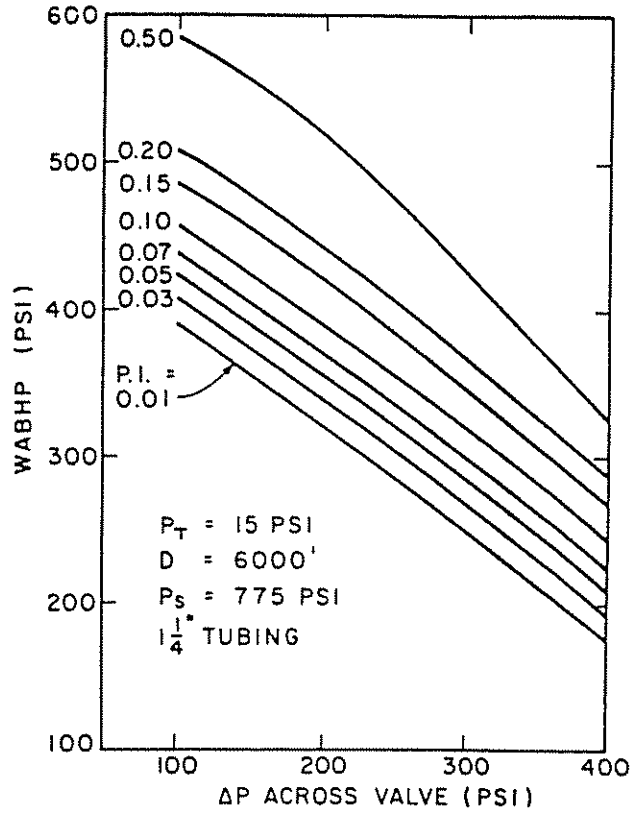


Fig. 3.4154 Weighted Average BHP vs. ΔP Across Valve (after Deschner & Brown,[9] Courtesy API)

of certain variable.[31] The following problem was worked:

Given data:
- Depth = 6000 ft
- Tubing diam = 1¼" and 2"
- Sp gr of fluid = 1.07
- Sp gr of gas = 0.65
- PI = 0.01 – 0.5 B/D/psi
- Static reservoir pressure = 775 psi
- Surface casing pressure = 550 psi
- Surface tubing pressure = 15, 50, 100, and 200 psi
- Pressure differential across valve = 100, 200, 300 and 400 psi
- Standing valve installed in all situations
- Fallback = 7%/1000 feet
- Time periods: for 1¼" (see Fig. 3.4153)
- For 2": Time I = 1.5 × D/1000
  Time II = 0.5 × D/1000
- Various figures were prepared to illustrate the effect of the variable involved.

3.43721 Effect of differential and PI

Figure 3.4154 shows the effect of pressure differential across the valve and PI on the weighted average BHP. For a constant PI, as Δp increases, the pressure at the valve decreases. For a given Δp the average BHP increases as the PI increases due to the higher fluid feed-in rate. Figure 3.4155 also shows the effect of differential and PI on the average flowing BHP.

3.43722 Effect of depth

The effect of depth on fluid production rate and BHP is noted in Figure 3.156. As expected, it is more difficult to maintain the same production rate as the depth increases. This is due to the fact that the average flowing pressure increases with depth.

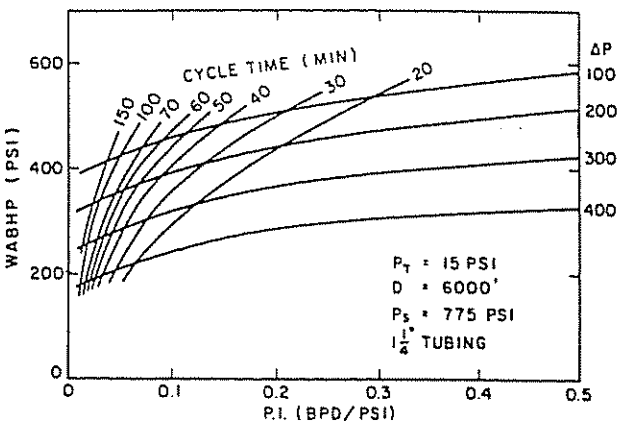


Fig. 3.4155 Weighted Average BHP vs. PI (after Deschner & Brown,[9] Courtesy API)

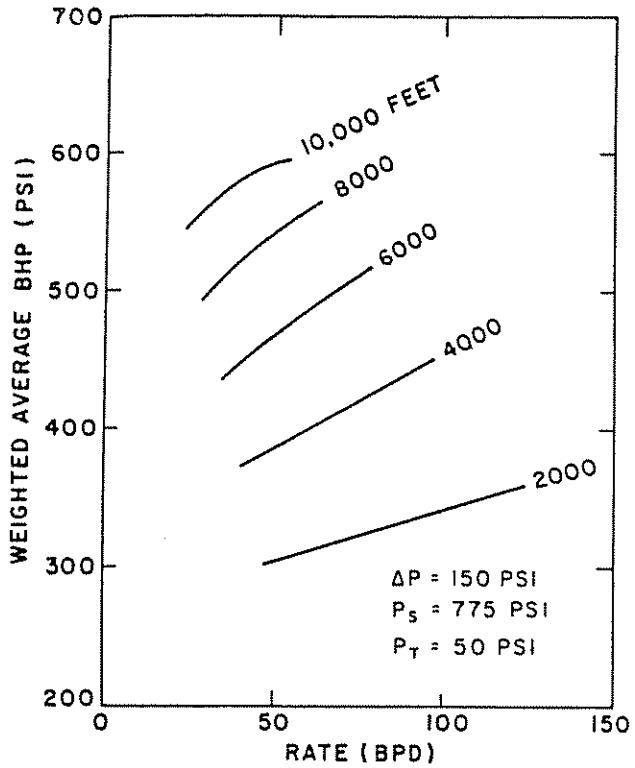


Fig. 3.4156 Weighted Average BHP vs. Production Rate (after Deschner & Brown,[3] Courtesy API)

3.43723 Prediction of BHPs

By using a computer solution, figures such as 3.4157 and 3.4158 can be easily prepared to predict weighted average BHP and cycle frequency for a specific installation. For a particular tubing size and Δp, the PI may be used to obtain the weighted average BHP. The cycle time may then be found by interpolation. The rate is calculated from the drawdown created. If it is not high enough, the Δp across the valve should be increased.

Table 3.412 shows the results for a particular condition of P_{wh} = 50 psi, $\overline{P}_R$ = 775 psi, depth = 6000 ft, and PI = 0.10 (assume linear).

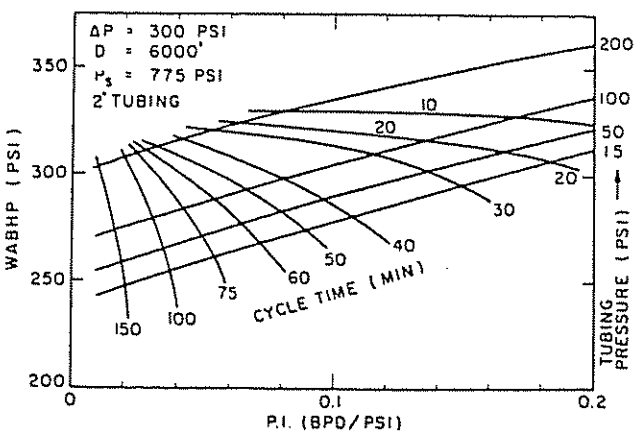


Fig. 3.4157 Weighted Average BHP Correlation for 2-in. Tubing (after Deschner & Brown,[3] Courtesy API)

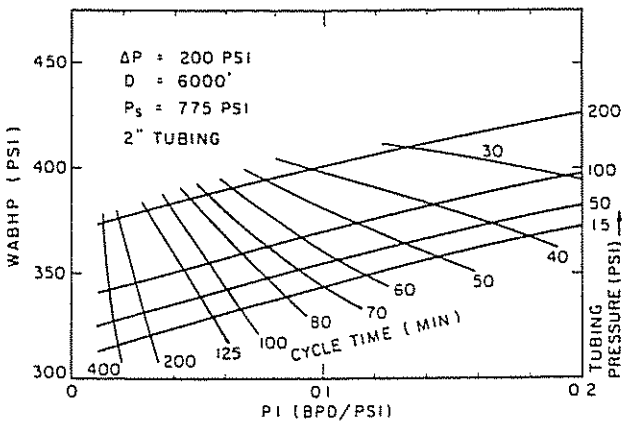


Fig. 3.4158 Weighted Average BHP Correlation for 2-in. Tubing (after Deschner & Brown,[3] Courtesy API)

TABLE 3.4112
CALCULATED WEIGHTED AVERAGE BHP AND CYCLE TIME:
P_{wh} = 50 psi, $\overline{P}_R$ = 775 psi; D = 6000 ft; PI = 0.10 B/D/psi

Δp across valve (psi)	1½ in. tubing			2 in. tubing		
	WABHP (psi)	Rate (B/D)	Cycle time (min)	WABHP (psi)	Rate (B/D)	Cycle time (min)
100	467	30.8	51	419	35.6	92
200	400		33	354	42.1	62
300	328	44.7	220	290	48.5	39
400	250		(too fast)	224	55.1	22

The 2″ tubing produces more fluid than the 1¼″ tubing for all differentials across the valve.

For the 2″ tubing a 400 psi Δp load would require 65 cycles per day, recovering 55.1 B/D for 65 C/D (cycles per day) = 0.85 B/C. This would represent the maximum production from this low productivity well for these conditions. From economic considerations, however, this rate may not be the optimum due to the high input GOR required.

3.438—Class problems: intermittent design problems

(1) Given Data:
- 2⅜″ O.D. tubing
- P_{so} = 800 psi
- γ_g = 0.65
- $\overline{P}_{wh}$ = 60 psi
- $\overline{P}_R$ = 1200 psi
- One test shows q_o = 80 B/D for P_{wf} = 500 psi
- P_b = 1500 psi (bubble point pressure)
- 5½″ casing
- 7000 ft = depth
- Temp bottom = 160°F.
- Temp surface = 80°F.

There is no additional pressure available for unloading the well. The well is loaded with fluid to the surface for work-over, and the gradient of this fluid is .45 psi/ft. Design a complete installation for intermittent lift for both choke control and time cycle control.

(a) Bellows charged pressure operated valves—nitrogen charge at 60°F (25 psi drop in pressure between valves at surface)
(b) Spring charged pressure-operated valve—(25 psi drop in P_{so} between valves)
(c) Fluid-operated valves—bellows charged
(d) Fluid operated—Spring-charged valve
(e) Combination pressure closed and fluid opened valve.
(f) Design a chamber for the above well using two packers.
(g) Repeat (a) and (b) using constant surface closing pressure.

(2) Given data:
2⅜" O.D. tubing
P_{so} = 900 psi = P_{ko}
Max estimated rate = 150 B/D (100% oil)
Gradient of kill fluid = 0.40 psi/ft
$\gamma_g = 0.7$
Depth = 8000 ft
Temp bottom = 180°F
Temp surface = 100°F
7" casing
P_{wh} = 50 psig

(a) Design a multipoint system for:
(1) Fluid operated valves—spring charge
(2) Fluid operated valves—bellows charge with nitrogen at 60°F
(b) Opti-flow design with standard pressure operated valves for both time cycle and choke control.

(3) Given:
7" casing (23#/ft)
P_{so} = 650 psi
P_{ko} = 700 psi
q_o = 200 b/d max
Depth = 5000 ft
Temp Surface = 80°F.
Temp Gradient = 1.3°F/100 ft
Well loaded to surface with salt water – Gradient = .465 psi/ft
2⅜" O.D. Tubing
P_{wh} = 100 psi
100% oil
$\gamma_g = 0.65$

Design for:

(a) Choke control—pressure operated valves
(b) Time cycle control—pressure operated valves
(c) Opti-flow—pressure operated valves
(d) Choke control—fluid operated valves
(e) Combination valves (pressure closed—fluid opened)—Choke control
(f) Design a chamber for this well using standard pressure operated valves (standard 2-packer chamber)
(g) Design an insert chamber for this well

(4) Given:
Depth = 8500 ft
2½" tubing
Rate = 150 B/D
P_{so} = 900 psi
$\gamma_g = 0.65$
Gradient of kill fluid = 0.50 psi/ft and liquid level is at 4000 ft. from bottom.
P_R = 1600 psi
PI = .01
P_{wh} = 80 psig
Temp bottom = 160°F.
Temp surface = 100°F.
P_{ko} = 950 psi

Use time cycle control and design with:
(a) Pressure valves—spring charged
(b) Pressure valves—bellows charged (80°F)

Use choke control and design with:
(a) Pressure valves—bellows charge (nitrogen at 80°F.)
(b) Fluid valves for multi-point injection—(spring charged)
(c) Pressure valves—multipoint injection

(5) Given:
Depth = 6000 ft
Temp at 6000 ft = 150°F.
Temp surface = 80°F.
Expected max rate = 100 B/D oil
2½" tubing
7" casing
Kill fluid gradient = 0.50 psi/ft
P_{ko} = 800 psi
P_{so} = 700 psi

Design for choke control and time cycle control:
(a) Space analytically, taking 25 psi drop in surface operating pressure between valves using pressure operated nitrogen charged valves at 80°F.
(b) Space graphically, using constant surface closing pressure (pressure operated valves).

(6) Given: (Intermittent problem)
Casing = 7"
Tubing = 2⅜" O.D.
Expected Rate = 100 B/D
8000 ft
Temp bottom = 170°F.
Temp top = 100°F.
P_{so} = 800 psi
$\gamma_g = 0.65$
Tubing load = 600 psi
G_s = 0.40 psi/ft

Design an intermittent installation for both "choke control" and "time cycle control."
(a) Standard pressure valves—25 psi drop between valves
(b) Fluid operated valves
(c) Pressure operated valves—multipoint injection
(d) Design a chamber installation, setting all valves for constant surfce opening pressure of 800 psi.
(1) Chamber valve = operating valve
(2) Valve immediately above chamber valve = operating valve

(7) Given: (Intermittent well)
2⅜" O.D. tubing
5½" 17#/ft casing
P_{so} = 550 psi
Top of 600 ft perforated interval = 4638 ft

$\overline{P}_R$ = 400 psia
γ_g = 0.7
Separator pressure = 50 psi
G_s = .40 psi/ft
Well loaded to surface
Rate less than 100 B/D
Temp bottom = 130°F.
Temp top = 80°F.

(a) Design an insert chamber for this well, using a 1" dip tube and allowing the dip tube to extend one joint (30 ft) up into the 2" tubing above the chamber.
(b) For 7" casing repeat the design.

(8) Given:
3000 ft
P_{so} = 500 psi
P_{wh} = 50 psi
G_s = 0.40 psi/ft
2⅜" O.D. tubing × 5½" (17#/ft) casing
$\overline{P}_R$ = 700 psi
PI = 0.15
γ_g = 0.65

Space unloading valves and design a chamber for this well.

(9) Given:
P_{so} = 700 psi
γ_g = 0.65
Expected Rate = 200 B/D
Static press = unknown
Well loaded to surface (G_s = 0.465)
Depth = 9000 ft
2⅜" O.D. × 7" (26#/ft) casing

Design a chamber installation without losing pressure with two options:
(a) Chamber valve = operating valve
(b) Valve immediately above chamber valve = operating valve

3.44 Comparisons of continuous, standard intermittent and chamber gas lift methods

3.441 Introduction

For the purpose of comparing continuous, intermittent and chamber gas lift methods, Esliat developed computer programs for each of these methods using the same input data.[32] The maximum capacity for intermittent lift was assumed to occur when the cycle time reached $\frac{(1.0)(\text{Depth})}{1000}$. This assumes that enough time has elapsed prior to starting the next slug but does not leave time for the tail spray to clear the tubing.

The effect of changing variables in these methods was investigated. Input data were divided into two groups (fixed and variable input data).

Fixed data:

Gravity of oil	= 35° API
Specific gravity of gas	= 0.65
Surface temperature	= 80°F.
Bottom hole temperature	= 180°F.
Water specific gravity	= 1.0
Water cut	= 35%

Variable data:

This data was kept constant at the following values when possible; at times it was changed to illustrate the effect of certain variables. Slippage factors for intermittent gas lift of 7% per 1000 feet and 5% per 1000 feet were used.

Injection gas pressure = 800 psia
Wellhead pressure = 100 psia
Differential pressure across the valve = 200 psi
Static bottom hole pressure was calculated as a function of depth (0.1 × depth in psi) normally, but values up to 2500 psi were used.
Depths of 5000, 7000, 8000, and 9500 feet were used.
Tubing sizes of 1¼", 1½", 2.0", 2½", 3.0", and 3½" were used.
Two different casing sizes were used: 5½" and 7.0".

To compare different values of static bottom hole pressure, wellhead pressure, productivity index, tubing size, differential pressure across the valve, slippage factor, and injection gas pressure, a typical well was selected. This well was 8000 feet deep, with 2½" tubing and 7.0" casing. Values of static pressure were varied from 600 to 2500 psi, and for each of these values several different values of productivity indices were considered.

3.442 Effect of changing static pressure

To observe the effect of change in static bottom hole pressure on the production rate, the results are plotted in Figure 3.4159. These results are tabulated in Table 3.4113.

For a constant valve of productivity index, an increase in static bottom hole pressure increases the total production rate.

Chamber lift is the most efficient method to use for a static bottom-hole pressure of 2000 psi or less and for productivity indices of 0.5. Continuous gas lift should be used for static bottom-hole pressures greater than 2000 psi for PI = 0.5. Of course, higher PI's at

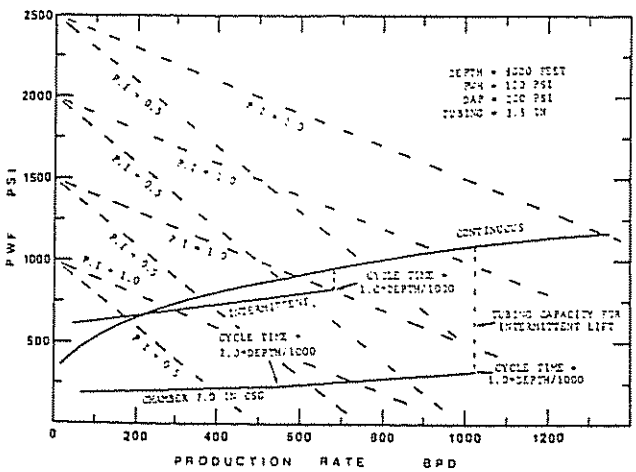


Fig. 3.4159 Comparison of Gas Lift Methods for Different Static Bottom-Hole Pressures

TABLE 3.4113
COMPARISON OF PRODUCTION RATES FOR CHANGES IN STATIC BOTTOM-HOLE PRESSURE

PI = 0.5 (constant)

	Continuous		Intermittent		Chamber	
$\overline{P}_R$ (psi)	P_{wf} (psi)	Rate (BPD)	P_{wf} (psi)	Rate (BPD)	P_{wf} (psi)	Rate (BPD)
800	585	108	635	83	215	293
1000	638	181	650	175	225	388
1500	800	350	730	385	250	625
2000	900	550	800	600	300	850
2500	988	756	•	•	•	•

*Intermittent lift reached its maximum production capacity before a value of productivity index of 0.5 B D/psi.

higher static pressures are better suited for continuous flow.

For values of static bottom-hole pressure less than 2000 psi, continuous lift shows the highest flowing bottom-hole pressure and, therefore, the lowest production rate.

Intermittent lift shows higher flowing bottom-hole pressures for rates less than 200 B/D.

3.443 Effect of wellhead pressure

The wellhead pressure has a decided effect on the total production of a well. This is shown in Figures 3.4160, 3.4161 and 3.4162 for wellhead pressures of 50, 100, and 150 psi.

With the increase in wellhead pressure, the total production decreases and the flowing bottom hole pressure increases. Results are tabulated in Table 3.4114.

As noted, chamber lift is the most efficient method because it gives the lowest flowing bottom hole pressure in all cases illustrated. An increase in the wellhead pressure leads to a decrease in total recovery. When the wellhead pressure is 50 psia, the maximum recovery for a chamber is 1115 B/D; when the pressure increases to 150 psia, the maximum recovery drops to 930 B/D (see Table 3.4115).

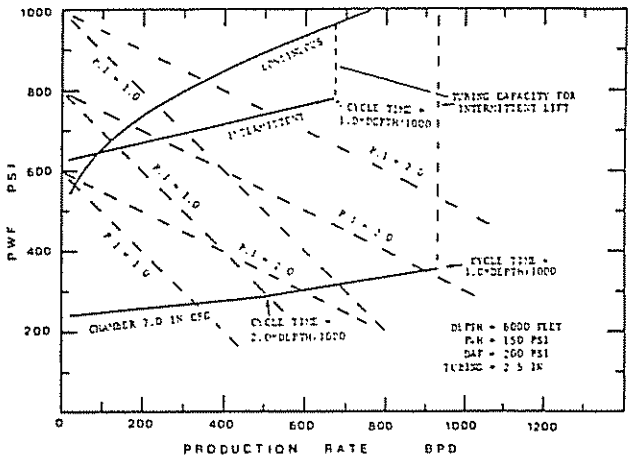


Fig. 3.4161 Comparison of Gas Lift Methods for a Wellhead Pressure of 150 PSIA

For a wellhead pressure of 50 psia, continuous lift has a lower pressure up to a production rate of 340 B/D. This difference will decrease with an increase in wellhead pressure. When the wellhead pressure is 150 psia, continuous lift has a lower flowing bottom hole pressure up to a production rate of 90 B/D. For higher rates than specified, continuous lift has a higher

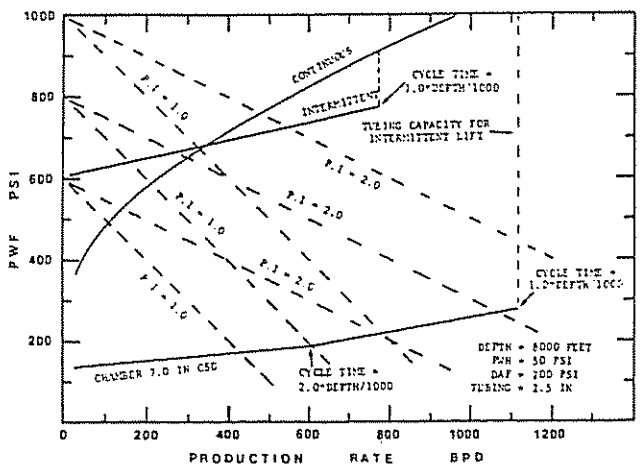


Fig. 3.4160 Comparison of Gas Lift Methods for a Wellhead Pressure of 50 PSIA

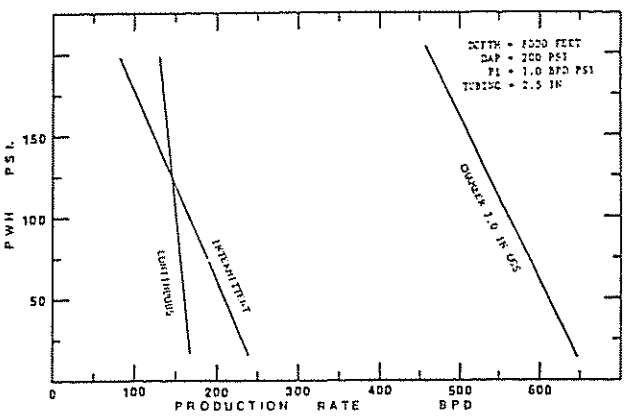


Fig. 3.4162 Comparison of Gas Lift Methods for a Wellhead Pressure of 200 PSIA

TABLE 3.4114
COMPARISON OF PRODUCTION RATES FOR CHANGES IN WELLHEAD PRESSURE

PI	Continuous		Intermittent		Chamber	
(B/D/psi)	P_{wf} (psi)	Rate (B/D)	P_{wf} (psi)	Rate (B/D)	P_{wf} (psi)	Rate (B/D)
P_{wh} = 50 psi						
1.0	590	210	640	160	190	610
2.0	650	295	660	280	270	1060
P_{wh} =100 psi						
1.0	635	165	650	150	240	560
2.0	685	235	670	260	310	980
P_{wh} = 150 psi						
1.0	677	123	660	140	290	510
2.0	715	170	680	240	355	890

TABLE 3.4115
COMPARISON OF TOTAL PRODUCTION RATES BY GAS LIFT METHODS

P_{wh}	Continuous		Intermittent		Chamber	
(psi)	P_{wf} (psi)	Rate (B/D)	P_{wf} (psi)	Rate (B/D)	P_{wf} (psi)	Rate (B/D)
50	800	525	775	770	280	1115
100	800	440	778	720	318	1021
150	800	300	780	660	360	930

flowing bottom hole pressure than straight intermittent lift. For all gas lift methods, an increase in wellhead pressure decreases the total production rate.

Results in Table 3.4115 show maximum capacities for intermittent lift and a chamber when the cycle time is 1.0 × Depth/1000, and for continuous flow for a flowing pressure of 800 psi.

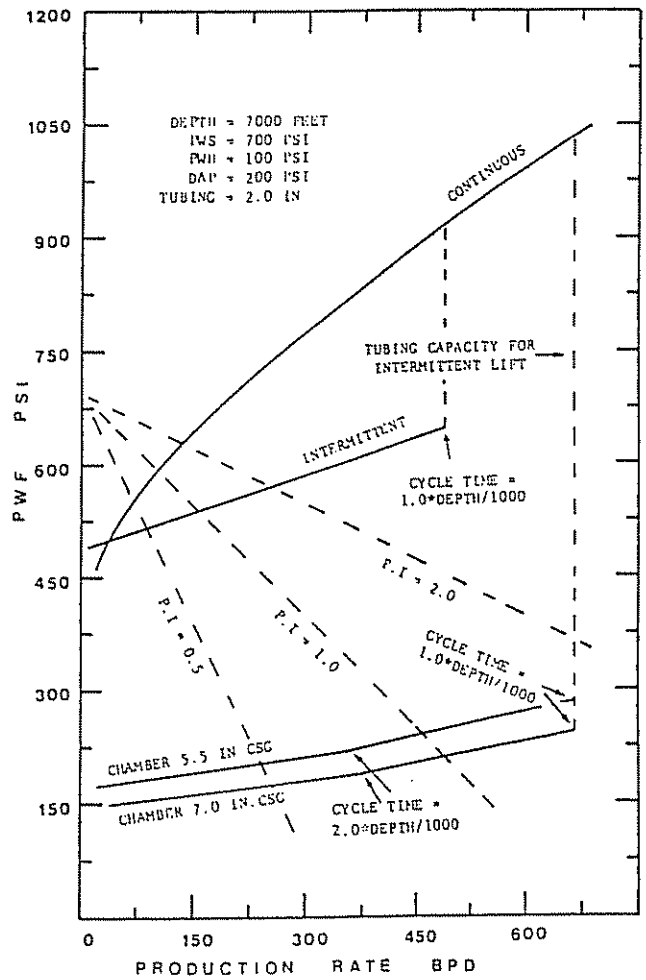


Fig. 3.4163 Comparison of Gas Lift Methods To Show Rates Possible from 7000-ft. with 2.0-in. Tubing

3.444 Effect of productivity index

For a constant static bottom hole pressure, the flowing bottom hole pressure increases with an increase in productivity index. For a given productivity index a lower flowing bottom hole pressure can be obtained with a lower static bottom hole pressure.

Typical results are shown in Figures 3.4163 and 3.4164 and are tabulated in Table 3.4116.

For a constant value of static bottom hole pressure, a higher production rate is obtained with chamber lift for small values of productivity index.

Use continuous gas lift for high values of PI and high static pressures.

Chamber lift is greatly affected by changes in productivity index, as shown in Figure 3.4163, i.e. for a value of PI of 0.1, a rate of 65 B/D is obtained. When the PI is 1.0 B/D/psi, the total production is 660 BPD; the maximum capacity for intermittent lift is reached before the PI becomes 2.0.

3.445 Effect of tubing size

An increase in the tubing size increases the production rate, keeping other variables constant. This is shown in Figures 3.4165 through 3.4167.

The value of static bottom hole pressure is maintained constant at 800 psi. Results are tabulated in Table 3.4117 and are shown in Figure 3.4168.

In some cases the tubing size is not large enough to handle the fluids.

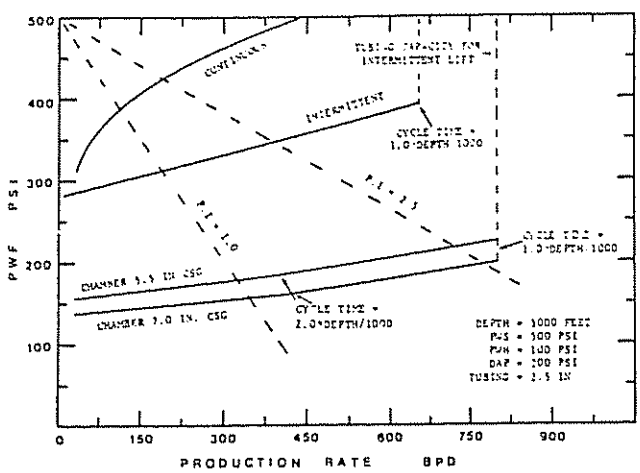


Fig. 3.4164 Comparison of Gas Lift Methods to Show Rates Possible from 5000-ft. with 2.5-in. Tubing

TABLE 3.4116
COMPARISON OF PRODUCTION RATES FOR CHANGES IN PRODUCTIVITY INDEX
Depth = 5000 feet

	Continuous		Intermittent		Chamber (7")	
PI (B/D/psi)	P_{wf} (psi)	Rate (B/D)	P_{wf} (psi)	Rate (B/D)	P_{wf} (psi)	Rate (B/D)
0.5	385	55	305	98	140	180
1.0	415	85	325	175	160	340
3.0	455	135	375	375		*

*The chamber lift reached its maximum production capacity before a value of PI = 3.0 B/D/psi.

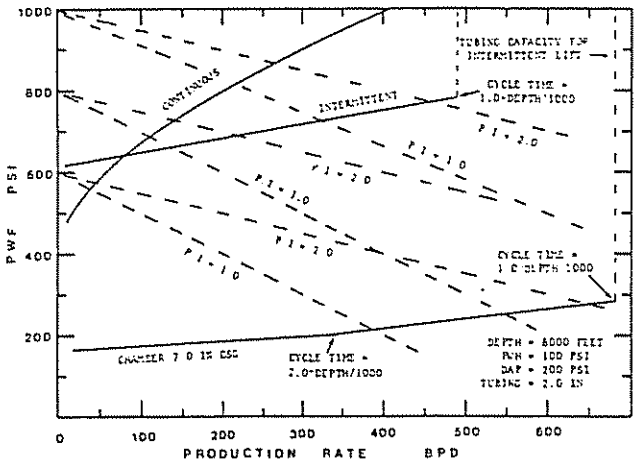


Fig. 3.4165 Comparison of Gas Lift Methods to Show Rates Possible from 8000-ft. with 2.0-in. Tubing

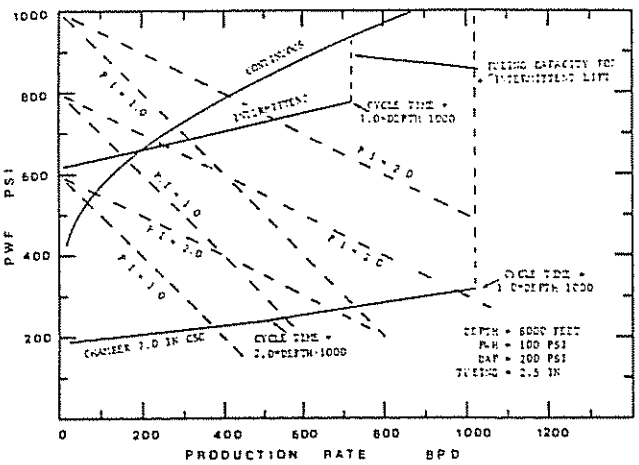


Fig. 3.4166 Comparison of Gas Lift Methods to Show Rates Possible from 8000-ft. with 2.5-in. Tubing

An increase in production is observed with an increase in tubing size, but a continued increase in tubing size eventually reaches a point at which no additional decrease in flowing pressure occurs and, in fact, the tubing becomes too large.

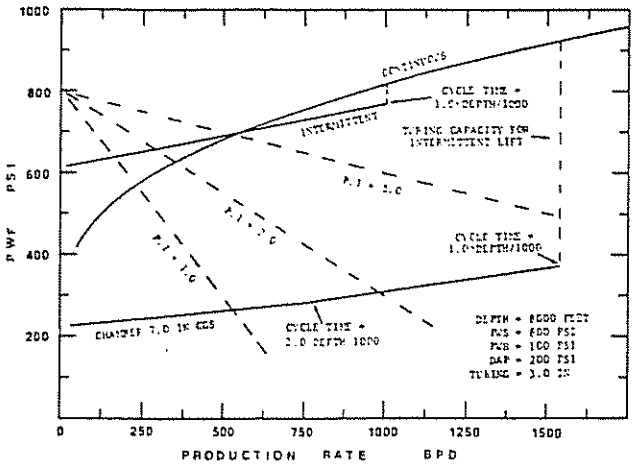


Fig. 3.4167 Comparison of Gas Lift Methods to Show Rates Possible from 8000-ft. with 3.0-in. Tubing

TABLE 3.4117
COMPARISON OF PRODUCTION RATE FOR CHANGES IN TUBING SIZE

	Continuous		Intermittent		Chamber	
Tubing Size (inches)	P_{wf} (psi)	Q (B/D)	P_{wf} (psi)	Q (B/D)	P_{wf} (psi)	Q (B/D)
1¼	800	45	780	240	245	327
1½	800	90	780	324	265	445
2.0	800	205	780	490	280	685
2½	800	440	775	720	315	1020
3.0	800	935	770	1000	370	1535
3½	790	1750	750	1225	380	1750

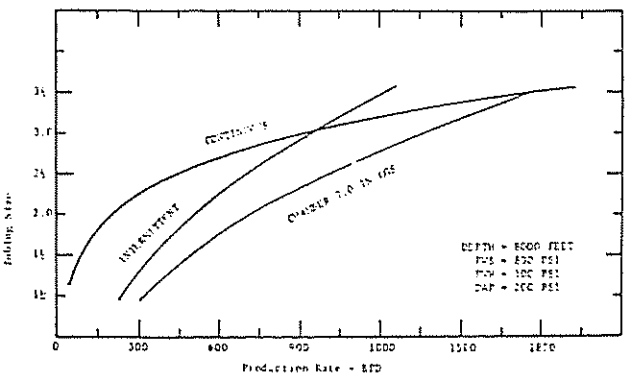


Fig. 3.4168 Comparison of Gas Lift Methods for Changes in Tubing Size

For large capacity tubing, continuous flow gives the highest production rate, as shown in Table 3.4117 for a tubing of 3½".

Up to a tubing size of 3.0", chamber lift gives the highest production rate. Continuous lift gives the highest value of flowing bottom hole pressure and the lowest production rate for the conditions of this example.

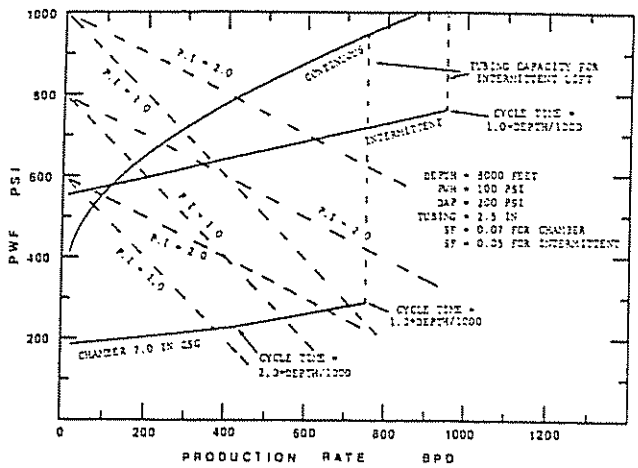


Fig. 3.4169 Comparison of Gas Lift Methods to Show the Effect of Slippage

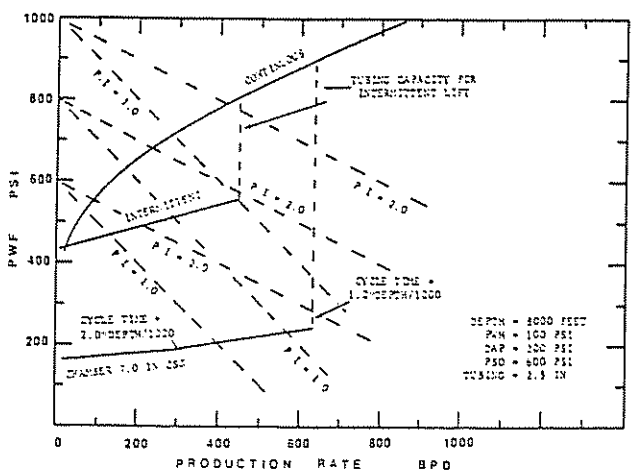


Fig. 3.4170 Comparison of Gas Lift Methods for a Surface Injection Pressure of 600 PSIA

3.446 Effect of the slippage

Slippage or liquid fall-back is that percent of liquid in the initial slug that fails to reach the surface, that is, liquid not produced during a cycle.

Figure 3.4169 shows the change in production of intermittent lift methods due to a change in slippage. Increases in slippage will lead to a decrease in production.

Table 3.4118 summarizes the effect of slippage.

For a value of slippage of 5% per 1000 feet in chamber lift, the maximum capacity is 1025 b/d. When a slippage of 7% per 1000 feet is used, the maximum capacity is reduced to 750 BPD.

For intermittent lift, the change when the slippage is changed from 7% to 5% per 1000 feet is from 720 to 945 B/D.

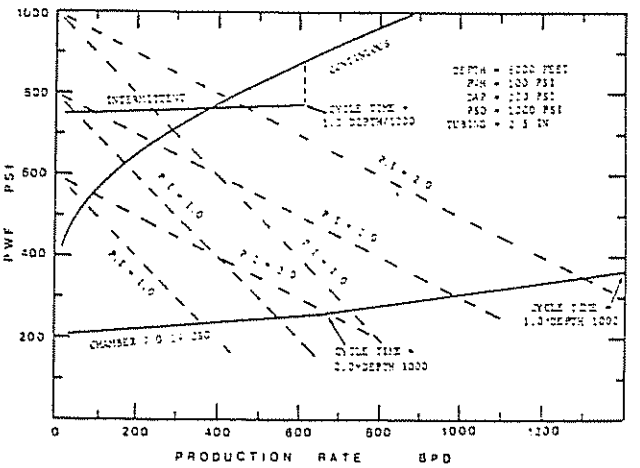


Fig. 3.4171 Comparison of Gas Lift Methods for a Surface Injection Pressure of 1000 PSIA

TABLE 3.4118
COMPARISON OF PRODUCTION BY INTERMITTENT LIFT FOR CHANGES IN SLIPPAGE

The following tables show the effect of slippage changing from 7% to 5% per 1000 ft.

	Chamber (SF = 5%)		Intermittent (SF = 7%)	
PI (B/D/psi)	P_{wf} (psi)	Rate (B/D)	P_{wf} (psi)	Rate (B/D)
1.0	240	560	650	150
2.0	310	980	670	260

	Chamber (SF = 7%)		Intermittent (SF = 5%)	
PI (B/D/psi)	P_{wf} (psi)	Rate (B/D)	P_{wf} (psi)	Rate (B/D)
1.0	255	545		
2.0	*			

*The chamber lift has reached its maximum capacity of 750 B/D before reaching a value of productivity index of 2.0 B/D/psi.

3.447 Effect of surface injection pressure

The effect of surface injection pressure on the total production from a well is shown in Figures 3.4170 and 3.4171. Results are tabulated in Table 3.4119.

An increase in surface injection pressure decreases the maximum production capacity for intermittent lift. For a good continuous flow well, the production rate increases with higher injection pressures.

When the injection pressure was increased the flowing bottom hole pressure for intermittent lift was increased; production was decreased for a constant value of productivity index and constant differential pressure of 200 psi. By requiring a constant differential across the valve, one must wait for a sufficient load to build up in the tubing to give a DAP of 200 psi across the valve. This feed-in of fluids begins to decrease and level off at the higher bottom hole pressures.

For the lower injection pressures the feed-in rate is faster initially because of the lower flowing pressure. If this load is removed, the well will feed in again at a faster rate. Notice the increase in flowing bottom hole

TABLE 3.4119
COMPARISON OF PRODUCTION RATES BY INTERMITTENT LIFT FOR CHANGES IN INJECTION PRESSURE

PI (B/D/psi)	Chamber P_{wf} (psi)	Chamber Rate (B/D)	Intermittent P_{wf} (psi)	Intermittent Rate (B/D)
P_{so} = 600 psi				
1.0	230	570	510	290
2.0	*			
P_{so} = 800 psi				
1.0	240	560	650	150
2.0	310	980	670	260
P_{so} = 1000 psi				
1.0	260	540	758	42
2.0	315	970	760	80

*Maximum capacity was reached before a value of PI = 2.0 BPD/psi.

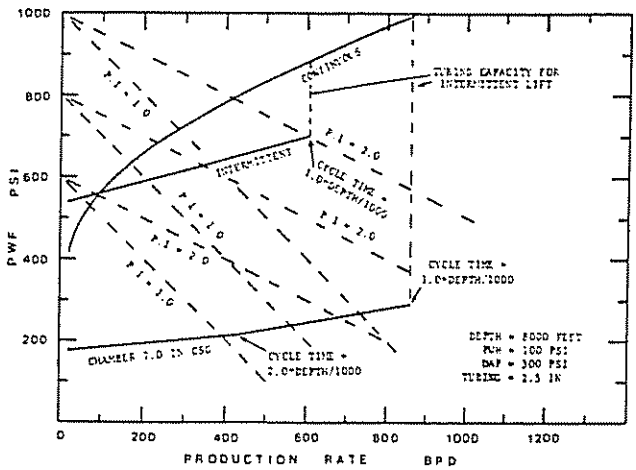


Fig. 3.4173 Comparison of Gas Lift Methods for a Differential Pressure of 300 PSI

pressure for intermittent lift when the injection pressure is changed from 600 to 1000 psi. In field operations it may not be necessary to use all pressure available for some cases.

3.448 Effect of pressure differential across the valve

Differential pressure across the valve will determine the size of the slug to be lifted. (The greater the pressure, the smaller the slug size.) Effect of differential pressure is shown in Figures 3.4172 and 3.4173. Results are tabulated in Table 3.4120. Production rates are increased with increases in differential pressure across the valve.

Note that when the differential pressure increases, the flowing bottom hole pressure decreases and fluids exert less pressure on the formation. As the differential pressure increases, the number of cycles per day is increased, using more gas. In chamber lift the production rate stays almost constant; the only change is the increase in gas volume used.

TABLE 3.4120
COMPARISON OF PRODUCTION BY INTERMITTENT LIFT CHANGES IN DIFFERENTIAL PRESSURE ACROSS VALVE

PI (B/D/psi)	Chamber P_{wf} (psi)	Chamber Rate (B/D)	Intermittent P_{wf} (psi)	Intermittent Rate (B/D)
DAP = 100 psi				
1.0	240	560	705	95
2.0	310	980	715	170
DAP = 200 psi				
1.0	240	560	650	150
2.0	310	980	670	260
DAP = 300 psi				
1.0	240	560	590	210
2.0	*			

*The maximum capacity was reached before a PI = 2.0 BPD/psi.

3.449 Effect of differential and cycle time

In Figures 3.4174 and 3.4175, the productivity index is kept constant and the differential pressure across the

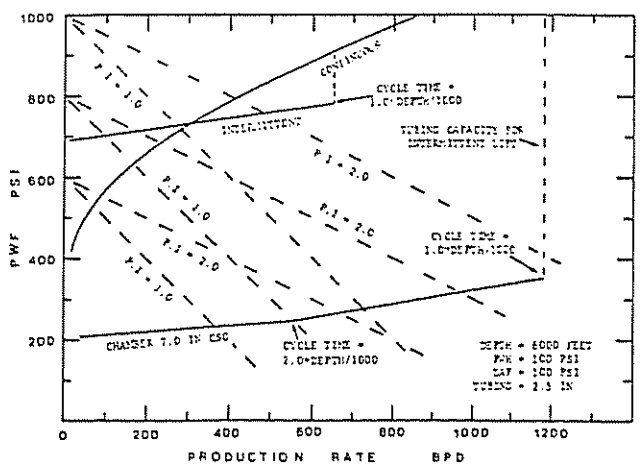


Fig. 3.4172 Comparison of Gas Lift Methods for a Differential Pressure of 100 PSI

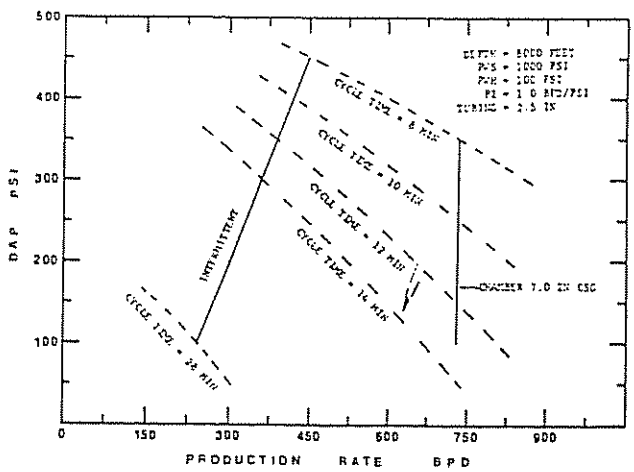


Fig. 3.4174 Comparison of Intermittent Lift Methods. Production Rate vs. DAP

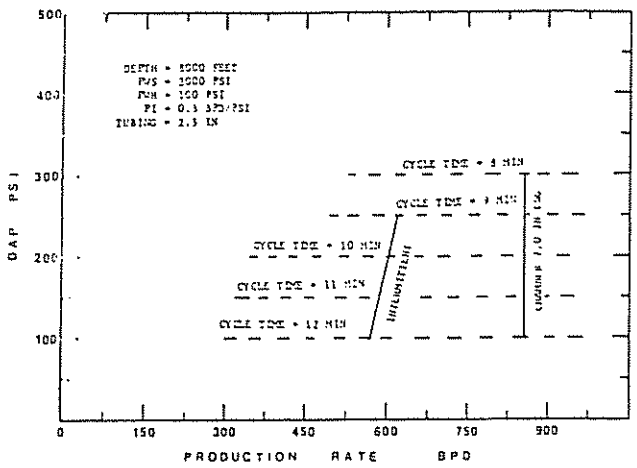


Fig. 3.4175 Comparison of Intermittent Lift Methods. Production Rate vs. DAP

valve is changed as well as the cycle time. Table 3.4121 shows the results of these changes in the total production.

In intermittent lift when the differential pressure is increased, a decrease in cycle time occurs. The load enters more quickly for lower pressures, that is, more production is obtained but more gas is used.

In chamber lift an almost constant production rate is obtained for changes of differential pressures. Because of the low flowing pressure in a chamber and due to the large volume of the chamber, the rate of fluid entry into the chamber is almost constant regardless of the differential pressure in the tubing.

The greater the differential pressure, the shorter the cycle time. This shows that more gas is needed to produce the desired amount of fluids.

TABLE 3.4121
COMPARISON OF INTERMITTENT LIFT FOR CHANGES IN CYCLE TIME

PI = 1.0 P_{ws} = 1000 psi

	Chamber		Intermittent	
DAP* (psi)	Cycle Time (min)	Rate (B/D)	Cycle Time (min)	Rate (B/D)
100	13	730	28	246
150	12	732	24	274
200	11	734	20	302
250	10	735	17	330
300	9	736	14	360
350	8	736	12	390
400			10	420
450			8	450

*DAP = Differential across valve.

3.4410 Summary

Section 3.44 shows a means of comparing intermittent (both standard and chamber) lift and continuous flow. Some decisions as to the best method are obvious. If the well has a high PI and medium to high bottom hole pressure, then continuous flow is the correct choice. For a low bottom hole pressure and high productivity index, a chamber installation is the best choice. For those wells having medium BHP's and low to medium PI's a standard intermittent installation should be considered.

If a question exists as to the best method of lift, then an analysis such as done in Section 3.44 can be made.

3.45 Multiple completions, by Jerry B. Davis and Kermit E. Brown

3.451 Introduction

Davis and Brown[33,34] presented a thorough discussion of dual installations, including types of valves, operation and design. This discussion essentially follows their paper.

The design of single gas lift installations, for both continuous and intermittent lift, has been resolved so it is no longer considered to be a difficult problem. If proper well information is available, an installation can be designed to produce and carry the well to depletion. With the advent of multiple completions, the design of two or more strings of gas lift valves to operate in a common annulus has become a reality.

The design of each zone of a dual installation can be the same as the design of that zone, assuming it is a single completion. Remember that the dual well must make use of the common annulus gas. In some cases a drop in pressure is taken between valves for single installations. This is normally not permitted in the design of dual installations unless one string utilizes fluid operated valves and the other pressure operated valves.

Dual installations are the most common of the multiple completions. The installation of gas lift valves to lift two zones from a common annulus for gas is common practice. However, the operation of these systems has been inefficient in many cases. For some installations this has resulted from faulty equipment, in others, faulty design. The faulty design may have been partly due to incorrect well information and to the procedure in selecting valve settings. The availability of accurate reservoir and well information for both zones is extremely important in the design of a good dual gas lift installation.

3.452 Types of installations

3.4521 Introduction

There are many types of installations that may be installed to lift fluid from dually completed wells. Some of these are dictated by prevailing conditions such as casing size, distance between zones, preinstalled gas lift mandrels; well conditions such as sand, scale and paraffin; and, finally, the operator's preference. The more common types of installations are as follows:

3.4522 Parallel tubing string installations

The most common and highly recommended type of dual installation is parallel tubing strings, noted in Figure 3.322 of Section 3.3. For casing sizes of 7" and larger, ample space is available for running two strings

of 2⅜" tubing or larger. This is true for most types of gas lift valves and mandrels. The clearance should be watched very closely and, for smaller casing sizes, an obvious limitation is the maximum production possible through the two tubing sizes selected. For example, in 5½" casing two strings of $2^{1}/_{16}$" O.D. tubing could be selected for the production requirements. This tubing size could curtail the producing rate in some instances when compared to the aforementioned 2⅜" tubing capacity.

3.4523 Concentric tubing string installations

A typical concentric string dual is shown in Figure 3.319 of Section 3.3. In the early stages of dual gas lift, this type of installation was fairly common. However, the advantages of parallel tubing strings have practically obsoleted running concentric strings in dual installations.

There may still be occasions where a macaroni string is run inside a tubing string to lift the lower zone of a dual (Fig. 3.320 of Section 3.3). This may be done when it is necessary to lift the bottom zone only with the upper zone still flowing out the annular space between the tubing and casing. In reality, this is not a dual lift problem since only one zone has been placed on gas lift.

3.4524 Commingling of zones

Two zones may be produced up one tubing string by commingling the production from the two zones at bottom. The immediate advantage of this type installation is the elimination of one string of tubing and gas lift valves. Production from both zones is produced up one string of tubing; therefore, design requirements for a dual installation are eliminated and will not be discussed in this section. The exact production from each zone becomes a measurement problem in this case; therefore, a limited number have been installed. The comingling of a gas zone with an oil zone may offer sufficient energy to flow both zones.

3.453 Valve selection for a dual installation producing both zones by continuous lift

3.4531 Introduction

To design a dual installation that will produce efficiently, correct productivity information must be known concerning each zone. It must be ascertained that each zone can be lifted from a depth that allows good continuous lift operations. The design will depend upon the use of the accurate multiphase flow correlations.

A number of possible valve selections are available for dual continuous lift installations. This selection should be made with extreme care to insure valve compatibility between strings. Since a common annulus is being used, use either valves that operate independently of the annulus operating injection pressure or valves that operate at the same injection pressure. The use of fluid operated valves or the use of valves such as the combination valve that opens on fluid load in the tubing and closes on casing pressure or tubing pressure drop appear to be suitable.

However, good installations have been made with pressure operated valves that are highly sensitive to tubing pressure. Each valve in both strings must of course operate at the same pressure. The combination of valves will be discussed in the following sections.

3.4532 Two strings of combination fluid opened and pressure or fluid closed valves

These valves open on a fluid load in the tubing and close on casing/tubing differential pressure or on a pressure drop in the casing. This valve incorporates a section that allows the lower portion of the valve to be operated by a difference in pressure between the casing and tubing. The upper portion of the valve can be a standard bellows valve adapted for continuous flow. Although slight modifications may exist, this type of valve is commonly available. A schematics illustrating this type valve is shown in Figure 3.228 (refer to Section 3.2 on valves). This represents a type of gas lift valve that will suit any type of continuous flow installation. In particular, for wells where a lack of detailed information is available concerning the depth of the point of gas injection, this valve will perform satisfactorily to the maximum continuous flow operating depth.

3.4533 Two strings of fluid operated valves (open and close on tubing fluid pressure)

This valve opens due to a fluid build-up in the tubing string and is almost completely independent of casing pressure. It also is sensitive to tubing pressure when closing. Again, this represents a valve that can be installed without knowing where the exact point of gas injection will be. This valve is normally equipped with a small port and may be choked further for continuous lift operations. It offers the advantage of operating independent of casing pressure. The valves can be set to open on increasing tubing loads progressively down the well. They should use accurate choke or seat sizes for correct gas passage and then they are easily controlled with full casing pressure at the surface.

3.4534 Two strings of pressure operated valves

It is not uncommon to run two strings of pressure operated valves (controlled primarily by casing annulus pressure, yet highly sensitive to tubing pressure) in controlling the gas passage rate. Numerous valve types are available including tapered stem, choke upstream from main seat, and variable orifice principle. However, one type that should not be run is the choking of a standard bellows valve downstream from a large port. This type of valve incorporates too much "spread." For tapered stems, a small port (⅛"-¼") is used. These valves are best suited to be set on a constant surface opening pressure. Both strings should function without interference.

3.4535 One string of fluid operated valves (open and close on tubing fluid pressure) and one string of pressure operated valves

The design of this installation is such that the fluid operated string operates independently of the injection

gas pressure. The string of pressure valves is designed for the usual 10-25 psi drop in surface operating pressures per valve, or they can all be set for the same surface operating pressure. The fluid operated valves can operate successfully over a limited range of casing pressures. This offers a good valve combination.

3.454 Valve selection for a dual installation producing both zones by intermittent lift

3.4541 Introduction

The design of installations wherein both zones are to go on intermittent lift is difficult. The problem of surface gas supply and valve characteristics can be somewhat troublesome under certain circumstances. The one serious difficulty arises when both strings require gas at the same time.

For example, if each string is designed such that a 100 psi drop in casing pressure is required to lift either zone and both strings require gas simultaneously, insufficient gas may be available for both strings. In this case they both may load up. In installations where gas control is by a surface injection choke and the valves are opened by a fluid buildup in the tubing, there is no present method for controlling the time at which each zone will require gas. This is an inherent problem with intermittent duals for which no feasible solution has been offered.

3.4542 Two strings of combination fluid opened and pressure closed valves

This type valve may be a standard single-ported bellows valve with a fluid pilot attachment or a standard bellows pilot control valve with fluid pilot attachment. One advantage to these valves is spread does not have to be directly built into the valve. Since the lower fluid pilot section requires a fluid load in the tubing string to open it, the upper portion may be open and cocked prior to the opening of the lower pilot section to admit gas.

For example, assume that a dual installation has both strings lifting with 900 psi operating pressure at depth from 8000 feet, requiring a 100 psi drop in casing pressure to supply the required amount of gas to each string. A typical setting would be to set the pressure section to close at 800 psi at depth and open at 850 psi. The fluid-operated pilot section of the valve may incorporate a 300 psi spring pressure, which means that it opens when the differential between tubing and casing reaches 300 psi.

The valve will then operate as follows: casing pressure will increase through a surface choke until a pressure of 850 psi is reached, causing the pressure-operated section of the valve to open. The fluid operated section will remain closed until a tubing pressure of 850−300=550 psi is reached. If there is no additional buildup of casing pressure, both sections of the valve will open at that time. The casing pressure will then drop to the closing pressure of the pressure section, which is 800 psi. However, since a 100 psi spread in casing pressure is required, the casing pressure must be increased to 900 psi before the fluid-operated pilot section opens.

As before, the pressure operated section will have opened at 850 psi and will remain in a cocked position until the lower fluid pilot section opens. Gas will be continually injected at the surface until the casing pressure reaches 900 psi. At this time a regulator will stop gas entry at the surface. The fluid pilot section of the valve will then remain closed until the fluid load in the tubing reaches 900−300=600 psi. The fluid section will open and gas will enter the tubing until the casing pressure drops to 800 psi.

The versatility of this type valve for a dual installation is immediately evident from the previous example. The amount of gas required for lift can be controlled from the surface buildup of casing pressure. For example, if a 130 psi drop in casing pressure is required, the casing pressure would be increased to 930 psi. The fluid pilot section of the valve would then open at 930−300=630 psi and the pressure section of the valve would still close at 800 psi.

This valve also minimizes the possibility of making an original design incorporating too much spread. When doubtful about the needed spread, incorporate a low value and inject the additional gas by an overbuild on the casing.

Both strings of these valves should be set with constant surface closing pressures.

The possibility of both strings requiring gas simultaneously still exists. A fairly large injection choke at the surface with a regulator to shut off on a high pressure will help to alleviate this problem. In this instance, gas is injected at a fairly high rate yet not so quickly that the operating valve cannot go closed.

3.4543 Two strings of pilot operated valves

The pilot operated valve has been used successfully on dual installations. This valve controls spread independent of the main port size. Its distinct advantage over a standard bellows valve is that a large main seat size can be maintained. However, this valve is not as flexible for controlling spread as is the combination fluid-opened, pressure-closed valve.

For example, the control port on a pilot valve may be set to give 100 psi spread in a valve opening at 900 psi and closing at 800 psi while lifting a 600 psi tubing load. However, the installation design for a dual operatin must necessarily be for a particular set of conditions. The casing pressure overbuild feature, as was explained for the combination fluid-opened, pressure-closed valve, cannot be utilized. However, some flexibility is available.

For example, the valve will open at 860 psi for a 675 psi load and will open at 940 psi for a 500 psi tubing load. A very important feature of the pilot valve is that some flexibility exists in the amount of load to be lifted and the volume of gas used to lift it. For example, if too much gas is being used per cycle, the casing pressure shut-off point can be reduced and a heavier tubing load will trip the valve. If more gas is needed, the casing pressure shut-off point can be raised and a lighter tubing load is then required to open the valve. Although this valve is not as flexible as the pressure-closed, fluid-opened valve, it can be utilized for intermittent dual lift.

3.4544 Two strings of standard pressure operated bellows valves

Although not normally recommended, there have been installations run where two strings of standard bellows valves were used. If both strings will lift through valves at the same depth from the bottom of the well and the two zones have somewhat similar characteristics, this type of equipment might be considered. In this instance the injection gas could be controlled through an intermitter at the surface, thus eliminating the problem of correct spread in the valve.

Once the valve and bellows size is set, the spread on these valves is controlled by the seat size. The limitations of these valves are therefore evident. The casing and tubing sizes may be such that a small spread between opening and closing pressure is needed, necessitating a small port size. Yet a large port size is necessary for good intermittent lift operations.

If a large spread on the valve is required, problems arise when a lighter tubing load than originally anticipated must be lifted. For example, assume that originally a valve is needed to open at 900 psi and close at 800 psi with a 600 psi tubing load. This will require a $^{9}/_{16}$" diameter port for a 1½" O.D. valve. Should it become necessary to lift a 300 psi load, an opening pressure of 1040 psi would be required. This requirement may make it impossible to open the valve.

3.4545 Two strings of fluid operated valves

If two strings of fluid-operated valves are to be used, they should be designed for multipoint injection. This will permit a constant casing pressure and an automatic operation for the opening of the fluid valves. Careful design is necessary for proper multipoint injection.

3.455 Valve selection for a dual installation producing one zone by continuous lift and one zone by intermittent lift

3.4551 Two strings of combination fluid opened and pressure closed valves

Similar to having both strings on continuous or intermittent lift, this type of valve is also good for one zone continuous and one zone intermittent lift. The valves in the continuous lift depth range should use small ports, whereas the valves for the intermittent depth range should use larger port sizes. This valve combination would be particularly suitable where very little well information is available; it should perform satisfactorily from any valve operating depth on either string.

There are two possible settings for this combination of valves. For example, assume the continuous lift string is set to operate at a surface pressure of 700 psi and lift from a valve at 4000 feet. This is set by the surface closing pressures on the valves and essentially represents the surface opening pressure on these valves since the seat sizes may be small.

There are two alternatives in setting the valves in the intermittent depth range. One alternative is to set the surface closing pressures at 700 psi or the same as the surface opening on the continuous lift side. In this particular case assume that an opening pressure of 775 psi is required to supply the needed gas. Gas for the intermittent string would be furnished by an intermitter from the surface which would supply gas to raise the casing pressure to 775 psi during each cycle (approximately 30 min.).

Assume that lifting from 8000 feet on this string is required. The surface arrangement would consist of a choke and an intermitter installed in parallel so the choke would continuously supply gas to the continuous lift side. The intermitter would open every 30 minutes and build the casing pressure to 775 psi which, in turn, would open the intermittent valve. With the higher differential pressure across each valve, there is no danger in opening upper continuous lift valves, since the fluid pilots have them locked out.

In fact, this represents a possibility of closing the operating valve on the continuous lift side due to the increased differential. With a lower continuous lift valve the operation on the continuous lift side would conceivably move down one valve at this time. With the casing pressure at a maximum of 775 psi, the intermitter would close, the intermittent valve would open and would immediately start lowering the casing pressure again to 700 psi. This would occur in approximately 8 minutes (depth of lift/1000=8000/1000=8 min.) where slug velocity is 1000 ft/min. The intermittent valve would then close. The continuous flow side should have no interruption in flow during this period.

The second alternative in this case is to set the surface closing pressure on the intermittent valve to be 625 psi. The normal casing pressure supplied by the choke to keep the continuous lift side operating properly would be 700 psi. This is also the opening pressure of the intermittent operating valve, but it will not open until a tubing load of approximately 500 psi has accumulated opposite the valve. Consequently, this installation must depend upon the feed-in characteristics of the well. However, it would mean that gas would be shut off from the continuous lift side for a period of about 8 minutes (until the intermittent valve closes) and for an additional few minutes to allow casing pressure to build back up to 700 psi. This, in itself, could be a serious objection to the latter valve setting.

An advantage to the system, however, is that no high pressure gas volume storage is needed. The type of valve settings would depend, to some extent, upon the required fluid production from the continuous flow side. Producing of the continuous lift string can be assured by setting the intermittent string to open higher. This type of setting is possible only with a combination fluid-opened, pressure-closed valve and should not be attempted with standard continuous flow valves unless they are choked drastically.

3.4552 One string pressure operated bellows continuous lift valves and one string of pilot operated intermittent valves

For this combination a string of bellows continuous flow valves can be run on the continuous lift side. These are all set for the same surface opening pressure.

A string of pilot-operated valves (all set for the same surface closing pressure) can be run on the intermittent string. Since the pilot valve is opened to a great extent

by fluid pressure buildup, it is not essential to have an intermitter on the surface. For example, should all continuous lift valves be set for a surface opening pressure of 700 psi and 75 psi casing pressure buildup or drop be needed to supply the necessary gas for the intermittent side, the closing pressures on the intermittent string can all be set for 625 psi and the opening pressures at 700 psi (which is the same as required for the continuous lift string).

When the desired fluid builds up on the intermittent valve, the valve opens and drops the casing pressure to 625 psi. This will lift the slug out but also closes the continuous lift valves until the casing pressure is built back up to 700 psi. Although this causes a somewhat erratic action by the continuous lift side, the type of valves being used practically prohibit having to build the casing above 700 psi to open an intermittent valve (such as setting it to open at 775 psi). Since these types of continuous lift valves do not lock out of service, several of them will open when the casing pressure is built above 700 psi. If fairly large chokes are installed, the needed 775 psi to open the intermittent valve may not be reached. Therefore, the operating pressure of the intermitting string in this type of installation must be set lower than the operating pressure of the continuous lift string.

3.4553 Two strings of pressure operated bellows valves

In this instance the string of pressure-operated valves on the continuous lift side would be set for the same surface opening pressures. Assume, for example, that the continuous lift string will require a surface operating pressure of 700 psi to retain good operations. The intermittent string should utilize large-ported valves set for a surface opening pressure of 700 psi and a surface closing pressure necessary to supply sufficient gas to lift the slug of liquid to the surface.

Assuming that 50 psi drop in casing pressure will supply the needed gas, the surface closing pressure on this string would then be set for 700−50=650 psi. Although the continuous lift string would be interrupted while the casing pressure dropped to 650 psi, it would immediately resume normal operations when the casing pressure again reached 700 psi. A regulator by-pass control can help in speeding this build-up.

Setting the surface closing pressure on the intermittent string at 700 psi and obtaining 50 psi casing pressure buildup to 750 psi by an intermitter can be considered. However, the upper continuous lift valves must be drastically choked so the opening of several of them will not prevent the build-up of casing pressure. This latter setting allows continuous operation on the continuous lift string.

3.455 One string of fluid operated valves (opened and closed by tubing fluid pressure) and one string of pressure operated bellows valves

In this instance the fluid operated valves will operate independently of the annulus casing pressure. They could be placed on either the continuous flow or intermittent flow string. If used on the intermittent string, they should be used in a multi-pointing manner. Hence, they would use small seat sizes so as not to drastically reduce the injection gas pressure. For example with 700 psi maintained on the annulus for the continuous flow pressure valves (all set at the same pressure), the fluid valves in the operating range could open with 500-550 psi tubing loads. By using a regulator in conjunction with a choke (parallel) additional gas could be supplied when the fluid valves opened to maintain 700 psi on the casing, hence keeping the continuous flow side in continuous operation.

If the pressure valves are used on the intermittent side an intermitter at the surface could be used to inject more gas periodically to lift the intermittent side and maintain 700 psi at all times on the casing for the continuous flow zone. An overbuild to open the pressure valves could be tolerated.

3.4555 Two strings of fluid-operated valves (opened and closed by tubing fluid pressure)

Two strings of fluid valves could be utilized. For example, the continuous flow string could be set to operate on a casing pressure of 700 psi and the intermittent side set to open on tubing pressure of from 500 to 550 psi in a multi-pointing manner. A regulator set in parallel with a choke at the surface would sense the need for additional gas when the intermittent valves open and maintain 700 psi on the casing. The intermittent valves would close on slug removal and the other side would flow continuously.

3.456 Design of dual gas lift installations

The design of each separate zone of a dual is handled in the same manner as if the well were to be a single installation. However, consideration must be given to the selection of compatible valves and the surface injection gas pressure. The dual installation may not be able to operate at the same high pressure as the corresponding single well, but in most cases very little loss of injection pressure is necessary. Therefore, the actual design procedures are omitted, but several example cases are given.

3.457 Example designs

3.4571 Example problem #1

The problem examples of Section 3.42921 (continuous flow) and Section 3.4352 (intermittent flow) can be placed together in a dual installation. The data for each well is given again in this section.

Zone 1—continuous flow zone
Given data:
Tubing size = 2⅞" O.D.
$\gamma_g = 0.70$
Well loaded with kill fluid of gradient = 0.40 psi/ft
GOR = 400 scf/B
Well produces 50% water
GLR = 200 scf/B
$\overline{P}_R$ = 1920 psi
Bubble point pressure = 1500 psi
°API = 35
PI = 5 = linear until reaching P_b
P_{wh} = 120 psig (constant)

P_{so} = 900 psi (950 available)
P_{ko} = 1000 (1050 available)
Depth = 8000 ft
Desired rate = maximum
Temp bottom = 170°F.
Geothermal Temperature Gradient = 1.2°F/1000 ft (109°F surface)

Zone II—intermittent flow
Given data:
2⅜" O.D. tubing (1.995" I.D.)
7" casing
P_{so} = 1000 psi (available pressure = 1050 psi)
G_s = 0.40 psi/ft = kill fluid gradient
Depth = 7600 ft
γ_g = 0.7
P_{wh} = 80 psi
P_R = 1500 psi
Temp bottom = 167°F.
Temp surface = 109°F.
°API = 35
Average PI = 0.2 Based on 50% drawdown
One well test shows 150 B/D at P_{wf} = 750 psi
Bubble point pressure greater than 1500 psi

Zone I produces 1000 B/D of which 500 B/D is oil. Zone II produces a maximum of 200 B/D of oil. Therefore the design should be such that Zone I produces without interruption.

Refer to Sections 3.42921 and 3.4352 for detailed design procedures for each zone. To illustrate the various valve settings, Tables 3.464, 67, 68, 91 and 96 have been reproduced in this section for convenience.

Nomenclature:
POV = Pressure operated valve
FOV = Fluid operated valve
CFOPC = Combination fluid opened pressure closed valve

From Table 3.4122, the various combinations of valves can be selected. Note that all continuous flow valves of Zone I are set to open with 900 psi on the casing at the surface, although the fluid operated valves would have some flexibility and would operate on either higher or lower casing pressures depending upon the tubing pressure.

TABLE 3.4122

No.	Table	Zone	Valves	P_{so} (psi)	P_{sc} (psi)	Remarks
1	3.464	I	POV	900		Continuous flow
2	3.467	I	FOV	900		Continuous flow
3	3.468	I	CFOPC	900		Continuous flow
4	3.491	II	POV		900	Intermittent flow
5	3.496	II	FOV		900	Intermittent flow
6	3.499	II	CFOPC		900	Intermittent flow

TABLE 3.464
VALVE INFORMATION FOR PRESSURE OPERATED VALUES
(Zone I—Continuous Flow)

Valve no.	Port size	Depth	Temp	P_{so}	P_{vo}	P_t = Design tubing pressure	P_{vc} = P_o at depth	P_{sc} $P_{t\ (Design)}$	P_d 60°F	P_{tro} 60°F	Final calib.
1	5/16	2350	127	900	955	570	917	862	800	886	885
2	5/16	3350	135	900	979	676	949	876	820	908	910
3	5/16	4150	141	900	998	762	975	881	830	919	920
4	5/16	4750	145	900	1012	825	993	884	840	930	930
5	5/16	5250	149	900	1024	879	1010	888	845	935	935
6	5/16	5650	152	900	1033	921	1022	890	855	946	945
7	5/16	5800	153	900	1037	937	1027	892	860	952	950
8	5/16	5900	154	900	1039	948	1027	892	860	952	950

R = .0996, 1-R = .9004, T.E.F. = .1106

TABLE 3.467
VALVE INFORMATION FOR FLUID OPERATED VALVES
(Zone I—Continuous Flow)

Valve no.	Depth	Temp	P_{so}	Tubing opening $P_{t\ design}$	$P_{vo\ casing}$	Port	(Tubing) P_d = P_{vc}	P_d 60°F	P_{tro} = P_t at 60°F	Adjusted P_{tro} (tubing press)
1	2350	127	900	570	955	1/4	595	520	555	555
2	3350	135	900	676	979	1/4	695	600	641	640
3	4150	141	900	762	998	1/4	777	655	700	700
4	4750	145	900	825	1012	1/4	837	707	755	755
5	5250	149	900	879	1024	1/4	888	745	796	795
6	5650	152	900	921	1033	9/32	930	780	849	850
7	5800	153	900	937	1037	9/32	945	790	859	860
8	5900	154	900	948	1039	9/32	955	800	870	870

TABLE 3.468
VALVE INFORMATION FOR COMBINATION FLUID OPENED-PRESSURE CLOSED VALVES
(Zone I—Continuous Flow)

Valve no.	Port size	Depth	Temp	P_{so}	P_{vo}	Design tubing pressure	P_{vc} = P_d at Depth	P_{sc} $P_{t\ (Design)}$	P_d 60°F	P_{tro} 60°F	P_{tro} adjusted
1	5/16	2350	127	900	955	570	920	865	800	886	885
2	5/16	3350	135	900	979	676	952	879	820	908	910
3	5/16	4150	141	900	998	762	978	884	830	919	920
4	5/16	4750	145	900	1012	825	996	887	840	930	930
5	5/16	5250	149	900	1024	879	1013	891	845	935	935
6	5/16	5650	152	900	1033	921	1025	893	855	946	945
7	5/16	5800	153	900	1037	937	1030	895	860	952	950
8	5/16	5900	154	900	1039	948	1033	895	860	952	950

TABLE 3.491
SUMMARY TABULATION—MULTI-POINT OR SINGLE POINT INJECTION PRESSURE OPERATED VALVES—CHOKE CONTROL
(Zone II—Intermittent Flow)

Valve	Depth	Temp	P_{sc}	P_t Unloading	Max. tubing press P_t	Press casing (max)	P_{vo} slug pass	P_{so}	P_d = P_{vc}	Calculated R	A_p	Control port diam. (nearest)	True R values	P_d 60°F	P_{tro}	P_{tro} adjusted
1	2350	128	900	230	670	1025	978	922	955	.088*	.06776	5/16	.0996	830	922	920
2	4275	142	900	354	785	1074	1045	937	1001	.1014*	.0804	5/16	.0996	850	944	945
3	6000	155	900	---	887	1117	1104	955	1042	.286	.22	½	.2562	860	1156	1155
4	6420	158	900	---	912	1128	1118	958	1052	.32	.246	9/16	.3227	860	1270	1270
5	6800	161	900	---	934	1138	1130	960	1060	.357	.275	9/16	.3227	868	1282	1280
6	7120	164	900	---	954	1146	1142	963	1068	.394	.303	5/8	.3990	870	1448	1450
7	7450	166	900	---	973	1154	1153	966	1076	.425	.327	5/8	.3990	870	1448	1450
8	7600	167	900	750 (to open)	982	1158	1158	966	1080	----	---	7/16	.1952	875	1087	1085

* Based on Maximum Casing Pressure of 1025 and 1074, respectively
All other valves set on P_{vo} (slug pass)

$$R = \frac{P_{vo} - P_{vc}}{P_{vo} - P_t}$$

TABLE 3.496—SUMMARY TABULATION
MULTI-POINT INJECTION—FLUID OPERATED VALVES (Zone II—Intermittent Flow)

Valve no.	Depth	Temp of	P_t Max	P_{casing} surface	P_{cso} valve	P_t unload	Port size	P_{to} (tubing)	P_d = P_{vc}	P_d 60°F	P_{TRO}	Final calib	
1*	2350	128	631	980	1040	230	¼	230	988	860	918	920	press oper. valve
2	4250	141	710	900	1000	--	¼	710	728	625	667	665	fluid valve
3	5000	147	742	900	1018	--	¼	742	760	640	684	685	fluid valve
4	5700	153	771	900	1035	--	¼	771	788	660	705	705	fluid valve
5	6380	158	800	900	1051	--	¼	800	816	675	721	720	fluid valve
6	7000	162	825	900	1065	--	¼	825	840	690	737	735	fluid valve
7	7600	167	850	900	1080	750 (to open)	¼	750	771	627	670	670	fluid valve

$$P_{to} = \frac{P_d}{1 - R} - P_{casing}\frac{R}{1 - R}$$

$$P_d = P_{vc} = P_{to}(1 - R) + P_{casing}(R)$$

*#1 valve only is a pressure operated valve to permit wider spacing.

P_{to} Tubing press to open
For a ¼" × 1½" valve
R = 0.0638
1 − R = 0.9362

TABLE 3.499
SUMMARY TABULATION COMBINATION PRESSURE CLOSED—FLUID OPENED VALVE
CHOKE CONTROL OR TIME CYCLE CONTROL (Zone II Intermittent Flow)

Valve no.	Depth	Temp	P_{sc}	$P_d = P_{vc}$	P_{vo}*	P_{so}	P_{casing} Max (at Valve)	P_{surf} casing max	P_{tubing} max	P_{tubing} start	Spring diff*	P_d 60°F	P_{trc}	Control port	Main port
1	2350	127	900	955	986	930	1015	956	651	—	365	830	922	5/16	5/16
2	3330	134	900	979	1011	930	1041	957	701	—	340	850	944	5/16	5/16
3	4230	141	900	1000	1033	930	1065	959	747	—	320	848	942	5/16	5/16
4	5100	148	900	1020	1053	930	1087	959	792	—	295	852	946	5/16	5/16
5	5900	154	900	1039	1073	930	1107	959	833	—	275	862	957	5/16	5/16
6	6600	159	900	1056	1090	930	1125	959	869	—	255	864	960	5/16	½
7	7280	165	900	1072	1107	930	1141	959	904	—	235	870	966	5/16	½
8	7600	167	900	1080	1115	930	1150	960	920	750	400	875	972	5/16	½

*P_{vo} set at P_{vc} + ½ spread
**Spring values rounded off to nearest (5)

5/16" Control Port
R = 0.0996
1 − R = 0.9004

All of the intermittent valves of Zone II are set to close on 900 psi casing pressure at the surface, that is, except the fluid valves which have some flexibility. Therefore, in each case the casing pressure would need to be increased to furnish gas to the intermittent side. The following combinations of valves and settings would work satisfactorily and are shown by number combinations from Table 3.4122:

(A) 2 and 4, 5, or 6
(B) 3 and 4, 5, or 6
(C) 1 and 5
(D) 4 and 2
(E) 5 and 1, 2 or 3
(F) 6 and 2

As noted, several combinations are possible and careful attention to valve spacings and settings should be adhered to. Some manipulations of valve settings and casing pressures would improve operations as compared to those shown. For example, for 2 and 6, up to 1000 psi could be carried on the casing and the continuous flow zone would maintain operation. When the fluid load opened the operating valve of the intermittent zone, it would drop the casing pressure to 900 psi with Zone I continuing to flow.

3.4572 Example problem #2

The following example problems were worked by Davis and Brown.[33] Refer to Table 3.4123 for given data on three wells having two zones each. These show all three possibilities for duals; that is, a well with both zones continuous flow (Well #1, Zones I and II), a well with both zones intermittent lift (Well #2, Zones A and B), and a well having one zone on continuous flow and one zone on intermittent lift (Well #3, Zone X on continuous lift and Zone Y on intermittent lift). Tables 3.4124, 3.4125, and 3.4126 show all these results.

3.4573 Example problem #3 (mandrels in place)

Davis and Brown gave another field case whereby side-pocket mandrels were already in place.[33] Many dual wells exist wherein the mandrels for retrievable gas lift valves were installed in both tubing strings when the well was originally completed. In many cases, these mandrels were spaced for running pressure-operated gas lift valves, leaving a situation that makes spacing for fluid-operated valves somewhat questionable. If fluid valves are installed, careful manipulation may be required for unloading purposes.

The following information concerning dual Well #4 represents an actual field example that required the installation of valves to lift Zone M. Zone N was expected to require artificial lift within a short period. The following information was available:

Depth of Side Pocket Mandrels	
Zone M	Zone N
1548	1488
2593	2533
3450	3390
4186	4126
4889	4829
5524	5464
6160	6100

Gradient saltwater = 0.457 psi/ft
Casing size = 7"
Tubing sizes = 2 strings of 2⅜" O.D.
Lift pressure available = 800 psi
Separator pressure = 40 psi
Estimated surface tubing pressure = 50 psi
Temperature = 1.2°F. per 100 ft + 74°F.
Gas gravity = 0.65
°API of oil = 45° (.348 psi/ft)
q_L = 100 B/D
q_w = 50 B/D
q_o = 50 B/D

1" O.D. equipment will be required for use in the side pocket mandrels. The lift volume of 100 BFPD results in the recommended use of the intermittent lift type operation for each zone. The mandrel depths have been predetermined with spacings of 1045, 857, 736, 693, 635, and 636 feet and must be used.

Find the maximum spacing that can be used for the

TABLE 3.4123
DUAL GAS LIFT INSTALLATION DESIGN GIVEN DESIGN INFORMATION, THREE SEPARATE DUAL WELLS

Reservoir and installation data	Well no. 1		Well no. 2		Well no. 3	
	Zone I	Zone II	Zone A	Zone B	Zone X	Zone Y
Static bottom hole pressure, SBHP (psi)	1200	2100	1200	2100	2250	2100
Productivity Index, P.I. (B/D/psi)	1.00	0.50	0.05	0.05	1.00	0.05
Production required (BFPD)	400	400	Max.	Max.	400	Max.
Calculated FBHP (psi)	800	1300	Min.	Min.	1850	Min.
Static fluid level (ft)	2500	1700	2500	1700	2000	1700
Perforation depth (ft)	5130-70	6240-60	5130-70	6240-60	6735-65	6240-60
Static fluid gradient (G, psi/ft)	0.457	0.457	0.457	0.457	0.457	0.457
Percent water (%)	50	50	50	50	50	50
Oil produced required (BOPD)	200	200	Max.	Max.	200	Max.
Formation GOR scf/B	300	400	300	400	300	400
Formation GLR scf/B	150	200	150	200	150	200
BHT (°F)	136	N.A.	136	N.A.	155	N.A.
Static surface Temperature (°F)	80	80	80	80	80	80
Flowing surface Temperature (°F)	110	110	80	80	110	110
K.O. gas available Pko (psi)	1000	1000	1000	1000	1000	1000
Max. operating gas pressure (psi)	800	800	800	800	800	800
Gas lift gravity	0.65	0.65	0.65	0.65	0.65	0.65
Flowing wellhead pressure, Pwh (psi)	100	100	50	50	100	50
Tubing size (in.)	2⅜	2⅜	2⅜	2⅜	2⅜	2⅜
Casing size (in.)	7	7	7	7	7	7

Discussion

Well #1 is to be equipped with 1" O.D. wireline retrievable gas lift valves installed in sidepocket mandrels. Both zones will produce by the constant lift method.

Well #2 is to be equipped with 1½" O.D. tubing retrievable gas lift valves. Both zones will produce by the intermittent lift method.

Well #3 is to be equipped with 1½" O.D. tubing retrievable gas lift valves. One zone will produce by the continuous lift and one zone by the intermittent lift method.

combination fluid opened-pressure closed valve by dividing the spring pressure of the valve by the gradient between it and the next lowermost valve. The maximum recommended spring pressure is 450 psi or 40-45% of the operating pressure, whichever is less.

In this example, the maximum recommended spring pressure becomes approximately 350 psi with a maximum spacing of 350/0.457 = 766 feet. (If ample operating pressure were available, the maximum spacing recommended would be 450/0.457 = 985 feet for unloading the saltwater gradient.) Therefore, if the 350 psi spring pressure valve were to be installed in the top two mandrels, unloading problems would occur should the well filled to these points with saltwater. Should the installation be filled with 45° API oil (0.348 psi/ft), then this maximum spacing becomes 350/0.348 = 1006 feet. If liquid with a gradient greater than 350/1045 = 0.335 psi/ft were to be unloaded from above the valve depth of 2593 feet or a liquid gradient of 350/857 = 0.41 psi/ft from above the valve depth of 3350 feet, then either standard pressure or fluid valves would be used. Pressure valves, set to operate at least 50 psi or more above all other valve operating pressures, may be used in these two mandrels. Interference may occur between these valves in some instances. Fluid valves can be used in the upper two mandrels with the combination valves installed in the lower four mandrels. Should the installation have been unloaded to the point that only the 45° oil is above the top two valves, then the installation would operate satisfactorily with the 350 psi springs installed in the combination valves above these depths. Spring pressures for both strings of valves are set alike, neglecting the 60 ft depth difference. Note summary tabulation in Table 3.4127.

TABLE 3.4124
DESIGN RESULTS—WELL #1, ZONES I AND II, DUAL GAS LIFT INSTALLATION, BOTH ZONES CONTINUOUS LIFT, ALL VALVES SET AT 800 PSI SURFACE OPENING PRESSURE

Zone II

Valve depth (ft)	Csg. open press. @ depth (psi)	Tbg. design open depth press. (psi)	Spring press. (psi) *	Close dome press. depth (psi)	Depth temp. (°F)	80°F test rack press. (psi) Dome	Set **	Orifice size (in.)
2100	840	495	350	811	121	744	815	6/64
2900	855	580	275	831	125	758	830	6/64
3525	867	650	225	848	128	770	845	6/64
4025	876	710	175	861	131	779	885	7/64
4425	884	750	125	873	133	786	860	7/64
4750	890	785	100	881	134	791	865	8/64
5000	895	825	100	889	135	796	870	8/64

* Set to nearest 25 psi
**Set to nearest 5 psi

Zone I

2100	840	480	350	809	121	742	815	8/64
2950	856	575	275	832	125	759	830	8/64
3625	869	640	225	849	128	771	845	8/64
4150	879	700	175	863	131	780	855	9/64
4600	887	740	150	874	133	787	860	9/64
4950	894	780	125	884	135	792	865	9/64
5100	897	795	100	889	136	795	870	9/64

* Set to nearest 25 psi
**Set to nearest 5 psi

TABLE 3.4125
DESIGN RESULTS—WELL 2, ZONES A AND B, DUAL GAS LIFT INSTALLATION, BOTH ZONES INTERMITTENT LIFT

Type valve*	Surf close press. (psi)	Valve depth (ft.)	Depth close press. (psi)	Spring press. (psi)	Depth temp. (°F)	80° F test rack press. (psi) Dome	Set **	Orifice size (in.)
CCL	700	2175	736	350	104	700	780	8/64
CCL	700	2940	749	350	112	702	780	8/64
CCL	700	3705	761	350	121	699	780	8/64
CCL	700	4470	774	350	129	699	780	8/64
CIL	700	5100	784	350	136	699	780	None

Zone A

Valve calculations at depths 2175 ft. through 4470 ft are identical to those for Zone B. The pilot operated valve installed in the short string at 5100' has the following design results:

P	730	5100	818	None	136	730	810	None

*Nomenclature:
CCL—Combination fluid-pressure valve for constant lift
CIL—Combination fluid-pressure valve for intermittent lift
P—Standard pilot valve
**Set to nearest 5 psi

TABLE 3.4126
DESIGN RESULTS—WELL 3, ZONES X AND Y, CONTINUOUS LIFT-X, INTERMITTENT LIFT-Y, ALL VALVES SET AT 700 PSI SURFACE OPENING PRESSURE

Zone X

Type valve*	Valve depth (ft.)	Depth close press. (psi)	Design depth tbg. press. (psi)	Spring press. (psi) **	Depth temp (°F)	80° F test rack set press (psi) Dome	Set ***	Orifice size (In.)
CCL	2100	735	480	250	124	672	745	6/64
CCL	2700	745	545	200	128	674	750	6/64
CCL	3175	752	600	150	131	576	750	6/64
CCL	3525	758	640	125	134	678	755	6/64
CCL	3800	763	665	100	136	680	755	7/64
CCL	4000	766	685	100	137	681	755	7/64

Zone Y—Pilot Valve set at 720 psi surface closing pressure; All other valves set at 700 psi surface closing pressure.

CCL	2175	736	386	350	125	670	745	8/64
CCL	2940	749	399	350	130	678	755	8/64
CCL	3705	761	411	350	135	683	760	8/64
CCL	4470	774	424	350	140	688	765	8/64
CCL	5253	787	437	350	145	693	770	8/64
CCL	6000	799	449	350	150	698	775	8/64
P	6200	836	390	None	151	720	800	None

*Nomenclature:
CCL—Combination fluid-pressure valve for constant lift
CIL—Combination fluid-pressure valve for intermittent lift
P—Standard pilot valve
**Set to nearest 25 psi
***Set to nearest 5 psi

TABLE 3.4127
DESIGN RESULTS—WELL #4, ZONES M AND N, DUAL GAS LIFT INSTALLATION, BOTH ZONES INTERMITTENT LIFT

Zone N

Type valve*	Surf close press (psi)	Valve depth (ft)	Depth close press (psi)	Spring press. (psi) **	Depth temp. (°F)	80° F test rack press (psi) Dome	Set ***	Orifice size (In.)
CCL	700	1548	726	350	92	701	765	8/64
CCL	700	2593	743	350	105	700	756	8/64
CCL	700	3450	757	350	115	701	765	8/64
CCL	700	4186	769	325	124	699	765	8/64
CCL	700	4689	781	330	133	701	765	8/64
CCL	700	5524	791	300	140	701	765	8/64
P	730	6160	836	None	148	731	800	None

Zone M

Although these valves are to be set 60 ft. shallower than the valves calculated for Zone N, the spacing will remain the same. Therefore, valves at depth 1488 through 5524 ft. will be set identically to the above valves from 1548 through 6160 ft. A combination valve for intermittent lift will be installed at bottom valve depth of 6100 ft. Calculations for this valve are as follows:

CIL	700	6100	801	300	147	701	765	None

*Nomenclature:
CCL—Combination fluid-pressure valve for constant lift.
CIL—Combination fluid-pressure valve for intermittent lift
P—Standard Pilot Valve
**Discussed in test of design problem
***Set to nearest 5 psi

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3.5 COMPRESSOR SYSTEMS

3.51 Introduction

The selection of a suitable compressor and the design of a complete gas lift system for compressor operation must be considered in many of today's gas lift systems.

In the early days of gas lift, it was common to compress air to lift wells. A so-called "air-for-hire" boom existed during early Spindletop days.

Air lift was introduced in 1901 in Spindletop and continued for more than 20 years. Air lift was also introduced before 1904 in the Sour Lake Field, Texas; Batson's Prairie, Texas; Evangeline Field, Louisiana; and Mamou, Louisiana. Air was used extensively for several years; therefore, many air compressors were utilized.

Air always presented the problem of corrosion as well as the danger of explosive mixtures. Therefore, the modern trend is to use compressed gas instead of compressed air.

A more recent trend has been the use of air by first removing all oxygen by combustion.[1,2] This essentially leaves pure nitrogen along with some CO_2. The CO_2 is generally conceded to be undesirable, but some controversy exists as to its corrosive action.

However, field gases are still used in most gas lift systems. Nitric oxide and all oxygen definitely must be removed to eliminate the corrosion. If this process proves to be as successful as predicted, a relatively economical source of lift gas will be available. This gas (mostly nitrogen) must then be compressed to the desired pressure to operate a gas lift system.

3.52 Classification of compressor systems

Compressor systems may be classified as follows:

(1) *Open system.* A system in which gas from a low pressure system is compressed to operating lift pressure, utilized for lift purposes, then returned to the low pressure system for sales, etc. No attempt is made to rotate the gas.

(2) *Semiclosed system.* This system is basically the same as the *open*, except the gas coming from the wells is rotated so it can be recompressed. However, ample make-up gas is available to keep the system charged. In other words, no attempt is made to rotate all the gas. A low-pressure sales outlet takes gas from the wells.

(3) *Closed rotative system.* In this system the gas is rotated from the compressor to the well, separator, and back to the compressor. The gas makes a complete cycle. A properly-designed system should require no make-up gas. However, make-up gas may be needed to charge the system initially or the system could possibly be started with air. Exercise extreme caution when using air. Air purifiers should be attached to the com-

pressor. The air-gas mixture should not be recompressed. Note a typical system in Figures 3.51 and 3.52.

3.53 Design of the compressor system

3.531 Introduction

When designing any system, a complete analysis of the entire gas lift system—gas pressures, volumes, line sizes, suction pressure, discharge pressure, hydrates—must be made.

The type of gas lift well (intermittent or continuous flow) will influence the design of the system. The most difficult design is a one well intermittent closed rotative system. To maintain a fixed horsepower, a constant suction pressure must be maintained at the compressor. This is difficult with a one well intermittent system.

Maintaining a constant suction pressure without using excessive make-up gas can be difficult if proper low pressure storage facilities are not considered. However, most compressors will tolerate quite a variance in suction pressure without affecting normal compressor operation. A compressor normally operating at 25 psig suction pressure may drop to 2-3 psig suction pressure between cycles without causing any difficulty. This would not be the ideal design situation since a typical unit might vary from 200,000 cfd discharge at 20 psig suction down to 100,000 cfd discharge at 2 psig suction for a constant discharge pressure.

Good operation can be expected by installing large low-pressure volume chambers. A typical system may have a suction pressure that varies from 30 psig down to 5 psig during an intermittent cycle. The discharge volume, in turn, may vary from 250,000 cu ft per day to 100,000 cfd for a variable, discharge pressure.

As the suction pressure decreases, the output volume decreases. Thus, the compressor sizing must consider a variable suction pressure. The compressor must then be capable of operating at the lower suction pressure so the volume chambers may be depleted between stages. Also the gas given by the well itself may be picked up

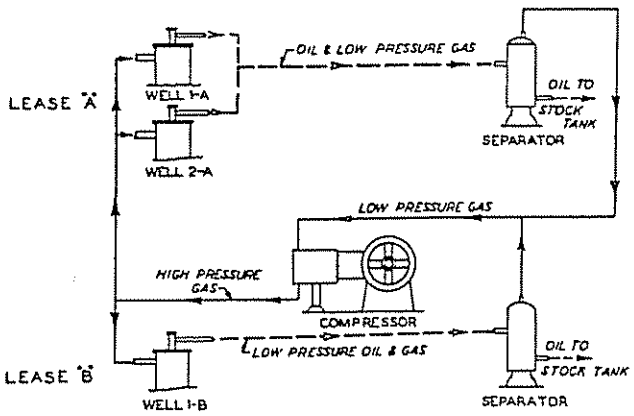


Fig. 3.51 Rotative Gas Lift System (after Winkler)[3.53]

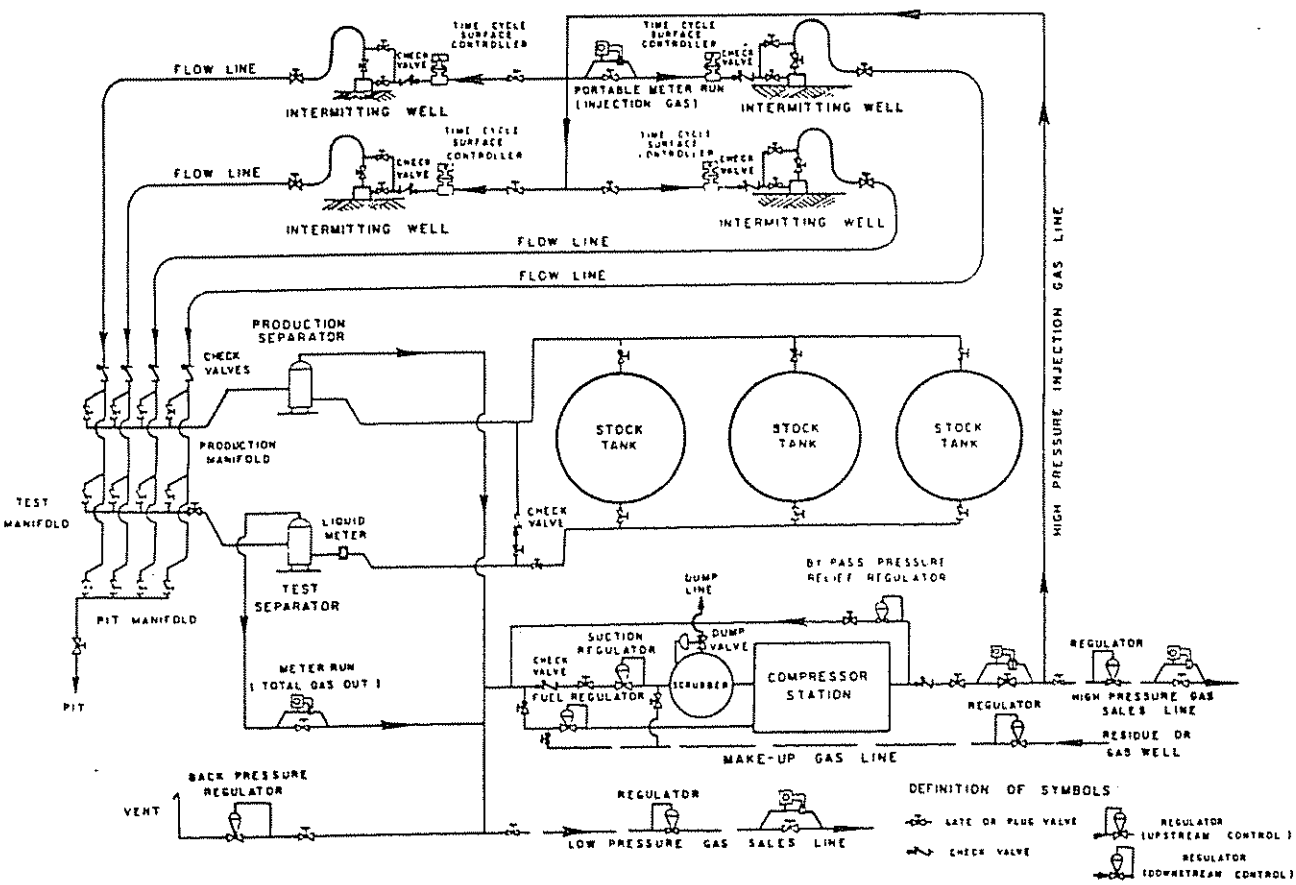


Fig. 3.52 Detailed Flow Diagram of Closed Rotative Gas Lift System for Multi-Well Installation (after Winkler)[3.53]

between injection cycles by the compressor at the lower suction pressure.

Compressors will operate satisfactorily under these conditions with the installation of carburetor actuators. This is necessary because two stages of compression will be utilizing the same power source and a variable suction pressure either underloads or overloads the low-pressure cylinder. If possible, the system should be designed with adequate low-pressure storage to prevent the suction pressure from varying over a wide range.

3.532 Factors to consider when designing a compressor system

Numerous factors are considered in the design of any system:

(1) Number and location of wells, the battery location, location of all equipment, survey of terrain, etc.
(2) Individual gas lift valve design for each well and the type of lift anticipated (continuous or intermittent)
(3) Gas volumes needed with an estimate of the peak demand at any one time
(4) Injection gas pressure needed at the well heads, which determines the discharge pressure of the compressor
(5) Separator pressure to be carried on each well and/or lease which in turn determine the suction pressure on the compressor.
(6) Gas distribution system
(7) Low-pressure gathering system
(8) Availability of make-up gas
(9) Availability of gas sales outlet
(10) Evaluation of system under freezing and hydrate conditions
(11) Sizing of the compressor

Each of these factors should be analyzed before starting the final layout of the system. A discussion of each factor is necessary if a good design is to result.

3.5321 Location of all lease equipment

The compressor or compressors must be centrally located if possible so a minimum pressure loss will occur because of friction loss in long lines. If lines are to be very long, the friction drop between the compressor and the injection well could be appreciable. Make a plan of the entire lease to show equipment location in complete detail, indicating the best location for all distribution and gathering lines. Study the best use of all present equipment at this time.

3.5322 The individual gas lift valve design for each well

The most important decision is to determine whether the well will be placed on continuous flow or intermittent flow. If it is a borderline well—performing equally well with either method—then the well should be placed on continuous flow for more efficient compressor operation. The continuous flow installation presents less trouble because a continuous output of fluid and gas results in a constant separator pressure, reflecting a constant suction pressure on the compressor. Also, a continuous injection of gas into the wells means a smoother operation compared to intermittent injections.

If the well is intermittent, determine the frequency of injection gas cycles as well as the gas volume needed per cycle. Determine the length of time required for the injection gas to enter the well per cycle and the length of time for this gas to leave the separator. Remember that slug interval time may vary considerably from well to well.

Intermittent lift offers a choice of gas lift valve types that can be employed. Cycles may be controlled from the surface by intermitters, and high-pressure volume chambers may be required to supply the peak gas demand. If a tubing-opened valve is used, the cycles can be controlled by the well feed-in characteristics. If a combination fluid-operated, pressure-operated valve is used, the cycles can be controlled to some extent by both the feed-in characteristics of the well and casing pressure buildup. A fluid-tripped valve with a casing pressure closing offers a good intermittent valve for compressor operation.

The peak volume of gas needed for intermittent wells can be controlled by placing a pilot-operated valve in the well. This valve can open on a predetermined fluid load only, can open on a casing pressure buildup, or will operate on a combination of tubing and casing pressure buildup. The casing annulus can be used for storage of high-pressure gas until it is needed.

This is one way to eliminate both a surface intermitter and a surface discharge volume chamber. The gas is metered into the casing through a choke or needle valve. One objection to this type of valve is that it takes control, to some extent, away from the operator when the well is to be "kicked." This may cause trouble for low-pressure volume storage. If several wells kicked at the same time, the compressor could not handle all the suction gas and a large amount of gas might be flared. This would increase the demand for make-up gas.

By establishing the difference in the two types of flow, it becomes evident that intermittent flow offers more of a problem for rotative compressor systems than does continuous flow. Ideal compressor operation is realized when constant suction pressures and constant discharge pressures are maintained. This condition is easy to maintain in continuous flow operations because of the continual supply of suction gas available from the separator, brought about by a constant production rate of fluid and gas. Also, the compressor discharge pressure varies very little in maintaining a constant rate of injection gas into the well. The only problem would be a loss of pressure in exceptionally-long gathering lines.

In the design of any system, establish the number of wells to go on continuous flow and the number of wells to go on intermittent flow. If surface controllers are used, determine the peak demand of gas. This can be found by determining the number of wells that could be kicking together at any one time. This information would come from the individual well design. If exact well characteristics are unknown, estimate the frequency of cycles, including gas volume per cycle.

3.5323 Gas volume needed

Determine the total gas needed for each continuous and intermittent flow well. Determine the peak demand for gas from each individual well design. This determination should be no problem for a few intermittent wells, but several wells on the same system might require gas simultaneously.

In this case the peak demand for gas would be the gas used continuously by all continuous flow wells plus the gas used by intermittent wells at the same time. If injection gas intermitters are used, their time intervals should be staggered so no two intermittent wells are taking gas at the same time. If fluid-tripped pilot valves are used, the peak demand for gas is partially or completely eliminated since gas is throttled into the casing by a choke or needle valve.

Determine gas volumes as closely as possible so correct compressor sizing can be made. Each rotative compressor system will have to be considered as an individual unit, and a gas excess of 5-10% should be allowed for so any variations (such as surging) can be properly handled.

3.5324 Injection gas pressure

The injection gas pressure needed will probably be a compromise between efficiency, high pressure lines, and fittings. For continuous flow wells, the most efficient lift possible is from the bottom. However, this pressure may be unreasonably high for some of the intermittent wells. The problems encountered for each system and each individual well must then be weighed and the injection pressure decided upon. Consider such factors as injection line length, main gas line size, and volume chamber size.

The main system discharge pressure must be high enough to take care of intermittent well peak demands. If the main system discharge line is large enough, high pressure volume chambers may be unnecessary and can be smaller.

The injection gas pressure needed also determines the compressor discharge pressure, which is very important in rating the compressor itself. Certainly, the higher the injection pressure needed, the higher the discharge pressure of the compressor needed thereby increasing the compressor hp requirements for the same volume of gas at a constant suction pressure.

The compressor should be designed to discharge some 5-15% higher gas pressure volume than calculated to take care of line losses, temperature reductions, safety factor for under-design, or increased fluid volumes at a later date over and above those anticipated in the original compressor estimates. If wells are to be intermitted from a storage of gas in the discharge system, make sufficient allowance for discharge pressure over and above that pressure required in the casing. Make this allowance so, should the wells be intermitted from a surface controller, sufficient available upstream gas will prevent the wells from being starved for gas.

3.5325 Separator pressure and suction pressure

A good understanding of the reservoir is necessary to evaluate the suction pressure on the compressor. The suction pressure will reflect on separator pressure, which, in turn, reflects on tubing pressure at the well head. This tubing pressure influences the back pressure on the sand face, which directly influences fluid feed-in rate.

For example, a low BHP high PI intermittent well with a PI of 2 would realize a decrease in production of 2 B/D for each 1 psi increase in sand face back pressure. High PI continuous flow wells would be influenced in the same manner. The low suction pressure must be balanced against compressor hp: the lower the suction pressure, the higher the compressor hp needed to obtain the gas discharge pressure and volume required.

3.5326 Distribution system

The distribution system consists of all injection gas lines and auxiliary equipment to route the compressor discharge gas to the various wells. The following items may be included.

Main injection line. Larger systems generally utilize one large main line from which individual takeoff lines go to each well. This line may vary in size, depending upon gas requirements.

Individual well lines. These are generally 2" lines leading from the main line to each well, but the size may vary depending upon demand.

High pressure volume chambers. These are volume chambers necessary to fulfill the peak gas demands of the intermittent wells. This chamber can be built from large pipe or the casing of an abandoned well.

Various formulas are available for calculating line loss in gas lines. One commonly used is Weymouth's formula (see Fig. 3.53).

$$q_g = 18.062 \frac{T_b}{p_b}\left[\frac{(p_1^2 - p_2^2)}{\gamma_g T L z_a} d^{16/3}\right]^{1/2} \qquad (3.51)$$

where:

q_g = gas flow measured at T_b and p_b in scf/hr
L = length flow line in mi
d = internal diam in in.
p = pressure in psia
γ_g = gas sp gr
T = average line temp in °R
z = average compressibility factor

EXAMPLE 1:

Given operating pressure at well = 800 psia
$\gamma_g = 0.60$
Compressor discharge pressure = 900 psia
Line length = 1 mi
q_g = 90,000 scf/hr
Average line temp = 110°F.
z = 0.90
T_b = base temp = 60°F = 520°R
p_b = base pressure = 14.7 psia

What line size is necessary?

$$90{,}000 = 18.062\left(\frac{520}{14.7}\right) d^{5/3} \sqrt{\frac{(900)^2 - (800)^2}{(0.60)(1)(0.90)(570)}}$$

$$d^{8/3} = \frac{(90{,}000)(14.7)}{(18.062)(520)(23.4)}$$

$$d^{2.67} = 6$$

d = 1.95 in.

Therefore, use 2" nominal line pipe.

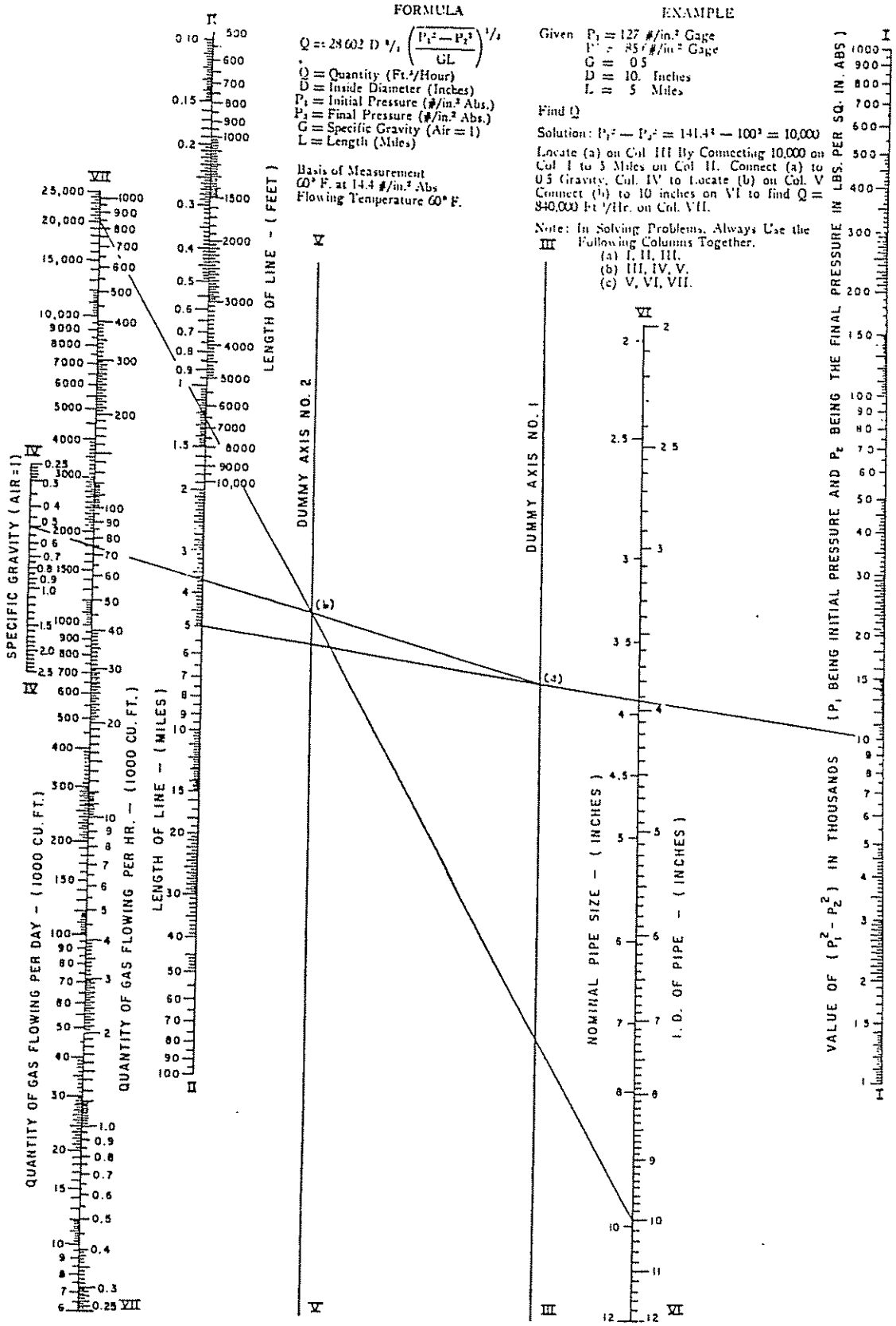


Fig. 3.53 Alignment Chart for Solution of Gas Flow in Small High Pressure Lines (Courtesy Natural Gasoline Supply Men's Association)[3.55]

EXAMPLE 2:

Two intermittent wells are equipped with 5½", 17 lb/ft casing and 2⅜" OD tubing. They are lifting from 7000 ft, each requiring 5000 scf/cycle of gas with a surface operating pressure of 700 psia. A valve requiring all gas to be injected by an intermitter is used. For a compressor discharge pressure of 900 psia, what size injection volume chamber is needed?

(a) Assume all gas will come from a 30" pipe near the well. Approximation method: The drop in pressure on the 30" pipe is:

$$900 - 700 = 200 \text{ psi}$$

$$\frac{200}{14.7} = 13.6 \text{ atm}$$

$$(13.6)(\text{vol}) = 10{,}000 \text{ scf}$$

$$\text{vol} = \frac{10{,}000}{13.6} = 736 \text{ cu ft of space}$$

$$L\frac{\pi}{4} \cdot \frac{(30)^2}{(144)} = \frac{736}{1}$$

$$L = \frac{(736)(144)}{(0.785)(900)} = 150 \text{ ft}$$

We have neglected temp and compressibility which changes very little for a 200 psi difference.

(b) Assume that an abandoned well equipped with 7 in., 23 lb/ft casing is being used for gas storage and is located near the intermittent well and is 6000 ft deep. How much gas could the storage well furnish in dropping from 900 psia to 700 psia?

Approximate solution:

$$\text{Volume of storage well} = (0.2210)(6000) = 1326 \text{ cu ft/atm}$$

$$\frac{900 - 700}{14.7} = 13.6 \text{ atm}$$

$$(13.6)(1326) = 18{,}000 \text{ scf supplied}$$

As recalled, the gas volume needed is 10,000 scf per cycle for two wells. This once-abandoned well can supply almost double the volume needed for the two wells.

(c) Assume that the compressor is supplying 500 scf/min to the wells of Example 2, and that the injection cycles cover a period of 2 minutes. This means the volume chamber could be reduced by the amount of (2 × 500) = 1000 scf, which is supplied by the compressor itself. Also, injection line storage could account for a portion of this gas.

A common practice is to carry a main line pressure of at least 100 psi greater than well operating pressures to ensure ample gas volumes and pressures. A choke downstream from the intermitter will increase the period of time by which a well is supplied with gas, thereby reducing peak demand requirements. However, this practice is not recommended, since the valve port itself may be robbed of gas.

Again, if the valve being used allows gas storage in the casing annulus, the volume chamber may be eliminated or may be reduced considerably in size. It is risky to design the valve with just the correct amount of spread so all the gas is choked in at the surface. If for some reason the well is weaker than anticipated and fails to feed in the required tubing load, too much spread will exist and too much gas will be used. A valve with a tubing-pressure trip and a casing-pressure close is an excellent valve in offering spread flexibility.

3.5327 Low-pressure gathering system

The design of the gathering system is particularly important since a good design eliminates the use of make-up gas. The gathering system is composed of all the lines, separators, and low-pressure volume chambers that supply gas to the suction side of the compressor. Again, the setup is relatively simple for continuous flow wells since a constant source of gas is available from injection gas and produced gas.

For intermittent systems the problem becomes more complicated. Where several wells are supplying one compressor, the production slugs from each well should be staggered to avoid overloading the separator with tail gas behind the slug and vent gas that should go to the compressor. This is difficult to obtain with automatic fluid-operated valves since the kicking-time decision is left to the well. It would be undesirable to have several slugs from different wells surfacing at the same time. Although intermittent control at the surface offers the additional problem of high-pressure volume storage, it does allow staggering of intermittent slug production. This simplifies the calculations to determine the necessary low-pressure volume storage chambers.

A one well intermittent system is the most difficult one to design. In particular, if the cycles are very far apart, the compressor will be starved of gas between cycles and excessive make-up gas will be required. The only solution requires that low-pressure volume chambers be constructed to save gas for the compressor.

However, certain other precautions must be taken to eliminate excessive surges on the separator, which vents much-needed gas. The combined fluid and gas slug from the well can be slowed considerably by choking at the surface. To prevent excessive fallback and slippage, place the choke at the separator instead of at the well head. This allows the fluid slug to accumulate in the flowline, yet prevents excessive surging on the separator. This in turn lengthens the time it takes for the gas to pass through the separator, the low-pressure volume chamber, and to the compressor.

The low-pressure volume calculations and line sizing can be handled by calculations similar to those given in Section 3.5326.

EXAMPLE PROBLEM:

A well requires 10,000 scf/cycle of gas and is kicked every 30 minutes. Solution gas is produced continually at the rate of 100 scf/min to the separator. The fluid and gas slug is dissipated in 10 minutes, leaving 20 minutes in which the compressor is being robbed of gas. The compressor suction can pick up gas at the rate of 500 scf/min at 75 psia.

How much low-pressure storage is necessary, assuming that the suction pressure is 75 psia? The separator volume is 100 cu ft. The line volume to the compressor is 300 cu ft. The lowest suction pressure advisable is 5 psig or 20 psia, at which the compressor is picking up at approximately 100 scf/min. Assume that the low-pres-

sure storage then supplies the additional 5000 scf needed during the next 20 minutes. The pickup rate will vary with suction pressure during this period.

Solution:

The total volume needed during the 20 minutes is 5000 scf to the compressor. The solution gas supplies (100)(20) = 2000 scf during this period. The low-pressure volume chamber must then be capable of delivering (5000 − 2000) = 3000 scf between pressures of 75 psia and 20 psia.

$$\text{Total vol} = \text{separator vol} + \text{line vol} + \text{chamber vol}$$

$$\text{Pressure drop} = \frac{75 - 20}{14.7} = 3.74 \text{ atm}$$

$$(3.74)(\text{vol}) = 3000$$

$$\text{Total vol} = \frac{3000}{3.74} = 800 \text{ cu ft}$$

The low-pressure volume chamber must contain:

$$800 - 100 - 300 = 400 \text{ cu ft}$$

Assuming this can be supplied by a 30" line, the volume per linear ft is:

$$\frac{\pi}{4} \cdot \frac{(30)^2}{144}(1) = 4.9 \text{ cu ft/ft}$$

$$\text{Total length} = \frac{400}{4.9} = 82 \text{ ft}$$

3.5328 Availability of make-up gas

Make-up gas is generally essential to charge a system originally unless some flowing wells are coming into the same battery. For one intermittent well closed system, make-up gas is almost a necessity to charge the system originally and to start again after shutdown periods. SPECIAL PRECAUTIONS ARE NECESSARY IF AIR IS TO BE USED.

For a one well system the make-up gas should enter the system downstream from the volume chambers and near the suction side of the compressor. The regulator controlling the make-up gas should be set to allow entry only after the low-pressure volume chamber has been depleted to its minimum low pressure.

3.5329 Availability of a gas sales outlet

The availability of a gas sales outlet is economical from an operational standpoint. The gas possibly lost from the separator because of surges can be redeemed by placing it into a gas sales line. In turn, this available sales line may directly influence the type of compressor system installed. It might be that a semiclosed system would be more desirable.

3.53210 Freezing conditions (hydrates)

If the possibility of the formation of hydrates exists, appropriate precautions should be incorporated in the design of the system. If the proper pressure condition exists, the formation of hydrates can occur well above the freezing point of water. Freezing can be caused by the combination of water and hydrocarbon vapors. The higher the injection pressure being used, the higher the temperature at which hydrates may be expected to form.

For example, at 2000 psig, hydrates are not uncommon at 80°F; while at 500 psig, hydrates may start forming below 50°F. Obviously, hydrates can offer a problem for practically any pressure system, especially during winter months. If hydrates will be a problem, one or more of the following procedures can be followed:

(1) Eliminate (as much as possible) all dips and sags in the distribution line. These are places where hydrates can collect.
(2) Install bleed-off valves and, preferably, some type of drip at all the low spots.
(3) Inject a chemical (alcohol, glycol, ammonia, or methanol) into the injection lines. This lowers the temperature at which hydrates will form.
(4) Install a dehydrator so the water may actually be removed from the gas.
(5) Install an injection line heater to heat the gas above the temperature at which hydrates form.

By considering all conditions that exist for any system including the lowest temperature for the year, the optimum procedure for hydrate elimination can be established. Dehydrators can be fairly expensive, but they will do an efficient job. If hydrates can be expected for only one or two months out of the year, a line heater may be all that is necessary; or chemical injection for only a few months can be used. If hydrate formation is not expected to be too severe, the installation of drips or bleed-offs in the low spots will generally be sufficient.

Figures 3.54-3.59 are from the paper "Prediction of Conditions for Hydrate Formation in Natural Gases," presented by Dr. Katz before the AIME (February 1944) and published in the *Journal of Petroleum Technology*, July, 1944.[54]

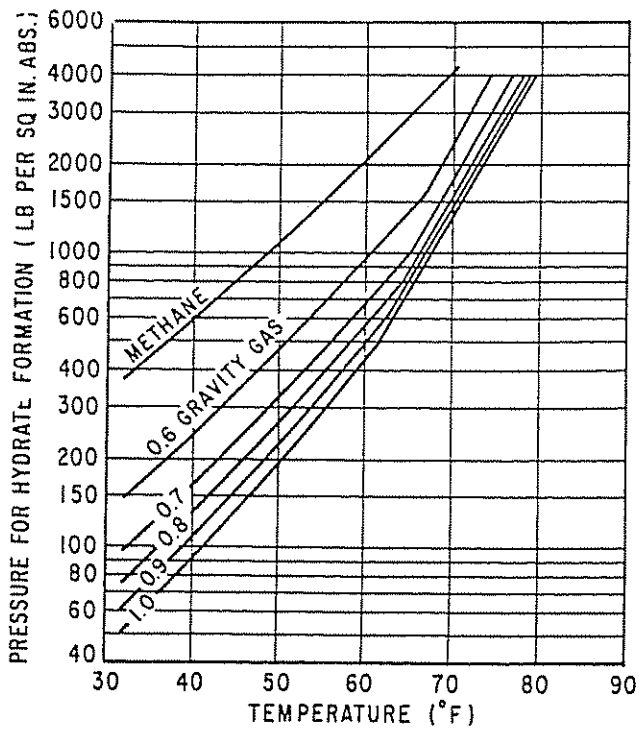


Fig. 3.54 Pressure-Temperature Curves for Predicting Hydrate Formation (after Katz[3.54], Courtesy JPT)

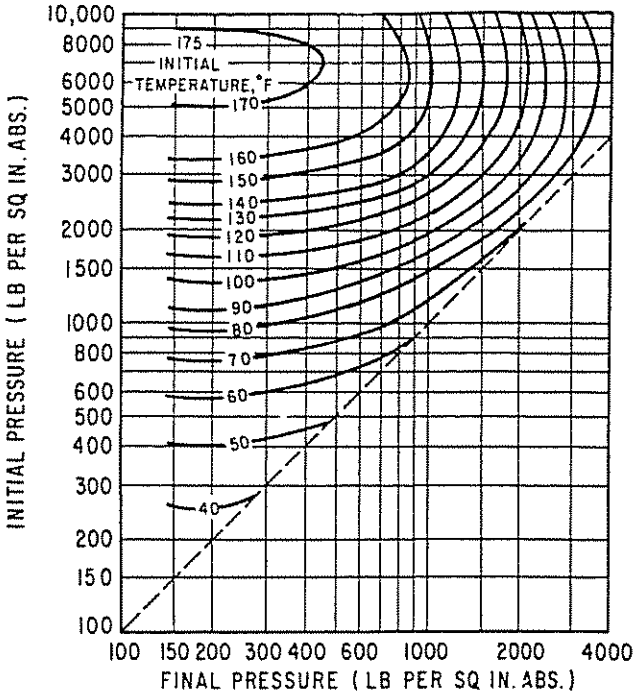


Fig. 3.55 Permissible Expansion of a 0.6 Gravity Natural Gas without Hydrate Formation (after Katz[3.54], Courtesy JPT)

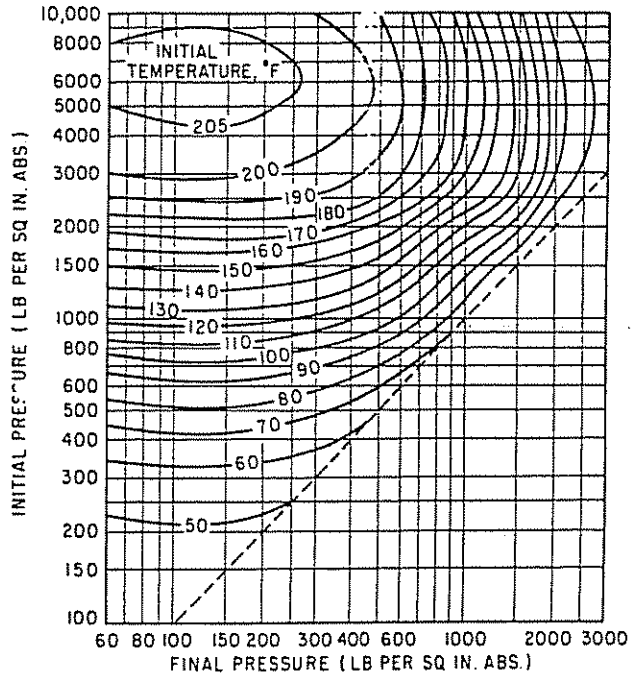


Fig. 3.57 Permissible Expansion of a 0.8 Gravity Natural Gas without Hydrate Formation (after Katz[3.54], Courtesy JPT)

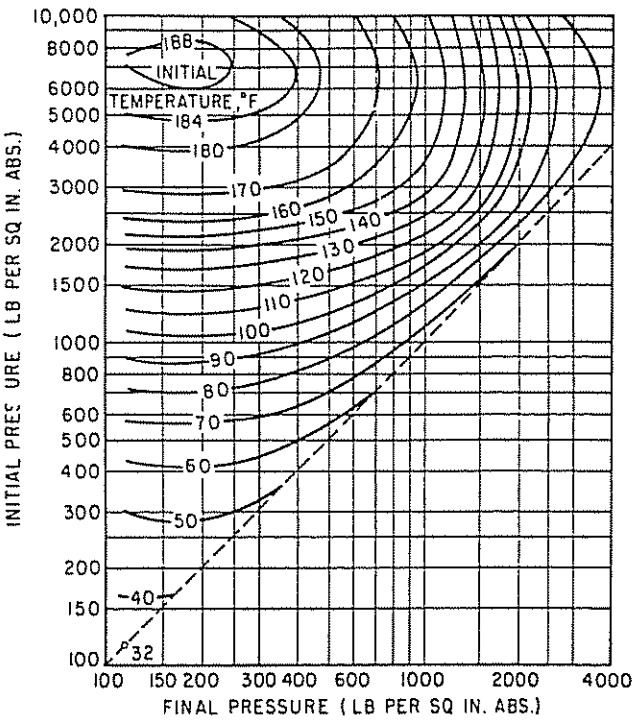


Fig. 3.56 Permissible Expansion of a 0.7 Gravity Natural Gas without Hydrate Formation (after Katz[3.54], Courtesy JPT)

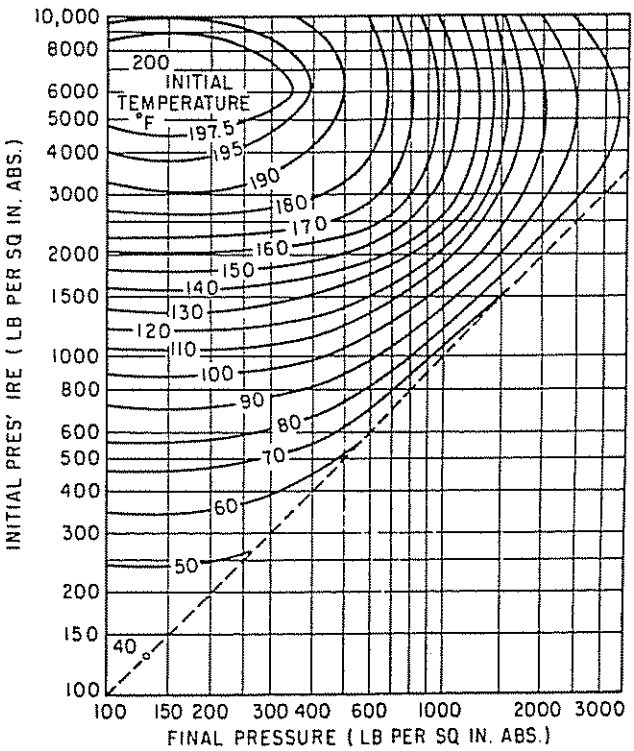


Fig. 3.58 Permissible Expansion of a 0.9 Gravity Natural Gas without Hydrate Formation (after Katz[3.54], Courtesy JPT)

The following examples are taken from the NGSMA handbook.[55]

USE OF THE CHARTS. Figures 3.54-3.59 may solve two types of problems with examples as follows:

(1) How far may a 0.6 sp gr gas at 2000 psi pressure and 100°F. be expanded without danger of hydrate formation? The answer, from Figure 3.55, is 1050 psia.

(2) How far may a 0.6 sp gr gas at 2000 psi and 140°F. be expanded without hydrate formation? The answer, from Figure 3.55, is hydrates will not form upon expansion to atmospheric pressure.

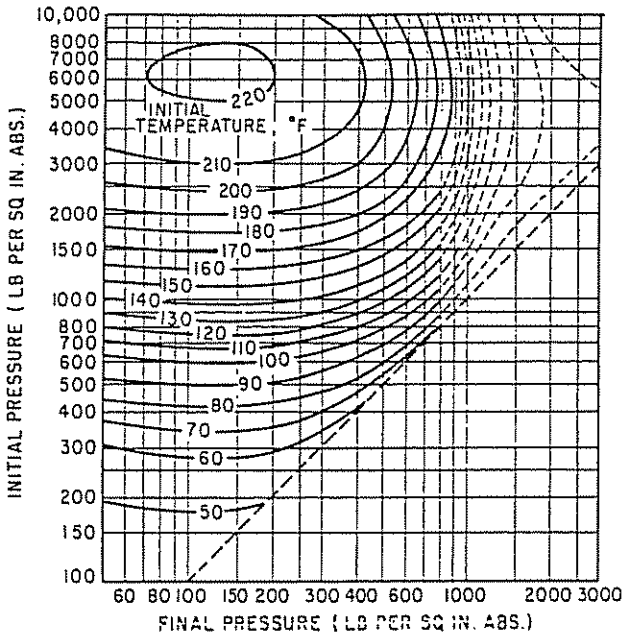


Fig. 3.59 Permissible Expansion of a 1.0 Gravity Natural Gas without Hydrate Formation (after Katz[3.54], Courtesy JPT)

(3) A 0.6 sp gr gas is to be expanded from 1500 psi to 500 psi. What is the minimum initial temperature that will permit the expansion without danger of hydrates? The answer, from Figure 3.55, is an initial temperature of 90°F. or above.

(4) A 0.6 sp gr gas at 10,000 psi and 140°F. is expanded freely and adiabatically:
   (a) At what lower pressure will the temperature become 140°F?
   (b) At what pressure would hydrates be likely to appear if sufficient water is present?

The answers, from Figure 3.55, are (a) 4300 psi and (b) 1150 psi.

3.533 Compressor selection

3.5331 Introduction

Compressors are available in every size and description to fill the needs for practically any gas lift problem. They are available in one, two, and three stages and may be direct-drive or belt-driven. Package units are available on a skid mount that can be moved easily from one location to another. Most compressors for gas lift use the available gas lift injection gas for fuel. An approximation of fuel consumption is 10 scf/hr per bhp (brake horsepower).

A compressor manufacturer representative should be consulted before making the final selection.

3.5332 Sizing the compressor

Estimating horsepower requirements. A rapid estimate of overall horsepower requirements can be made as shown below:

$$R' = \frac{P_d}{P_s}$$

$$R = (R')^{1/n}$$

$$R = (P_d/P_s)^{1/n}$$

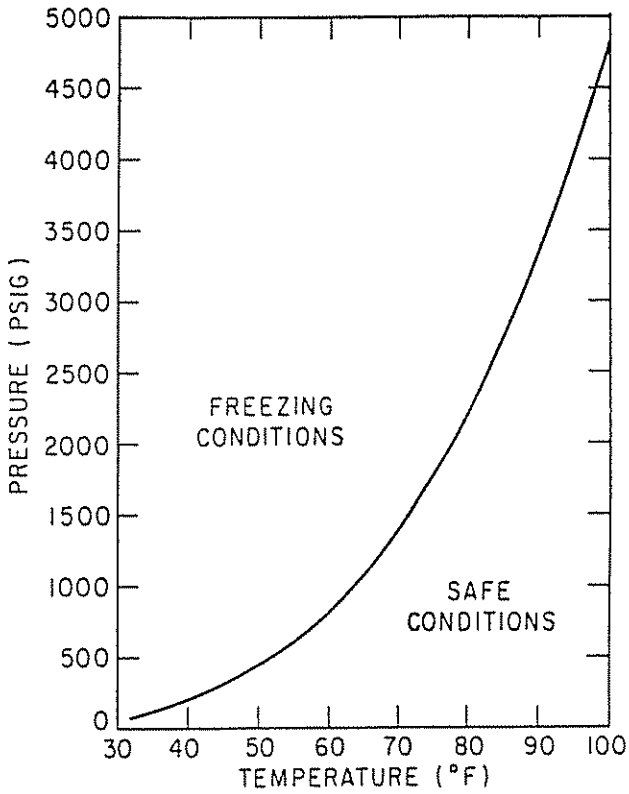


Fig. 3.510 General Correlation for Hydrate Formation

where:

R = the absolute compression ratio/stage
R′ = the overall absolute compression ratio
n = number of stages
P_d = absolute discharge pressure, psia
P_s = absolute suction pressure, psia

and,

$$P_d = \frac{(P_s)(R)^n}{(P_s)(R')}$$

where:

terms are same as above.

a. Multiply the per stage compression ratio by 1.05 to correct for pressure drop and imperfect gas cooling.
b. Multiply the corrected ratio by 23 to obtain BHP/MMCF/stage.
c. Multiply the result by the number of stages to get the brake hp per million cubic feet of gas at standard conditions (BHP/mmcf).

Usually, 10% is added to gas volume requirements to account for system losses and fuel. When R′ does not exceed 5, can use single stage compression. Use two stage compression for R′ between 4 and 30. Use three stage compression when R is between 20 and 100. When individual intake and discharge pressures and quantities are known for each stage, include approximately 5 psi pressure drop between stages to allow for loss through the cooler or coil.

EXAMPLE PROBLEM—ESTIMATED COMPRESSOR REQUIREMENTS

Given:

Suction pressure P_s = 40 psig = 55 psia
Discharge pressure P_d = 750 psig = 765 psia
Volume of gas required = 2.5 MMcf/d
Suction temperature = 90°F @ 15.0 psia

Solution:

(a) Determine overall compression ratio—C

$$R' = \frac{P_d}{P_s} = \frac{765}{55} = 13.9$$

(b) For two stages of compression

$$R = (R')^{1/n} = (15.3)^{1/2} = 3.74$$

Since 3.74 is less than 4.00, two stages are fine.

(c) Multiply by 1.05 to account for pressure drop between stages

$$(3.74)(1.05) = 3.93$$

(d) Multiply by 23 to obtain BHP/MMcf/stage

$$(3.93)(23) = 90.4$$

(e) Multiply by number of stages

$$(2)(90.4) = 180.8 \text{ BHP/stage}$$

(f) Multiply by number of MMcf to obtain total BHP required

$$(2.5)(180.8) = 452 \text{ BHP}$$

Detailed Solution Procedure

The following procedure is adapted from *Gas Compressor Calculations* by the Gardner-Denver Company.[6] This procedure does not consider supercompressibility (however, its effect on capacity is generally negligible when compressing normal field gases). When supercompressibility will produce a considerable difference in efficiency, it should be taken into consideration.

Single-stage compressor calculations

(1) Obtain operating conditions of compressor
  (a) Suction pressure (gage or absolute)
  (b) Suction temp (°F. or °C.)
  (c) Discharge pressure (gage or absolute)
  (d) Capacity required in cubic feet per minute (cfm) or cubic feet per day (cfd) at a given pressure and temp
  (e) Elevation
  (f) Atmospheric pressure at elevation (psia or in. Hg absolute)
  (g) Gas characteristics
    (1) K or N value (Fig 3.515 and 3.516)
    (2) Sp gr or molecular weight
    (3) Percentage breakdown of gas mixture by volume or weight.

(2) Determine compression ratio (CR)
  Divide absolute discharge pressure by absolute suction pressure. If CR is greater than 6.0, a two-stage compressor should be used.

(3) Determine brake horsepower (bhp) required per MMcf/D of gas

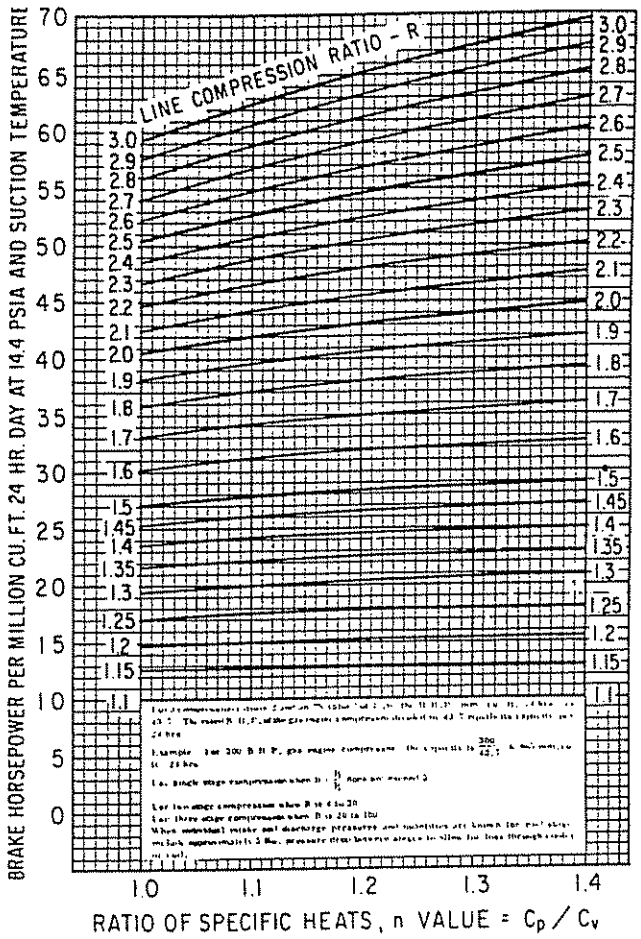


Fig. 3.511 Brake Horsepower Selection Chart (Courtesy NGSMA[3.55])

Refer to hp curves, Figures 3.511, 3.512, 3.513, and 3.514, using N value from Step 1 (g) and CR from Step 2. For using these hp curves, measure the capacity at 14.4 psia.

(4) Calculate total compressor BHP required
  Multiply BHP required per MMcf/D (determined in Step 3) by the required capacity expressed in MMcf/D measured at 14.4 psia. To correct required capacity to 14.4 psia, multiply the capacity by the given pressure in psia [step 1 (d)] and divide by 14.4 psia.

(5) Select appropriate frame size for compressor
  Refer to the cylinder and frame data sheets for the particular compressor manufacturer showing maximum allowable frame hp at full rated speed. The total BHP found in step 4 must be less than the rating of frame selected. This will vary somewhat, depending upon the type of compressor being used.

(6) Estimate probable displacement required
  Divide capacity required [from Step 1 (d)] by number of suction atmospheres* and by an aver-

*Number of suction atmospheres = absolute suction pressure divided by absolute atmospheric pressure. When capacity is measured at 14.7 psia, and the atmospheric pressure is lower than 14.7 psia, divide absolute suction pressure by 14.7 psia to get corrected suction atmospheres.

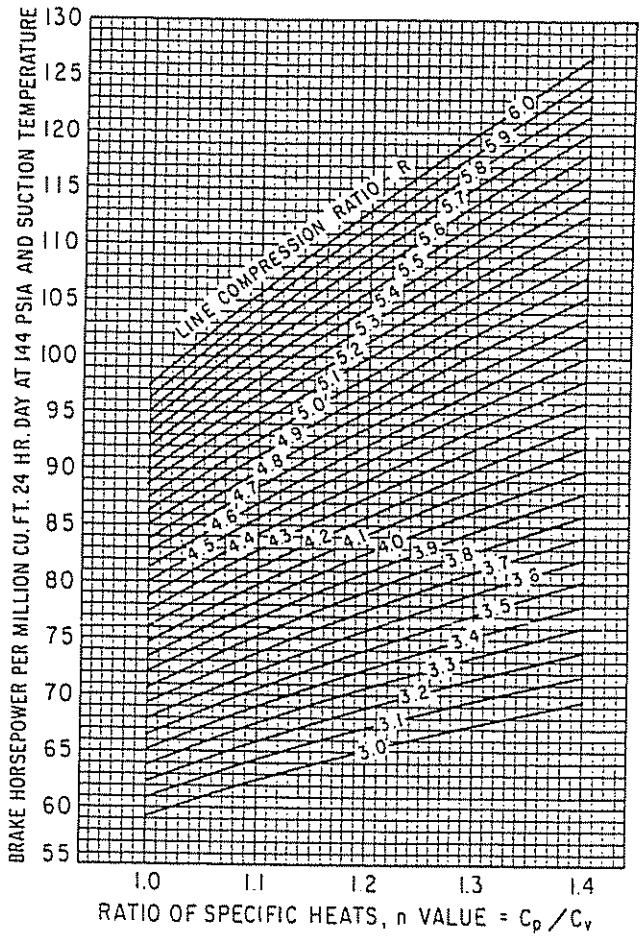


Fig. 3.512 Brake Horsepower Selection Chart (Courtesy NGSMA[3.55])

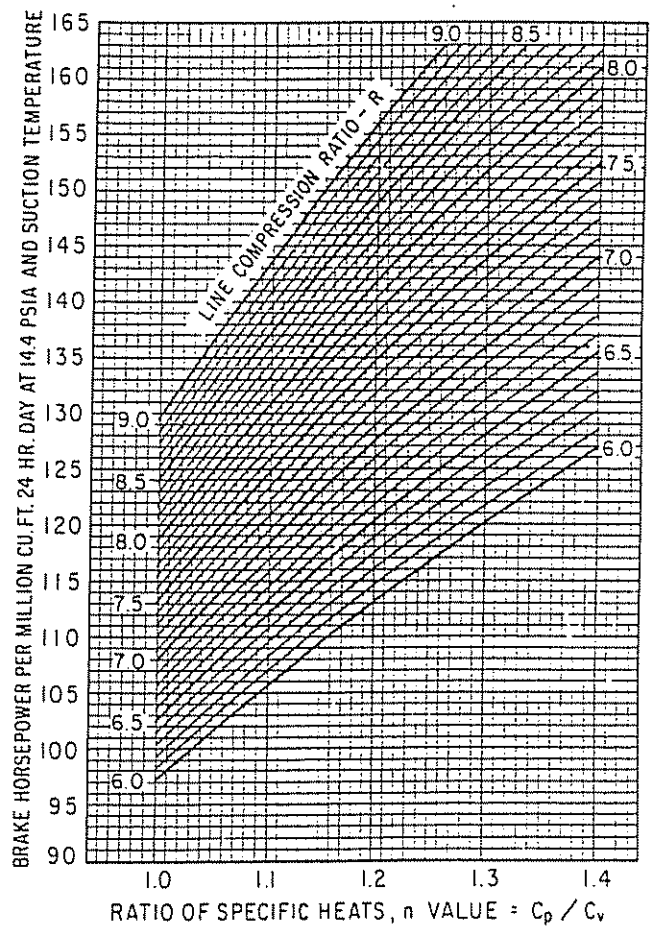


Fig. 3.513 Brake Horsepower Selection Chart (Courtesy NGSMA[3.55])

age volumetric efficiency (usually 0.75). If capacity required is stated at standard conditions and the suction temperature [Step 1 (b)] is other than 60°F, multiply the required capacity by a temperature correction which is the ratio of the suction temperature to standard temperature (both in °R).

(7) Select probable cylinder

Refer to cylinder and frame data sheets for frame size selected in Step 5. Select cylinder with displacement just larger than estimated displacement required (Step 6). Check maximum allowable pressures and maximum pressure differential of cylinder selected against operating pressures [Step 1 (a) and (c)] and operating differential pressure. Operating differential pressure = discharge pressure—suction pressure. Do not exceed maximum allowable pressures in either case. A compressor representative will furnish this information for his particular compressor.

(8) Determine exact volumetric efficiency

Find this by using the following formula:

VE = constant
(inherent cylinder clearance) ($CR^{1/n}$ — 1)

where:

Constant = VE for a particular cylinder, correcting for minor inefficiencies such as incomplete filling of the cylinder during intake stroke.

Inherent cylinder clearance = a constant for any particular cylinder expressed as a percent of the total volume.

Obtain these factors from the compressor representative.

(9) Determine the capacity of the compressor

(a) Actual capacity. Multiply the displacement of the selected cylinder (Step 7) by the volumetric efficiency (step 8) by the number of suction atm (Step 6).

(b) Capacity for calculating hp. Multiply actual capacity [Step 9(a)] by a pressure correction.

(1) The pressure correction is absolute atmospheric pressure divided by 14.4 psia.

(2) If capacity is expressed in cubic feet per minute (cu ft/min), multiply by 1440 to get cubic feet per day (cu ft/D).

Note: Inlet temperature does not affect compressor hp; therefore, make no temperature correction in Step 9(b).

(c) Capacity at standard conditions of 14.7 PSIA and 60°F Multiply actual capacity [Step

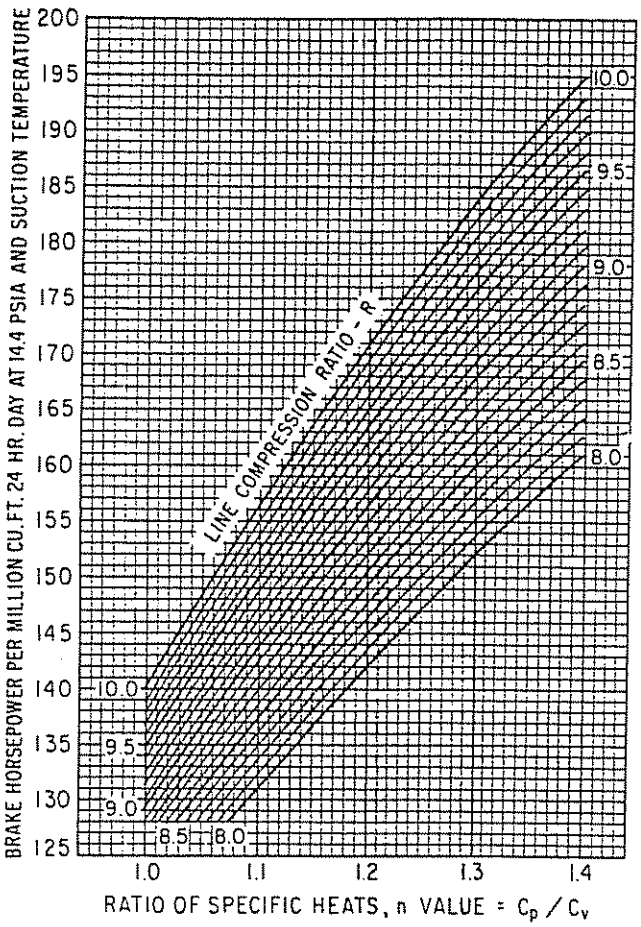


Fig. 3.514 Brake Horsepower Selection Chart (Courtesy NGSMA[3.55])

9(a)] by a temperature correction and a pressure correction.

(1) The temperature correction is 520°R. divided by operating suction temp expressed in °R. (°F. + 460°).

(2) The pressure correction is absolute atmospheric pressure divided by 14.7 psia (normally air and gas are measured at 14.7 psia, so no pressure correction is required).

(10) Calculate total compressor BHP required Multiply the capacity [Step 9(b)] expressed in MMcf/D times the bhp per MMcf/D determined in Step 3. This bhp must not exceed the maximum frame hp rating chosen in Step 5. Should the total hp required exceed the maximum frame hp, use the next larger frame size. Make new calculations starting with Step 5.

Final results:

The complete operating performance of the compressor has now been determined. It is usually expressed as follows:

Suction pressure in psig or psia
Suction temp in °F. or °C.
Discharge pressure in psig or psia
Capacity in cu ft/min or cu ft/D at a given pressure and temp
Compressor power in bhp
Compressor speed in rpm

EXAMPLE PROBLEM #1 (single-stage):

(1) Operating conditions
- (a) Suction pressure = 50 psig
- (b) Suction temp = 80°F.
- (c) Discharge pressure = 150 psig
- (d) Capacity required = 475,000 cu ft/D at 14.7 psia and 80°F.
- (e) Elevation = sea level
- (f) Atmospheric pressure = 14.7 psia
- (g) Gas characteristics = natural gas, $N = 1.26$, MW = 17.75 (molecular weight)

(2)
$$CR = \left(\frac{150 + 14.7}{50 + 14.7}\right) = 2.54$$

(3) bhp per MMcf/D = 56.3

(4) Total compressor bhp = (56.3)(0.475)(14.7/14.4) = 27.3 bhp

(5) Use frame—rated 35 bhp at 675 rpm

(6) Displacement estimate = (475,000)/(0.75)(4.4) = 144,000 cf ft/D
Number of suction atm = (50 + 14.7)/14.7 = 4.4

(7) Use 6" cylinder—displacement 155,520 cu ft/D
Maximum pressures:
Suction = 1500 psig
Discharge = 1500 psig
Differential = 176 psi

(8) Volumetric efficiency = $83 - 10.2\,(2.54^{1/1.26} - 1)$ = 71.8%

(9) Capacity of compressor
- (a) (155,520)(0.733)(0.98)(4.4) = 491,000 cu ft/D measured at 14.7 psia and suction temp (80°F)
- (b) (491,000) (14.7/14.4) = 501,000 cu ft/D measured at 14.4 psia and suction temp (80°F)

(10) Total compressor bhp = (0.501)(56.3) = 28.2 bhp

Final results:

Suction pressure = 50 psig
Suction temp = 80°F.
Discharge pressure = 150 psig
Capacity = 491,000 cu ft/D measured at 14.7 psia and suction temp (80°F)
Compressor bhp = 28.2
Compressor rpm = 675
N value of gas = 1.26

EXAMPLE PROBLEM #2 (single-stage):

(1) Operating conditions
- (a) Suction pressure = 50 psig
- (b) Suction temp = 100°F.
- (c) Discharge pressure = 150 psig
- (d) Capacity required = 800,000 scf/D (at 14.7 psia and 60°F)
- (e) Elevation = 8000 ft
- (f) Atmospheric pressure = 10.91 psia

(g) Gas characteristics = natural gas, $N = 1.26$, MW = 17.75

(2)
$$CR = \left(\frac{150 + 10.91}{50 + 10.91}\right) = 2.64$$

(3) bhp per MMcf/D = 58.6 bhp

(4) Total compressor bhp = (58.6)(0.80) (14.7/14.4) = 47.8 bhp

(5) Use frame—rated 60 hp at 514 rpm

(6) Displacement estimate = (800,000)/(0.75)(4.15)(1.077) = 276,300 cu ft/D
Number of suction atm = (50 + 10.91)/14.7 = 4.15
Temp correction = (100 + 460)/520 = 1.077

(7) Use cylinder—displacement 296,640 cu ft/D
Maximum pressures:
Suction = 300 psig
Discharge = 300 psig
Differential = 179 psi

(8) Volumetric efficiency = $83 - 8.63\ (2.64^{1/1.26} - 1)$ = 73%

(9) Capacity of compressor
(a) (296,640)(0.747)(0.978)(4.15) = 898,000 cu ft/D measured at 14.7 psia and suction temp (100°F)
(b) (898,000)(14.7/14.4) = 917,000 cu ft/D measured at 14.4 psia and suction temp (100°F)
(c) (898,000)(0.928) = 834,000 scf/D measured at 14.7 psia and 60°F
Temp correction = 520/(100 + 460) = 0.928

(10) Total compressor bhp = (0.917)(58.6) = 53.7 bhp

Final results:

Suction pressure = 50 psig
Suction temp = 100°F.
Discharge pressure = 150 psig
Capacity = 834,000 scf/D at 14.7 psia and 60°F.
Compressor bhp = 53.7
Compressor rpm = 514
N value of gas = 1.26
Elevation = 8000 ft
Atmospheric pressure = 10.91 psia

Two-stage compressor calculations

(1) *Obtain operating conditions of compressor.*
(a) Suction pressure (gage or absolute)
(b) Suction temp (°F or °C)
(c) Discharge pressure (gage or absolute)
(d) Capacity required in cu ft/min or cu ft/D at a given pressure and temp
(e) Elevation
(f) Atmospheric pressure at elevation (psia or in. Hg absolute)
(g) Gas characteristics
(1) K or N value
(2) Sp gr or MW
(3) Percentage breakdown of gas mixture by volume or weight

(2) *Determine compression ratio* (CR). Divide absolute discharge pressure by absolute suction pressure.

(3) *Determine ideal compression ratio* (ICR) *for each stage.* The ICR for each stage is found by taking the squre root of the CR (Step 2).

(4) *Calculate ideal interstage pressure and interstage atm.*
(a) INTERSTAGE PRESSURE. Multiply absolute suction pressure by the ICR (Step 3) for interstage pressure in psia. Subtract absolute atmospheric pressure (normally 14.7 psia) for interstage pressure in psig.
(b) INTERSTAGE ATM. Divide absolute interstage pressure psia by absolute atmospheric pressure. When capacity is measured at 14.7 psia and the atmospheric pressure is lower than 14.7 psia, divide absolute interstage pressure by 14.7 psia to get corrected interstage atm.

(5) *Determine bhp required per* MMcf/D *of gas.* Refer to hp curves, Fig. 3.511, 3.512, 3.513, and 3.514, using N value from Step 1(g) and ICR (Step 3) for each stage. For using these hp curves, measure the capacity at 14.4 psia.

(6) *Calculate bhp for each stage and total* bhp.
(a) bhp FOR EACH STAGE. Multiply bhp required per MMcf/D (Step 5) by the required capacity expressed in MMcf/D at 14.4 psia. To correct required capacity to 14.4 psia, multiply the capacity by the given pressure in psia [Step 1(d)] divided by 14.4 psia.
(b) Total bhp equals the sum of each stage.

(7) *Select appropriate frame size for compressor.* Refer to the cylinder and frame information showing maximum allowable frame hp at full rated speed. Consult data from particular make of compressor being used.

(8) *Estimate probable cylinder displacement for each stage.*
(a) FIRST STAGE DISPLACEMENT. Divide capacity required [from Step 1(d)] by number of suction atmospheres and by an average volumetric efficiency (usually 0.70). If capacity required is stated at standard conditions and the suction temp [Step 1(b)] is other than 60°F, multiply the required capacity by a temp correction which is the absolute suction temp in °R. (°F. + 460°) divided by 520°R.
(b) SECOND STAGE DISPLACEMENT. Divide capacity required from Step 1(d) by number of interstage atm [Step 4(b)], by an average volumetric efficiency (usually 0.70) and multiply by a temp correction. The second stage temp correction is the absolute temp of the gas to the second stage cylinder in °R divided by the absolute given temp [Step 1(d)]. Normally, 130°F. is used as the temp of the gas to the second stage cylinder.

(9) *Select probable cylinders.* Refer to the cylinder and frame information on make of compressor being used for frame size selected in Step 7. Select first and second stage cylinders with displacements just larger than estimated in Step 8. Check maximum allowable pressures and maximum pressure differential of cylinders selected against operating pressures [Steps 1(a) and (c)] and operating differential pressures. Operating

differential pressure for the first stage equals the interstage pressure (Step 4) minus the suction pressure. The operating differential pressure for the second stage equals the discharge pressure minus the interstage pressure.

(10) *Determine exact volumetric efficiency for each cylinder.* Refer to Step 8 of single-stage calculations using the ICR.

(11) *Determine first stage cylinder capacity.*
- (a) ACTUAL CAPACITY. Multiply the displacement of the selected first stage cylinder (Step 9) by its volumetric efficiency (Step 10) by the number of such atm [Step 8(a)].
- (b) CAPACITY FOR FIGURING hp. Multiply actual capacity [Step 12(a)] by a pressure correction.
  - (1) The pressure correction is absolute atmospheric pressure divided by 14.4 psia.
  - (2) If capacity is expressed in cu ft/min, multiply by 1440 to get cu ft/D.
- (c) CAPACITY AT STANDARD CONDITIONS OF 14.7 PSIA AND 60°F. Multiply actual capacity [Step 11(a)] by a temp correction and a pressure correction.
  - (1) The temp correction is 520 °R. divided by operating suction temp expressed in °R.
  - (2) The pressure correction is absolute atmospheric pressure divided by 14.7 psia. (Normally air and gas are measured at 14.7 psia, so that no pressure correction is required.)

(12) *Calculate bhp required for first stage.* Multiply the capacity [Step 11(b)] expressed in MMcf/D by the bhp per MMcf/D determined in Step 5.

(13) *Determine second stage cylinder capacity.*
- (a) ACTUAL CAPACITY. Multiply the displacement of the selected second stage cylinder (Step 9) by its volumetric efficiency (VE)(Step 10) by the number of interstage atm [Step 4 (b)].
- (b) CAPACITY FOR FIGURING hp. Multiply actual capacity [step 13(a)] by a pressure correction.
  - (1) The pressure correction is absolute atmospheric pressure divided by 14.4
  - (2) If capacity is expressed in cu ft/min, multiply by 1440 to get cu ft/D.

(14) *Calculate bhp required for second stage.* Multiply the capacity [Step 13(b)] expressed in MMcf/D by the bhp per Mmcf/D determined in Step 5.

(15) *Total compressor* hp. This is the sum of the horsepowers found in Step 12 and Step 14.

(16) *Compare first and second stage bhp.* The difference between the first and second stage bhp as calculated in Step 12 and Step 14 should not exceed the following tabulated values. If the difference is exceeded, two methods are available for balancing the load:
- (a) Change cylinder combination starting from step 8 using a revised VE based on previous calculations.
- (b) Add clearance volume to appropriate cylinder.

In some cases a combination of the above two methods will be required to balance the bhp within the tabulated values. Refer to procedure for adding cylinder clearance.

Frame size	bhp difference
1	5
2	5
3	10
4	10
5	15
6	15
7	15

Final results:

The complete operating performance of the compressor has now been determined. It is usually expressed as follows:

Suction pressure in psig or psia
Suction temp in °F. or °C.
Interstage pressure in psig or psia
Discharge pressure in psig or psia
Capacity in cu ft/min or cu ft/D at a given pressure and temp
Compressor power in bhp
Compressor speed in rpm
N value of gas

Note: In most cases theoretical delivery of the first and second stage cylinder will not be equal. This will be taken care of by a slight variation in interstage pressure and temp causing cu ft/min delivery to balance.

EXAMPLE 1 (two stage):

(1) Operating conditions
- (a) Suction pressure = 30 psig
- (b) Suction temp = 80°F.
- (c) Discharge pressure = 500 psig
- (d) Capacity required = 340,000 cu ft/D at 14.7 psia and suction temp (80°F)
- (e) Elevation = sea level
- (f) Atmospheric pressure = 14.7 psia
- (g) Gas characteristics = natural gas, N = 1.26, MW = 17.75

(2) $$CR = \left(\frac{500 + 14.7}{30 + 14.7}\right) = 11.52$$

(3) $$ICR = \sqrt{11.52} = 3.39$$

(4) (a) Interstage pressure = (44.7)(3.39) = 151.7 psia; (151.7 − 14.7) = 137 psig
- (b) Interstage atm = (151.7/14.7) = 10.32

(5) bhp per MMcf/D for each stage = 73.85

(6) (a) bhp for each stage = (73.85)(0.34)(14.7/14.4) = 25.65 bhp
- (b) Total bhp = (25.65 + 25.65) = 51.3 bhp

(7) Use frame—rated 60 hp at 514 rpm

(8) Displacement estimate
- (a) First stage = (340,000)/[(3.04)(0.70)] = 159,800 cu ft/D
  Number of suction atm = (30 + 14.7)/14.7 = 3.04

(b) Second stage = {(340,000)/[(10.32)(0.70)]} (1.093) = 51,400 cu ft/D
Temp correction = (130 + 460)/(80 + 460) = 1.093

(9) First stage—Use 6 in. cylinder (tandem displacement = 161,142 cu ft/D)

Maximum pressures:
Suction = 1500 psig
Discharge = 1500 psig
Differential = 315 psi

Second stage—Use 3½ in. cylinder (displacement = 53,280 cu ft/D)

Maximum pressures:
Suction = 1500 psig
Discharge = 1500 psig
Differential = 935 psi
Total rod load (lb):

(137 − 30)(28.27) + (500 − 137)(9.62) = 6515

(10) Volumetric efficiencies
First stage—6 in. cylinder:

$VE = 83 - 7.55\,(3.39^{1/1.26} - 1) = 70.6\%$

Second stage—3½ in. cylinder:
$VE = 83 - 13.9\,(3.39^{1/1.26} - 1) = 60.2\%$

(11) Capacity of first stage
(a) (161,142)(0.706)(3.04) = 346,000 cu ft/D
Measured at 14.7 psia and suction temp (80°F)
(b) (346,000)(14.7/14.4) = 353,400 cu ft/D
Measured at 14.4 psia and suction temp (80°F)

(12) First stage bhp

(0.3534)(73.85) = 26.1 bhp

(13) Capacity of second stage
(a) (53,280)(0.602)(10.32) = 353,000 cu ft/D
Measured at 14.7 psia and second stage suction temp (130°F)
(b) (353,000)(14.7/14.4) = 360,500 cu ft/D
Measured at 14.4 psia and second stage suction temp (130°F)

(14) Second stage bhp
(0.3605)(73.85) = 26.6 bhp

(15) Total compressor bhp
(26.1 + 26.6) = 52.7 bhp

Final results:

Suction pressure = 30 psig
Suction temp = 80°F.
Interstage pressure = 137 psig
Discharge pressure = 500 psig
Capacity = 346,000 cu ft/D measured at 14.7 psia and suction temp (80°F)
Compressor bhp = 52.7
Compressor rpm = 514
N value of gas = 1.26

EXAMPLE 2 (two stage):

(1) Operating conditions
(a) Suction pressure = 60 psig
(b) Suction temp = 70°F.
(c) Discharge pressure = 850 psig
(d) Capacity required = 2,000,000 cu ft/D at 14.7 psia and suction temp (70°F)
(e) Elevation = sea level
(f) Atmospheric pressure = 14.7 psia
(g) Gas characteristics = natural gas, N = 1.26, MW = 17.75

(2) $$CR = \left(\frac{850 + 14.7}{60 + 14.7}\right) = 11.57$$

(3) $$ICR = \sqrt{11.57} = 3.4$$

(4) (a) Interstage pressure = (74.7)(3.4) = 254 psia; (254 − 14.7) = 239.3 psig
(b) Interstage atm = (254/14.7) = 17.27

(5) bhp per MMcf/D for each stage = 74

(6) (a) bhp for each stage = (74)(2) (14.7/14.4) = 151 bhp
(b) Total bhp = (151 + 151) = 302 bhp

(7) Use frame—rated 350 hp at 1000 rpm

(8) Displacement estimate
(a) First stage = (2,000,000)/[(5.08)(0.70)] = 562,000 cu ft/D
Number of suction atm = (60 + 14.7)/14.7 = 5.08
(b) Second stage = {(2,000,000)/[(17.27)(0.60)]}(1.113) = 215,000 cu ft/D
Temp correction = (130 + 460)/(70 + 460) = 1.113

(9) First stage—use 10 in. cylinder (displacement = 640,800 cu ft/D)
Maximum pressures:
Suction = 800 psig
Discharge = 800 psig
Differential = 318 psi

Second stage—use 6 in. cylinder (displacement = 223,200 cu ft/D)

Maximum pressures:
Suction = 1500 psig
Discharge = 1500 psig
Differential = 884 psi

(10) Volumetric efficiencies
First stage—10 in. cylinder

$VE = 90 - 15.5\,(3.4^{1/1.26} - 1) = 64.5\%$

Second stage—6 in. cylinder

$VE = 90 - 20.8\,(3.4^{1/1.26} - 1) = 55.9\%$

(11) Capacity of first stage
(a) (640,000)(0.645)(5.08) = 2,100,000 cu ft/D
Measured at 14.7 psia and suction temp (70°F)
(b) (2,100,000)(14.7/14.4) = 2,143,000 cu ft/D
Measured at 14.4 psia and suction temp (70°F)

(12) First stage bhp

(2.143)(74) = 158.6 bhp

(13) Capacity of second stage
(a) (223,200)(0.559)(17.27) = 2,152,000 cu ft/D
Measured at 14.7 psia and second stage suction temp (130°F)

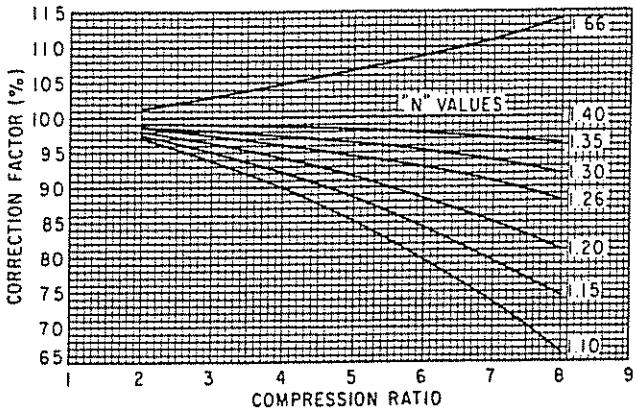


Fig. 3.515 "N" Value Correction Curve (Courtesy Gardner Denver Co.[3.55])

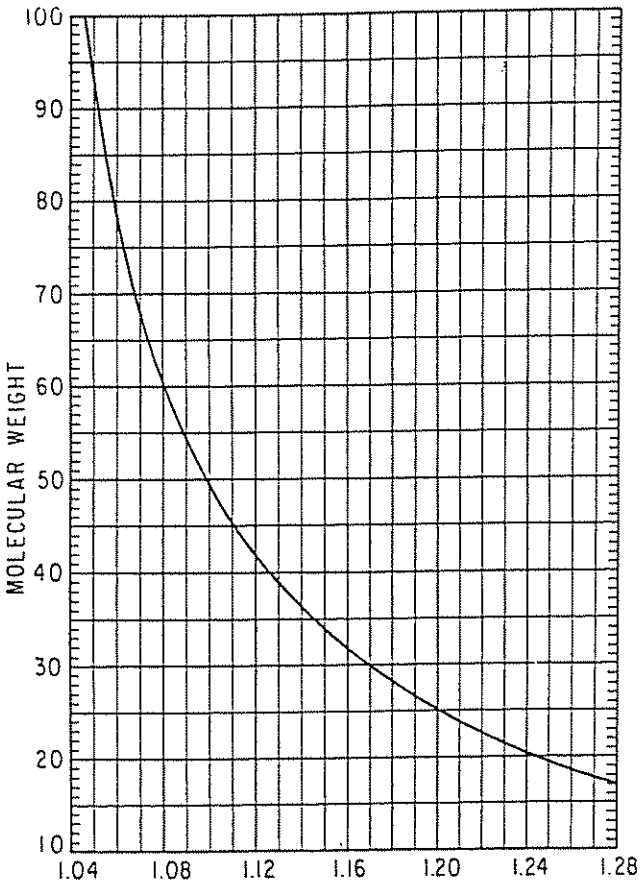


Fig. 3.516 Approximate Specific Heat Ratio for Hydrocarbon Gases (Avg. Temp. = 150°F.) (Courtesy NGSMA[3.55])

(b) (2,152,000)(14.7/14.4) = 2,199,000 cu ft/D
Measured at 14.4 psia and second stage suction temp (130°F)

(14) Second stage bhp

(2.199)(74) = 162.7 bhp

(15) Total compressor bhp

(158.6 + 162.7) = 321.3 bhp

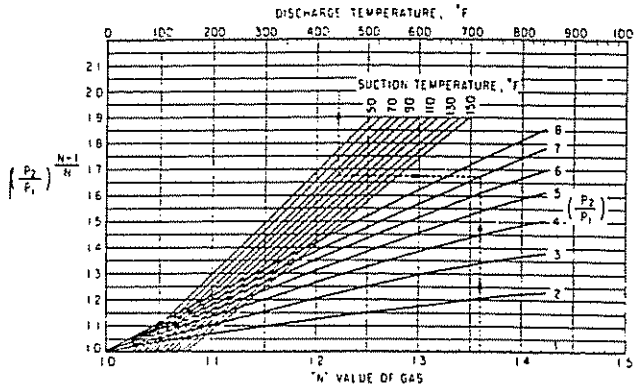


Fig. 3.517 Theoretical Gas Compression Temperature for Various Values of "N" and "CR" (Courtesy NGSMA[3.55])

Final results:

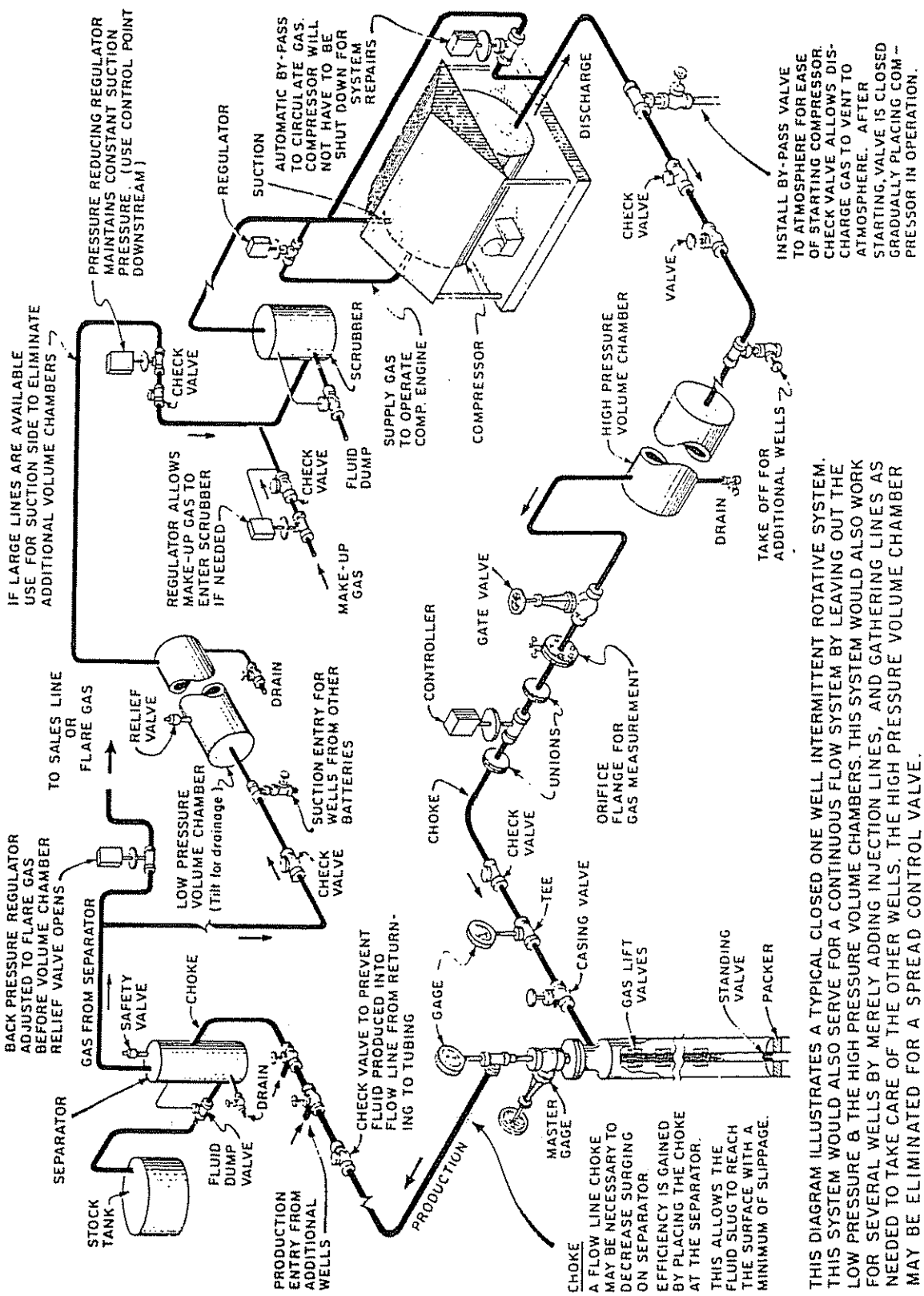
Suction pressure = 60 psig
Suction temp = 70°F.
Interstage pressure = 239.3 psig
Discharge pressure = 850 psig
Capacity = 2,100,000 cu ft/D
Measured at 14.7 psia and suction temp (70°F)
Compressor bhp = 321.3
Compressor rpm = 1000
N value of gas = 1.26

3.54 Design of a rotative compressor system

Figure 3.518 shows a typical layout for a one well rotative compressor system. This represents the most difficult system to design, particularly if surface, high-pressure volume chambers are required to store the gas until needed by the well (see Fig. 3.518). If a gas lift valve incorporating the correct spread was utilized, this chamber, as well as the surface controller shown, could be eliminated. Both of these could also be eliminated for a continuous flow well.

Regardless of whether an intermittent valve is opened by gas supplied by the intermitter or from pressure stored in the annulus through a choke, the problem of storing the produced gas is still present in a one well intermittent system. This is a tough task because good intermittent lift operation requires that gas be injected into the tubing as rapidly as practical and that slug velocity be high to ensure good recovery. This means that most of the injection gas will attempt to dissipate at the surface in a matter of a few minutes. Since this tail gas hits the separator in a hurry, the working pressure will probably be exceeded immediately and a large percentage of the gas will tend to be flared. If much of this gas is flared, the compressor will be robbed of suction gas between cycles and a great deal of make-up gas will be required.

As noted in Figure 3.518, a surface choke should be placed at the separator. This allows the whole liquid slug to leave the vertical tubing before reaching a restriction, but its main purpose is to feed the tail gas to the separator at a slower rate. This rate decrease retains a greater portion of the gas to supply the compressor suction.



THIS DIAGRAM ILLUSTRATES A TYPICAL CLOSED ONE WELL INTERMITTENT ROTATIVE SYSTEM. THIS SYSTEM WOULD ALSO SERVE FOR A CONTINUOUS FLOW SYSTEM BY LEAVING OUT THE LOW PRESSURE & THE HIGH PRESSURE VOLUME CHAMBERS. THIS SYSTEM WOULD ALSO WORK FOR SEVERAL WELLS BY MERELY ADDING INJECTION LINES, AND GATHERING LINES AS NEEDED TO TAKE CARE OF THE OTHER WELLS. THE HIGH PRESSURE VOLUME CHAMBER MAY BE ELIMINATED FOR A SPREAD CONTROL VALVE.

Fig. 3.518 One Well Intermittent Rotative Compressor System

Probably, additional low-pressure storage will be needed over and above the separator and line volumes (see Fig. 3.518). The line layout in Figure 3.518 notes a place for make-up gas entry. To keep utilization of make-up gas at a minimum, this regulator is set to open only when the suction gas is practically depleted or when the compressor suction reaches a minimum low, threatening good compressor operations.

As noted, a scrubber and fluid dump precede the compressor suction intake. This is good operation procedure unless a very dry gas is being processed. Take all precautions to keep liquids out of the compressor cylinders.

The system in Figure 3.518 can be extended to take care of numerous wells. As more and more wells are added to the system, its operation becomes smoother and smoother. In turn, less suction storage is needed because the compressor must become larger and larger to take care of greater gas demands. With a great number of intermittent wells kicking at staggered intervals, the gas is supplied more smoothly and regularly to the suction side of the compressor. If some of the wells are on continuous flow, the system is still smoother.

The following procedure is recommended in designing a compressor system:

TABLE 3.51
N value correction table for $N = 1.26$
(courtesy Gardner-Denver Company)

CR	Correction	CR	Correction	CR	Correction
1.50	0.992	3.10	0.972	4.70	0.949
1.55	0.991	3.15	0.971	4.75	0.948
1.60	0.990	3.20	0.971	4.80	0.947
1.65	0.989	3.25	0.970	4.85	0.947
1.70	0.989	3.30	0.970	4.90	0.946
1.75	0.988	3.35	0.969	4.95	0.945
1.80	0.988	3.40	0.968	5.00	0.944
1.85	0.987	3.45	0.967	5.05	0.943
1.90	0.986	3.50	0.967	5.10	0.943
1.95	0.986	3.55	0.966	5.15	0.942
2.00	0.986	3.60	0.965	5.20	0.942
2.05	0.985	3.65	0.964	5.25	0.941
2.10	0.984	3.70	0.964	5.30	0.940
2.15	0.983	3.75	0.963	5.35	0.939
2.20	0.983	3.80	0.962	5.40	0.938
2.25	0.982	3.85	0.962	5.45	0.937
2.30	0.982	3.90	0.961	5.50	0.937
2.35	0.981	3.95	0.960	5.55	0.936
2.40	0.981	4.00	0.960	5.60	0.935
2.45	0.980	4.05	0.959	5.65	0.934
2.50	0.980	4.10	0.958	5.70	0.934
2.55	0.979	4.15	0.958	5.75	0.933
2.60	0.979	4.20	0.957	5.80	0.932
2.65	0.978	4.25	0.956	5.85	0.931
2.70	0.977	4.30	0.955	5.90	0.930
2.75	0.976	4.35	0.954	5.95	0.929
2.80	0.976	4.40	0.953	6.00	0.929
2.85	0.975	4.45	0.952		
2.90	0.975	4.50	0.952		
2.95	0.974	4.55	0.951		
3.00	0.974	4.60	0.950		
3.05	0.973	4.65	0.949		

(1) Analyze the system with regard to all previously mentioned factors. Make a complete layout of the entire system, including all wells to be gas lifted, location of all batteries, separators, treaters, flowlines, etc. Note low spots and hilly terrain as well as all roads, etc.
(2) Determine the number of wells to go on continuous flow gas lift and the number to go on intermittent flow gas lift.
(3) Design the gas lift system for each well, choosing the optimum operating pressure for good operations. This design must be complete for all wells and must include cycle frequency for intermittent wells as well as total injection gas volumes needed to lift each well. In addition, determine total fluid volumes including solution gas. Solution gas can be very important in helping to supply the compressor suction.
(4) Estimate abandonment time for each well going on lift and also determine the approximate time at which additional wells must be added to the system.
(5) Select the type of gas lift valve with particular emphasis on a smoothly operating compressor system.
(6) Determine the number of cycles per day for intermittent wells and the maximum number of intermittent wells that are likely to be kicking at the same time. This information is necessary to determine peak volume demands as well as possible low-pressure storage chambers.
(7) Determine the back pressure to carry on the separators, which will be the suction pressure carried on the compressor. Do not increase back pressure to save compressor costs when it is known that excessive back pressure is materially reducing the oil production rate from individual wells. This serious error is often made in designing compressor systems. Excessive surface back pressure is the worst enemy of low BHP, high productivity wells.
(8) Locate the compressor and/or compressors. Determine the size of all distribution lines to give the design operating pressure at each well. A main distribution line may be appropriate with 2 in. takeoff lines to each well. I do not recommend using lines smaller than 2 in. because of high frictional resistance.
(9) Determine the size of all gathering lines, including any necessary low-pressure volume chambers. Make a tabulation of all low-pressure volume space including separators, treaters, lines, etc.
(10) Select an appropriate compressor capable of discharging 100 psi over the design pressure and capable of exceeding gas needs by 10-15%. This should be in excess of the 10 scf per bhp-hr needed for fuel.

It may be desirable to install permanent-type compressors or skid-mounted compressors. Also, if only a few wells are going on gas lift at the time of design, additional compressors can be added as needed. An economic analysis must be made to account for all considerations.

TABLE 3.52
COMPRESSOR SYSTEMS
Table 3.52 Bhp per MMcf/D at 14.4 psia and suction temp—for air (or gas) with $N = 1.4$
(courtesy Gardner-Denver Company)

CR	bhp	CR	bhp	CR	bhp	CR	bhp	CR	bhp
1.10	10.2	2.10	47.5	3.10	71.8	4.10	92.1	5.10	110.9
1.12	11.2	2.12	48.0	3.12	72.2	4.12	92.5	5.12	111.3
1.14	12.4	2.14	48.5	3.14	72.6	4.14	92.9	5.14	111.6
1.16	13.6	2.16	49.0	3.16	73.0	4.16	93.3	5.16	112.0
1.18	14.6	2.18	49.5	3.18	73.5	4.18	93.7	5.18	112.3
1.20	15.5	2.20	50.1	3.20	74.0	4.20	94.1	5.20	112.7
1.22	16.5	2.22	50.6	3.22	74.4	4.22	94.5	5.22	113.1
1.24	17.7	2.24	51.2	3.24	74.8	4.24	94.9	5.24	113.4
1.26	18.6	2.26	51.8	3.26	75.2	4.26	95.3	5.26	113.8
1.28	19.9	2.28	52.4	3.28	75.6	4.28	95.7	5.28	114.1
1.30	20.9	2.30	52.9	3.30	76.1	4.30	96.1	5.30	114.5
1.32	21.7	2.32	53.3	3.32	76.5	4.32	96.5	5.32	114.9
1.34	22.6	2.34	53.8	3.34	76.9	4.34	96.9	5.34	115.2
1.36	23.4	2.36	54.3	3.36	77.3	4.36	97.3	5.36	115.6
1.38	24.3	2.38	54.7	3.38	77.7	4.38	97.7	5.38	116.0
1.40	25.0	2.40	55.2	3.40	78.1	4.40	98.1	5.40	116.3
1.42	25.7	2.42	55.7	3.42	78.5	4.42	98.5	5.42	116.7
1.44	26.7	2.44	56.1	3.44	79.0	4.44	98.9	5.44	117.0
1.46	27.6	2.46	56.7	3.46	79.4	4.46	99.3	5.46	117.4
1.48	28.5	2.48	57.2	3.48	79.8	4.48	99.7	5.48	117.8
1.50	29.1	2.50	57.7	3.50	80.2	4.50	100.1	5.50	118.2
1.52	29.8	2.52	58.2	3.52	80.6	4.52	100.5	5.52	118.6
1.54	30.5	2.54	58.7	3.54	81.0	4.54	100.8	5.54	118.9
1.56	31.3	2.56	59.2	3.56	81.4	4.56	101.2	5.56	119.3
1.58	32.0	2.58	59.7	3.58	81.8	4.58	101.6	5.58	119.7
1.60	32.6	2.60	60.1	3.60	82.2	4.60	101.9	5.60	120.0
1.62	33.3	2.62	60.5	3.62	82.6	4.62	102.3	5.62	120.3
1.64	34.0	2.64	61.1	3.64	83.0	4.64	102.6	5.64	120.7
1.66	34.7	2.66	61.6	3.66	83.4	4.66	103.0	5.66	121.0
1.68	35.5	2.68	62.2	3.68	83.8	4.68	103.4	5.68	121.4
1.70	36.0	2.70	62.7	3.70	84.2	4.70	103.7	5.70	121.7
1.72	36.7	2.72	63.1	3.72	84.5	4.72	104.1	5.72	122.0
1.74	37.4	2.74	63.5	3.74	84.9	4.74	104.4	5.74	122.4
1.76	38.0	2.76	64.1	3.76	85.3	4.76	104.8	5.76	122.7
1.78	38.6	2.78	64.6	3.78	85.7	4.78	105.1	5.78	123.1
1.80	39.3	2.80	65.0	3.80	86.1	4.80	105.5	5.80	123.4
1.82	39.7	2.82	65.4	3.82	86.5	4.82	105.9	5.82	123.7
1.84	40.3	2.84	65.9	3.84	86.9	4.84	106.2	5.84	124.0
1.86	40.9	2.86	66.4	3.86	87.3	4.86	106.6	5.86	124.3
1.88	41.5	2.88	66.9	3.88	87.7	4.88	106.9	5.88	124.8
1.90	42.0	2.90	67.3	3.90	88.1	4.90	107.3	5.90	125.1
1.92	42.5	2.92	67.7	3.92	88.5	4.92	107.7	5.92	125.5
1.94	43.2	2.94	68.3	3.94	88.9	4.94	108.0	5.94	125.8
1.96	43.8	2.96	68.7	3.96	89.3	4.96	108.4	5.96	126.2
1.98	44.3	2.98	69.1	3.98	89.7	4.98	108.7	5.98	126.5
2.00	44.9	3.00	69.6	4.00	90.1	5.00	109.1	6.00	126.9
2.02	45.4	3.02	70.0	4.02	90.5	5.02	109.5		
2.04	46.0	3.04	70.4	4.04	90.9	5.04	109.8		
2.06	46.4	3.06	70.8	4.06	91.3	5.06	110.2		
2.08	47.0	3.08	71.3	4.08	91.7	5.08	110.5		

3.55 Summary

The installation of compressors can be critical due to the high cost of compressor horsepower. In many cases, the changing out of flow lines or the addition of another parallel flow line will permit an increase in suction pressure without lowering the wellhead pressure. This may permit a drastic reduction in compressor hp to as much as 100% depending upon the suction pressures.

One of the first things to be considered in design of a new installation or to be checked on existing installations is restrictions from the separator to the suction of the compressor. Sometimes, the indiscriminate use of restrictions, such as a regulator, may materially reduce suction pressure.

Also, the cost of additional or larger surface flow lines to reduce wellhead pressure and, hence, raise the suction pressure should be compared to cost of additional compressor hp to accomplish the same results.

A complete analysis should be conducted on compar-

ing production rates possible by incrementally increasing the discharge pressure up to as high as 1500-2000 psi along with incremental increases in wellhead pressure. The general practice of using 60-80 psi suction pressure as a rule of thumb may be very costly in compressor hp. The suction pressure might be doubled with only a negligible decrease in production rate. Therefore, an analysis should be conducted to determine the effect of wellhead pressure on the production rate from the well.

If large compressor units are required, a quick systems analysis can be important and may reduce compressor costs by 100%.

3.56 Problems

(1) It is desired to place a well on time cycle control. The operating valve is located on 2" tubing inside 5½", 17 ft/lb casing at 7000 ft. It is to produce against 50 psig surface pressure. All gas must be supplied from a high-pressure storage chamber made from 30" ID pipe. The opening pressure of the valve at depth is 800 psi. Assuming the storage chamber can be built up to a pressure of 150 psi in excess of the surface opening pressure of the valve, determine the length of 30" ID pipe needed to supply the necessary gas per cycle ($\gamma_g = 0.65$). Temp avg. = 100°F.

(2) An abandoned well contains 2" tubing and 5½", 17 lb/ft casing to a depth of 7000 ft. If it is used for storage, what pressure buildup is required on it to lift the well in Problem 1?

(3) A compressor is located 10 miles from a field system requiring 20 MMcf/D of a 0.65 sp gr gas (avg temp = 100°F). The compressor discharge pressure is 1000 psig. What size surface line is needed to bring 925 psig to the field system? 975 psig?

(4) Explain the process whereby exhaust gases are used for gas lift.

(5) Design a one well rotative system for an intermittent well to lift from 8000 ft. The well is to be cycled every 20 minutes with a valve operating at 8000 ft at a pressure of 800 psi at depth. The gas sp gr is 0.65. Additional information:

Separator pressure = 30 psig − G/O
= 300 scf/B
Flowline = 1000 ft, 2" ID — Rate = 150 B/D
Discharge lines = 2"
Assume a valve with 100% choke control

Design the necessary low-pressure volume chambers to keep the compressor suction from dropping below 5 psig without make-up gas.

(6) A gas lift system will need 6 MMcf/D of gas at a discharge pressure of 900 psig. The suction pressure is to be 50 psig. Select a compressor or compressors to perform this job.

(7) Why are continuous flow compressor systems easier to design than intermittent flow systems?

(8) A continuous flow well is to produce 600 B/D of oil and water with a GLR of 1000 scf/B. The injection pressure is 800 psig and the separator pressure is 60 psig. Select a compressor for this one well system. ($\gamma_g = 0.65$)

(9) A gas lift system consists of 12 gas lift wells. Eight of these wells will be continuous flow with one each requiring 180 Mcf, 160 Mcf, 150 Mcf; and 5 wells requiring 140 Mcf/D at an injection pressure of 800 psig. The four intermittent wells are to cycle each hour 80 minutes, 40 minutes, and 20 minutes, respectively, requiring 4500 scf/cycle for each well. Three continuous flow wells make 800, 600, and 500 B/D, respectively, while the remaining five wells make 400 B/D each with 10% oil, having a solution GOR of 500 scf/B. The intermittent wells make 60, 80, 100, and 120 B/D (40% oil) with a solution GOR = 500 scf/B.

The separator pressure is to be 50 psig and the system is to be designed so no make-up gas is needed. Design the intermittent wells for two options:

(a) Gas is controlled by the valve spread with valves having a surface closing pressure of 650 psig and an opening pressure of 775 psig.

(b) The gas is controlled completely by an intermitter at the surface. The valves' surface opening pressure is 650 psig. Design volume chambers if needed.

In addition, there is a two mile, 4" ID main line with 500 ft, 2" lines to each well. There are 500 ft flowlines from each well head to a central separator (4 ft × 16 ft with 6 ft of liquid). A 1000 ft, 4" suction line leads from the separator to the compressor.

Design the complete system, selecting the compressor and any high-pressure or low-pressure volume chambers that are needed.

(10) A 0.70 sp. gr. gas is used at 800 psig in a gas lift system. At what temperature should this system operate to be free of hydrate formation? If this was a 0.60 sp gr gas, what should the operating temperature be?

(11) A 1000 psi system is operating at 80°F. The pressure drops to 8000 psi across a choke while lifting a well. Are hydrates likely to form? ($\gamma_g = 0.70$)

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3.6 GAS LIFT OPERATION, ANALYSIS, AND TROUBLE SHOOTING

3.61 Introduction

The success or failure of any gas lift installation depends almost entirely upon the field personnel who operate the system. Although gas lift valves have been improved until they are at least partially automatic in

operation, an installation still requires close supervision during a portion of the unloading process and during the period of adjustment until the injection gas has been properly regulated. The proper analysis of a gas lift installation is extremely important in determining whether or not it is performing satisfactorily. In most cases the general opinion and accepted practice has been to wait until some type of trouble develops before analyzing a gas lift installation.

Preferably, each well should be analyzed while it is performing satisfactorily to show whether or not the installation has been properly designed, to provide valuable information for future reference in the event of trouble, and, where improvement appears possible, to indicate design changes which may then be made with complete confidence. If such information is not obtained and the tubing and gas lift valves are pulled because of some trouble, a change of design including such things as valve spacing, valve pressures, and type of gas lift valve would still involve some guesswork.

3.62 Operation of gas lift systems

3.621 Unloading processes

After gas lift valves have been installed in a well, the next problem is the unloading of fluids from the well. This is to enable injection gas to reach the operating gas lift valve without excessive kick-off pressure with consequent final stabilization of the production rate. During the initial installation of gas lift valves the well may or may not be loaded with a "kill" fluid all the way to the surface.

Also, depending upon whether the well is to be placed on intermittent or continuous flow, the well may be unloaded intermittently or continuously. Because a well is going on continuous flow does not preclude its being unloaded intermittently. In most cases a lower pressure traverse can be created in the tubing string by intermittent unloading than by continuous unloading. However, it is usually the case that a well to be placed on continuous flow will be unloaded continuously, while a well to be placed on intermittent flow will be unloaded intermittently.

3.6211 Continuous flow unloading process

Refer to Figure 3.61, which is a typical well to be unloaded continuously. Note that there are four gas lift valves in the string and their respective surface operating pressures are 625, 600, 575, and 550 psi. Assume that the well is initially standing with "kill" fluid all the way to the surface. The following procedure will follow the unloading process (see Fig. 3.61).

(1) Inject gas slowly through a choke to the annulus. Immediately the "kill" fluid will start coming out of the tubing string. A solid stream of fluid will be unloading. The well must be kicked off very slowly so fluids passing through the valves will not damage them and so excess pressure differential will not be placed on the valve's bellows.

(2) As gas is continuously applied to the annulus, a gradual increase in casing pressure is required to keep fluid moving out the tubing string. This fluid is being U-tubed from the annulus to the tubing.

(3) Valve 1 (625 psi) will eventually be uncovered, whereupon gas is injected into the tubing string. This can be noted at the surface by an immediate increase in the velocity of the stream of fluid coming out the tubing string.

(4) A mixture of gas and liquid will now be coming out at the surface and the casing pressure will level off at the 625 psi (surface operating pressure of Valve 1).

(5) Gas injection into the annulus will continue, forcing the liquid level in the annulus to lower until Valve 2 (600 psi) is uncovered. Valve 2 can be

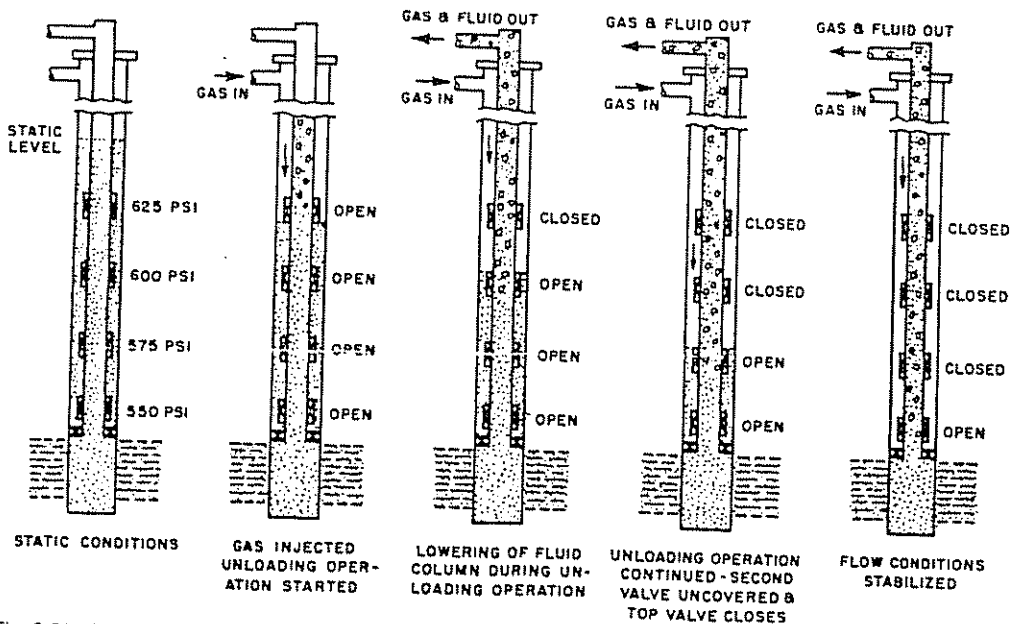


Fig. 3.61 Unloading Operation - Continuous Flow Well

uncovered because the fluid gradient in the tubing string from Valve 1 to the surface has been lightened considerably by the gas.

For example, if the original "kill" fluid had a gradient of 0.50 psi/ft, it may now have been lowered to 0.10 psi/ft in the tubing string by the injection of gas. If the distance to Valve 1 is 1250 ft, the pressure traverse of this 1250 ft has been changed from (1250)(0.50) = 625 psi to (1250) × (0.10) = 125 psi, a difference of (625 − 125) = 500 psi.

The pressure in the tubing string after gas starts passing through Valve 1 is 50 psi (surface pressure) + (1250)(0.10) = 50 + 125 = 175 psi. This leaves (625-175) = 450 psi with which to work down the hole to Valve 2. This also determines the spacing of Valve 2 which is (450/0.50) = 900 ft. Therefore, Valve 2 is located at (1250 + 900) = 2150 ft.

(6) As soon as Valve 2 is uncovered, gas will be injected through it at 2150 ft. Also, the casing pressure will drop to 600 psi since Valve 2 is set 25 psi less than Valve 1. The gradient in the tubing string will now be lowered to 0.10 psi/ft from Valve 2 to the surface, or the pressure in the tubing at Valve 2 will be 50 psi (surface) + (0.10)(2150) = (50 + 215) = 265 psi. This now leaves a difference of (600-265) = 335 psi with which to work down to Valve 3 located at (335/0.50) = 670 + 2150 = 2820 ft

(7) Gas is continuously injected until Valve 3 is reached and the same sequence is repeated until Valve 4 is reached. At some point during the unloading process, the BHP will be lowered to the extent that fluids will start feeding in from the formation. At this point the composition of tubing string fluid will change, becoming a combination of U-tubed annulus fluid and produced formation fluid. This will slow the unloading process until the operating valve is reached.

(8) Once Valve 4 is reached at 3306 ft, casing pressure will stabilize at 550 psi surface operating pressure and the well will be placed on production.

The continuous flow unloading process should be a continuous operation, as the name implies. Valves are spaced to enable the well to unload itself. The gas should be choked in at the surface and the well should unload automatically. If for some reason the well fails to unload, it should be placed on intermittent gas injection and unloaded in the manner described next in the intermittent unloading process.

3.6212 Intermittent flow unloading process

If a well is to be placed on intermittent flow gas lift (say, 50 B/D from 8000 ft), it can be unloaded with fewer valves than a continuous flow installation. Unloading gradient factors can be found in Appendix 3F.1 and 2. The unloading gradient depends on expected production rate and tubing size. Since most intermittent wells will produce 50-200 B/D, an unloading gradient of 0.04 psi/ft will suffice for most of the wells in this category and is generally used as a minimum.

The unloading process for an intermittent well is very similar to the continuous flow unloading process except that the fluid load is kicked out in the form of piston-like slugs of liquid to the surface. For example, noting the four pressured valves used in Figure 3.62, which are set the same as for the continuous flow unloading process, Valve 1 would still be placed at 1250 ft if the well was loaded with "kill" fluid to the surface. The following procedure will follow the intermittent unloading process:

(1) The well may be unloaded exactly the same as for the continuous flow process until Valve 1 is uncovered.

(2) Utilizing a time cycle controller at the surface, the well can then be placed on short time intervals and short injection periods. A good start

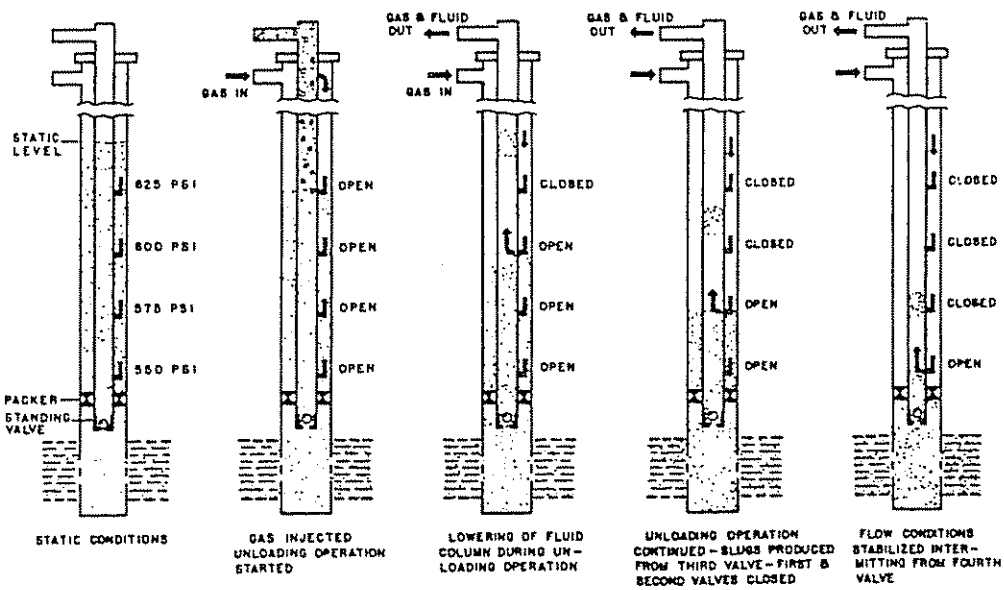


Fig. 3.62 Unloading Operation - Intermittent Flow Well

would be to put the injection gas on a five to ten-minute cycle with a one-minute injection period. This allows ample time between slugs, particularly until a lower depth is reached. When a slug has left the tubing string from Valve 1, the pressure traverse becomes (0.04)(1250) = 50 psi compared to 125 psi for the continuous unloading process. Actually, the unloading gradient can be reduced to less than 0.04 psi/ft, but this is a safe figure to use in the design procedure.

(3) Liquids in the annular space will continue to be U-tubed out similar to the continuous flow unloading process. However, Valve 2 will be deeper. With 50 psi (surface pressure), the total pressure in the tubing at 1250 ft is [50 + (0.04)(1250)] = 100 psi. This leaves (625-100) = 525 psi with which to work down to the next valve. The spacing distance between valves is (525/0.50) = 1050 ft, or Valve 2 will be at 2300 ft compared to 2150 ft for continuous flow unloading.

(4) Once Valve 2 is uncovered, a gradient of 0.04 psi/ft will be created from Valve 2 to the surface and casing pressure will drop to 600 psi. The pressure in the tubing at Valve 2 is now [50 + (0.04)(2300)] = (50 + 92) = 142 psi, leaving (600 - 142) = 458 psi with which to work down to the next valve. The spacing to Valve 3 will be (458/0.50) = 916 ft and will be located at (2300 + 916) = 3216 ft, compared to 2820 ft for continuous flow unloading.

(5) The same sequence of events will occur until Valve 4 is unloaded at 4008 ft. The cycle time will then by adjusted to suit the well's production characteristics. Note that for the same number of gas lift valves a depth of 4008 ft is reached in the intermittent unloading process compared to 3306 ft in the continuous flow unloading process.

There may be instances in the intermittent unloading process when it is difficult to jump from one valve to the next. If this occurs, one remedy is to relieve all surface back pressure and utilize excess gas to unload the well. As greater depths are reached, a good unloading cycle time is (1.5 × depth/1000). This gives the valve time to close between cycles, thereby creating a minimum pressure in the tubing string. This is particularly important if you are striving to reach a lower valve.

3.622 Types of gas injection control

Numerous ways exist of controlling the injection gas to a well. The type of installation (intermittent or continuous flow) influences the type of control to a large degree. There are instances whereby adverse conditions such as freezing have an effect on the type of control utilized. Several types of surface control are discussed here with applications to both continuous and intermittent flow.

3.6221 Choke control

Continuous flow. This is probably the simplest and best method of control for a continuous flow gas lift installation (Fig. 3.63). A choke is installed on the injection gas line and is sized to pass the desired volume of gas into the tubing string. Some trial and error

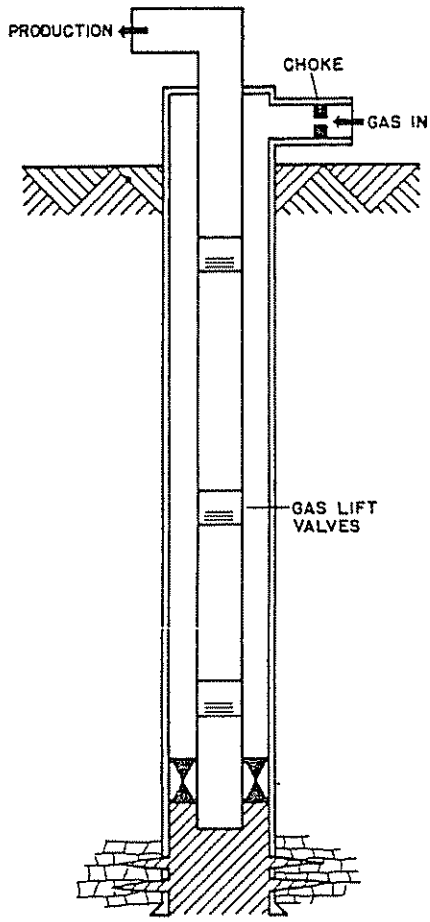


Fig. 3.63 Choke Control of Injection Gas

adjustments may be made to obtain the correct choke size at the surface. An adjustable choke makes it easier to reach the correct size, but it is also more likely to cause freezing problems. A jacket can be built to surround the choke and the well production stream can be routed through this jacket. The warm well fluid usually eliminates the freezing in the choke. A disadvantage to flow through this jacket is the additional surface back pressure it may create due to turns and restrictions. A calibrated needle valve or other valves can be used in lieu of a positive or adjustable choke.

Intermittent flow. Choke control can be used in controlling gas injection to intermittent flow wells utilizing unbalanced gas lift valves with enough spread. This type of control may require much attention before finding the correct surface choke size. The injection gas rate must be timed to coincide with the well feed-in characteristics. For example, it may be desirable to kick a well every thirty minutes with a valve opening pressure of 750 psig and a valve closing pressure of 700 psig. This means that the choke size must produce a 50 psig buildup on the casing (700 to 750 psig) in a 30-minute time interval. Other wells may require this same buildup of casing pressure in 2 hours or some other time interval depending upon the well's characteristics. In most cases this type of control is for wells with a low volume available gas supply, making it desirable to utilize the annulus for storage.

This represents one method of control, but exercise extreme care in selecting a gas lift valve with proper spread characteristics to give the required injection volume. Excessive spread will cause the use of too much gas, whereas too little spread will cause the well to load up. Also, choke control can increase freezing problems. A standard bellows intermittent valve is generally not suitable for this job. A valve with a large spread or marked tubing pressure sensitivity is best suited for this type of operation. Again, some type of calibrated valve may be used for intermittent lift, also.

3.6222 Regulator control in conjunction with a choke intermittent flow

A regulator is used in series with a choke so when casing pressure builds up to the desired pressure before the end of the waiting period necessary for the tubing fluid pressure buildup, the regulator will shut off the gas (Fig. 3.64). This means that the choke sizing does not have to be quite as sensitive. Generally, a choke somewhat larger than normally needed can be installed since the regulator will prevent the choke from allowing too much gas to enter the annular space.

For example, in the previous example needed 50 psig buildup (from 700 to 750 psig) in 30 minutes. If the choke allowed this 50 psig buildup to occur in twenty minutes, the regulator would close off the injection gas on a high pressure shutoff at 750 psig. This may be called choke control with casing pressure shutoff. The gas lift valve will not open until the fluid load in the tubing builds up to trip it. If casing pressure buildup occurred in 20 minutes, the cycles would still be 30 minutes long due to the valve characteristics and liquid feed-in. However, a setting of this nature would again require strict attention.

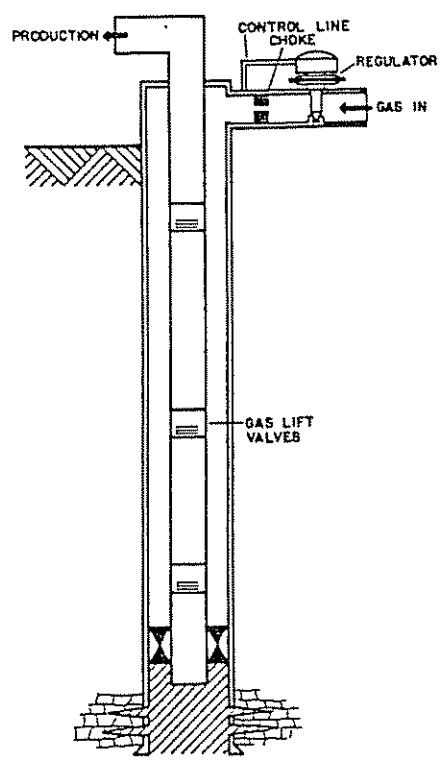


Fig. 3.64-A Regulator in Series with a Choke

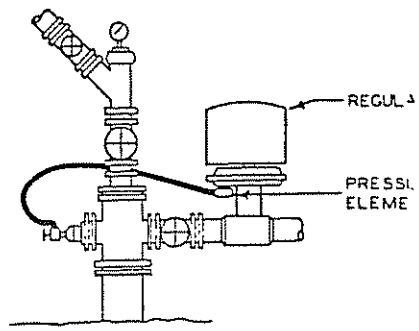


Fig. 3.64-B Casing Control

Another use for this type of control is in a well where the necessary gas volume has not been allowed entry through the valve spread. For example, a valve spread of 50 psig may not allow sufficient gas entry to lift the liquid slug. The choke at the surface can be sized up to half the size of the gas lift valve port. When the gas lift valve opens and the casing pressure drops, the regulator immediately opens, allowing the entry of the additional required gas to lift the slug.

Since the gas lift valve port is larger than the choke, the casing pressure will still drop, allowing the gas lift valve to close. As soon as the valve closes the regulator will continue to pass gas until the predetermined surface opening pressure is reached. The well is then primed, ready for the next fluid load to open the valve.

Continuous flow. A regulator can be used in series for continuous flow with the control line connected to the tubing wellhead (Fig. 3.65). This will shut off the gas completely for predetermined high tubing pressures. For wells that have a tendency to flow naturally, this represents an excellent means of control. Many wells after being "kicked-off" will flow naturally, and this is a means of assuring that gas is not used while the well flows naturally. A regulator can be used in parallel with a choke to take care of fluctuating gas injection pressures (Fig. 3.66). For example, to maintain 700 psig on the casing, an $^{6}/_{64}$" choke might be installed with 800 psig upstream. There might be times when the upstream line pressure drops to 750 psig and the casing pressure begins to drop below 700 psig. A bypass regulator would then open to maintain the 700 psig even though the upstream line pressure had dropped.

3.6223 Time cycle controller

Continuous flow. This is not a recommended type of control for continuous flow except where freezing may be prevalent if a choke is used. By placing a time cycle controller on short fast cycles, the required amount of gas is injected through a large seat and freezing is reduced.

Intermittent flow. A time cycle controller or intermitter is the most common means of injection gas control for intermittent flow and is quite easily adjusted. A controller at the surface can be adjusted to inject gas for

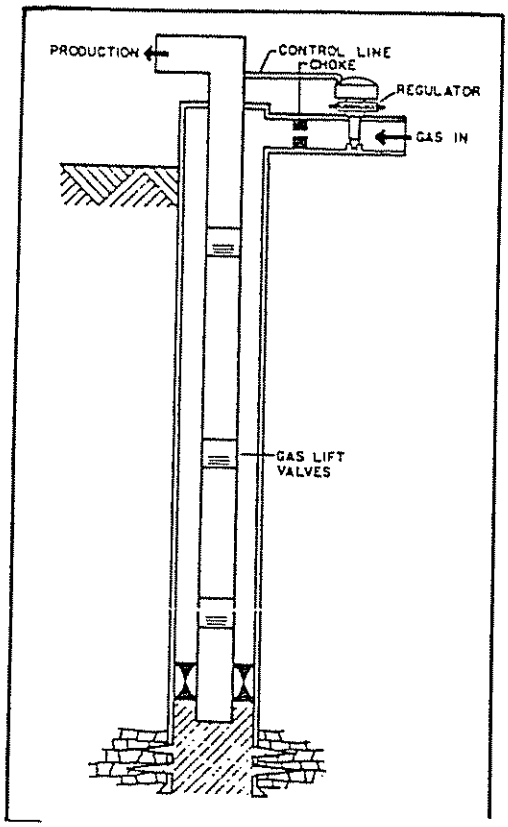


Fig. 3.65-A Tubing Control

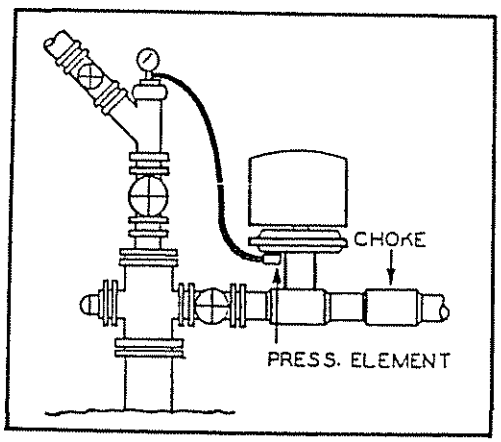


Fig. 3.65-B Tubing Control

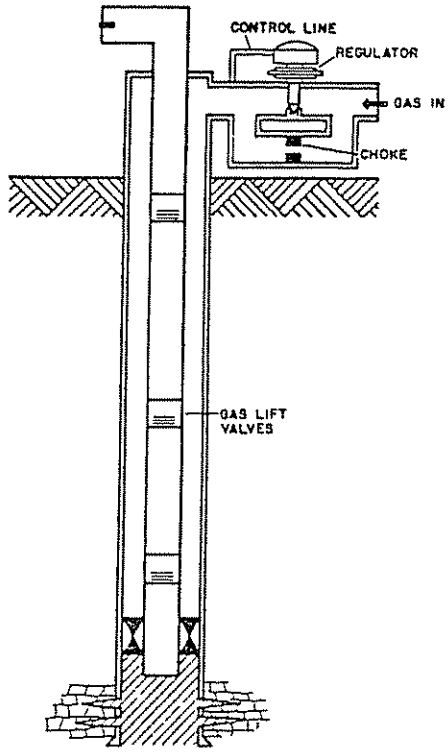


Fig. 3.66 Regulator in Parallel with a Choke

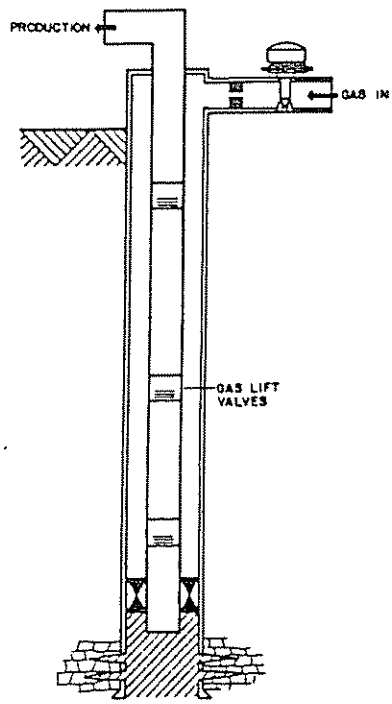


Fig. 3.67 Intermittent Control for Intermitting Lift

almost any length of time and for almost any time interval. This is done by adding or removing pins and/or changing clock revolutions. For example, a typical setting would be to inject gas for 1 minute out of every 30 minutes, or for 2 minutes every 4 hours, etc.

Figure 3.67 shows a typical controller at the surface for injecting gas into the well; Figure 3.68 shows a standard clock-wind controller; Figure 3.69 shows a particular controller which offers automatic operation without having to wind a clock.

3.63 Analysis and trouble shooting

3.631 Introduction

There are many ways to analyze both continuous and intermittent flow installations. These can be divided

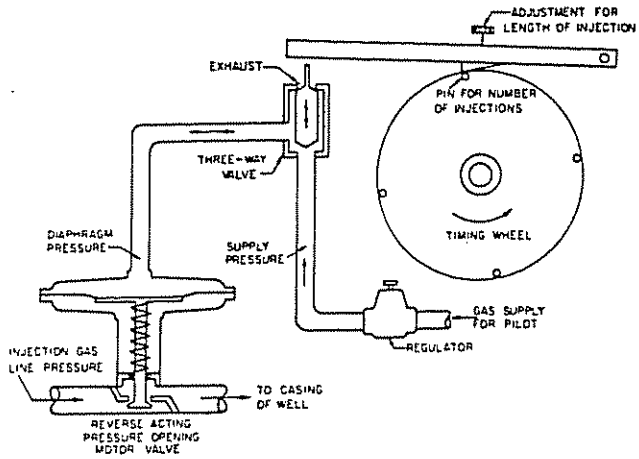


Fig. 3.68 Time-Cycle Controller for Intermittent Gas Lift Installation (Courtesy API[1])

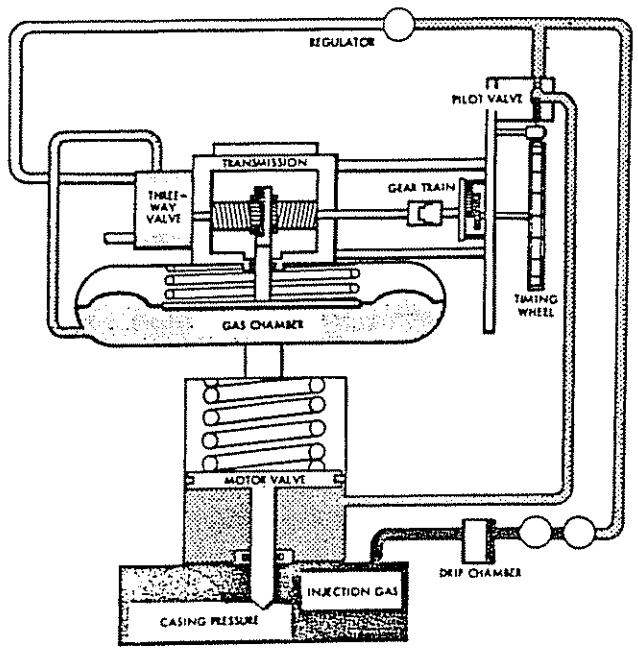


Fig. 3.69 Automatic Wind Controller

into detailed methods and observation methods. The following can serve as a check list of questions to ask and check prior to a commitment to conduct downhole analysis such as flowing pressure and temperature surveys. Although some of these may seem trivial, all have occurred in practice.

Checklist of questions

(1) Is the wing valve open?
(2) Is valve on gas line open?
(3) Is there a choke in the flowline?
(4) Is the master valve open?
(5) Is the well producing emulsions?
(6) Is the gas injection input choke freezing up?
(7) What is the injection gas pressure?
(8) What is the wellhead pressure?
(9) What is the separator pressure?
(10) What is the flowing wellhead temperature?
(11) What tubing size is in the well?
(12) What is the flowline size?
(13) What is the flowline length?
(14) Are there numerous elbows in flow path at the wellhead?
(15) Is the choke body still in the wellhead?
(16) Are other restrictions still in the surface wellhead?
(17) What is the producing rate of oil, water, gas?
(18) What is the gas injection rate?
(19) Is the well on intermittent or continuous flow?
(20) Do we have a 2-pen pressure chart on the well showing casing and tubing pressure?
(21) Do we have a copy of the valve installation record to compare with records of what was supposed to be installed?
(22) Does the surface flow line feel hot, cool, warm, etc.?
(23) Does the well have a history of making paraffin, forming scale, severe corrosion, etc.?

In continuous-flow gas-lift installations, there is a tendency to over-inject gas when a well is not making its required production and to leave the installation alone when the well flow appears reasonable. Considerable improvement can be made in the rate of production and in the injection gas usage by a thorough analysis of the well's performance.

The following list summarizes the various measurements and procedures used to analyze a continuous flow gas-lift installation.

(1) Recording surface pressures on tubing and casing (two and three pen recorders)
(2) Measurement of gas volumes
(3) Surface temperature readings
(4) Visual observation of surface installation
(5) Gauging well for oil and water production
(6) Subsurface pressure surveys
(7) Subsurface temperature surveys
(8) Combination flowing temperature and pressure surveys
(9) Fluid level determination by acoustic methods

Intermittent gas-lift performance should also be checked before making equipment changes. Often, changes in the surface regulation of the injection gas may be all that is required to improve gas-lift efficiency. At any rate, a careful analysis of the well's performance will provide the data on which to base recommendations for changes in design that may have to be made. Many times, the analysis may show that an installation cannot be improved by change of design and such a conclusion prior to making expensive renovations can be valuable in extracting the maximum profit from a gas-lift operation.

A good indication of intermittent gas-lift performance can be obtained from surface data. Both the surface casing and the surface tubing pressures of a gas-lift installation are important for indicating the performance of the operation. A multiple pen pressure recorder is a necessity for recording casing and tubing pressure measurements during all phases of the lift cycle.

3.632 Pressure surveys—continuous flow

3.6321 Introduction

Subsurface pressure surveys are the best and most widely-used method of properly analyzing gas-lift

installations. Static pressure surveys will determine static fluid level, static gradient, and static bottom hole pressure. A flowing pressure survey will locate the point of gas injection, leaks in the tubing, valve failures or multi-point injection. A flowing pressure survey will determine the flowing gradient above and below the point of injection, and the flowing bottom hole pressure. Accurate well gauging used along with pressure surveys can determine the inflow capability of the well.

A common fallacy in running flowing pressure surveys is to wait until some trouble develops. Admittedly, the trouble can probably be located; but vital information on how to improve the installation will not be obtained. The importance of the correct spacing of gas-lift valves on high PI continuous flow wells cannot be over-emphasized. Therefore, a pressure survey should be run while the well appears to be performing satisfactorily. This type of information will in turn allow a correct respacing of gas-lift valves.

A flowing survey would give the exact operating valve, the working fluid level, the flowing BHP, the flowing gradient above the point of gas injection, and, of course, static BHP if the bomb remains on the bottom long enough for pressure buildup. Given several surveys on a well that is performing satisfactorily, curves can be drawn showing decline trends and changes which will prove valuable at a later date and will facilitate accurate respacing if tubing must be pulled for any reason.

There is no difficulty in running a bomb in a well making 1500-2500 B/D in 2" or 2½" tubing if the smallest diameter pressure bomb available is used and if the well is flowing smoothly without surges. If some surges are noted, however, it would be advisable to add a suitable weighted section to the bomb. The following procedure would serve as a guide in running a pressure survey and was given by Brown.

(1) Before running the survey, have a temporary valve placed above the normal flow valve as shown in Figure 3.610. This ensures that the well will be flowing in its completely normal and stabilized condition before the survey starts and ensures that this stabilized flow rate need not be interrupted before starting the survey.
(2) Start the survey by leaving the bomb in the lubricator long enough to obtain a flowing tubing pressure. If the bomb is one of several types whose initial tension must be exceeded before the pressure recordings begin and if the tubing pressure is below this point, a recording should be made of the tubing and casing pressure on a recording instrument during the entire survey. Actually, this recording should be made anyway, so a record of pressures will be obtained during the survey.
(3) Allow the bomb to enter the tubing slowly, keeping a hand on the wire line so if it slackens it may be immediately started back out of the hole to prevent kinking of the wire line. If the bomb goes freely in the first 100 ft, no further trouble should be encountered due to flow because this is where the maximum fluid velocities occur. However, if the velocities are so great that difficulty is encountered, the flowline valve may be closed, the bomb run to bottom as fast as possible, and then, after reopening the well, the survey may be started from the bottom. In this case, flow has been interrupted so the bomb should remain on bottom long enough to reestablish stabilized flow conditions before commencing the upward trip.
(4) In running the survey, make a stop directly below each gas lift valve. If this is a trouble-shooting survey and a leak is being pinpointed, a stop between valves may be made, also. If the well is not equipped with valves all the way to bottom, make several stops below the last valve to establish the flowing gradient of the fluid below the point of gas injection.
(5) If no trouble is encountered on running in, the same stops may be made on the way out, providing a double check is desired.
(6) A check on total fluid production, along with a measurement of the input and output gas liquid ratios and a recording of casing and tubing pressures, should be made simultaneously with the pressure survey because, even with a small diameter instrument, there may be a reduction of flow rates during the survey.

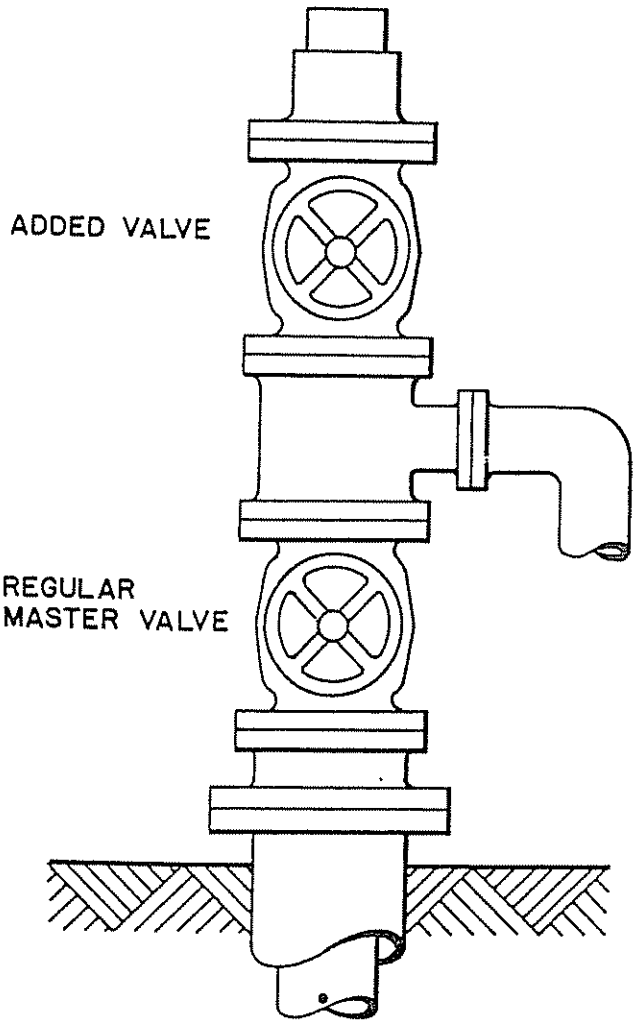


Fig. 3.610 Tree Arrangement to Allow Running of Pressure Bomb without Interrupting Flow

(7) Plot pressure curves on suitable graph paper; a complete analysis of the well has been made.

3.6322 Hypothetical case of flowing pressures surveys

To best illustrate the use of flowing pressure surveys, several hypothetical examples will be given followed by numerous field cases. Refer to Figures 3.611A through 3.611G for these illustrations.

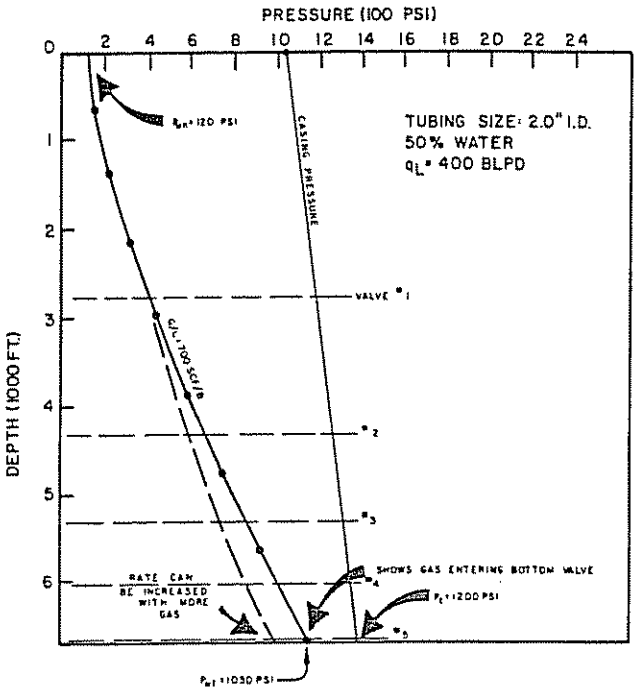


Fig. 3.611-A Flowing Pressure Survey (For Constant P_{wh}, Rate Can Be Increased with More Gas. However, for Long Flowline More Gas May Not Increase the Rate Due to Increase in P_{wh})

Figure 3.611A:

This shows a case where gas is entering through the lowermost valve at the bottom of the well. This is made possible by having more operating casing injection pressure at bottom than pressure in the tubing. Also, there is no abrupt change in flowing pressure gradient from the bottom to the top of the well. This well might be improved by adding additional gas, if available, to obtain the flowing gradient as shown by the dashed line. This assumes that a constant wellhead pressure will be maintained.

Figure 3.611B:

This flowing pressure traverse shows that gas is entering through Valve 1 because an abrupt change in flowing gradient occurs at the depth of Valve 1. Closer spacing between Valves 1 and 2 would have permitted reaching Valve 2 and would have increased production.

Figure 3.611C:

Operation is from Valve 2 at 3600 ft. The balance point between casing pressure and tubing pressure is at 4300 ft. Therefore, Valve 3 cannot be reached at 4500 ft. This indicates that spacing is too wide between Valves 2 and 3. Once Valve 3 has been reached, it may

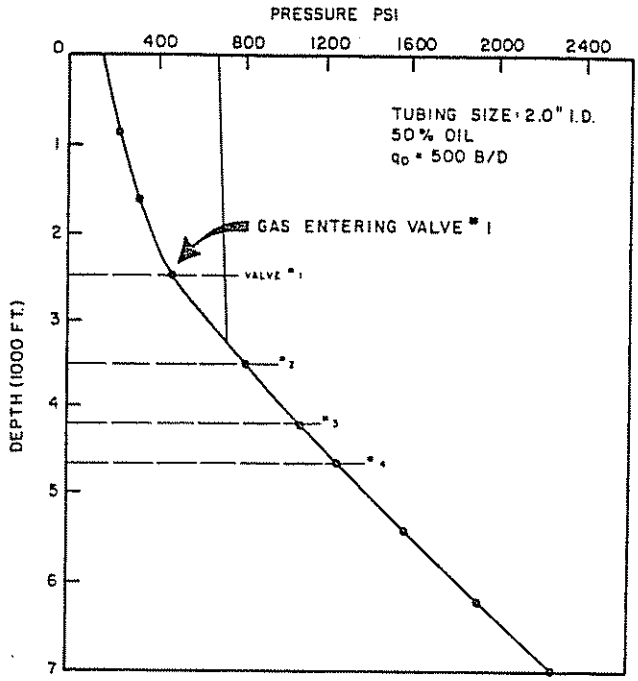


Fig. 3.611-B Flowing Pressure Survey (Operation from Top Valve)

be possible to remain there even after returning the wellhead pressure to its normal operating pressure. If the well is pulled for any reason at a later date, closer spacing will improve the production rate from this high PI well.

Figure 3.611D:

This survey shows Valve 3 should be open but for some reason it is not passing gas. It is probably plugged, but could also have gained pressure or possibly failed in a closed position. Valve 2 is presently the

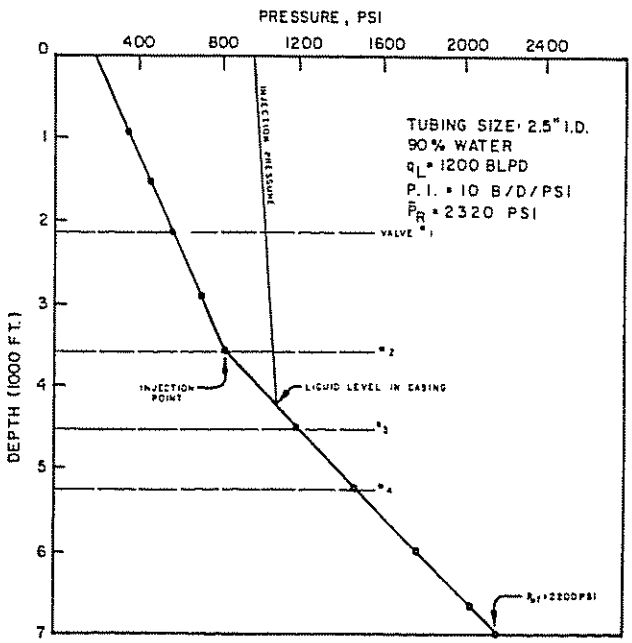


Fig. 3.611-C Spacing Too Far Apart (Rate Can Be Greatly Increased by Closer Spacing on High PI Wells)

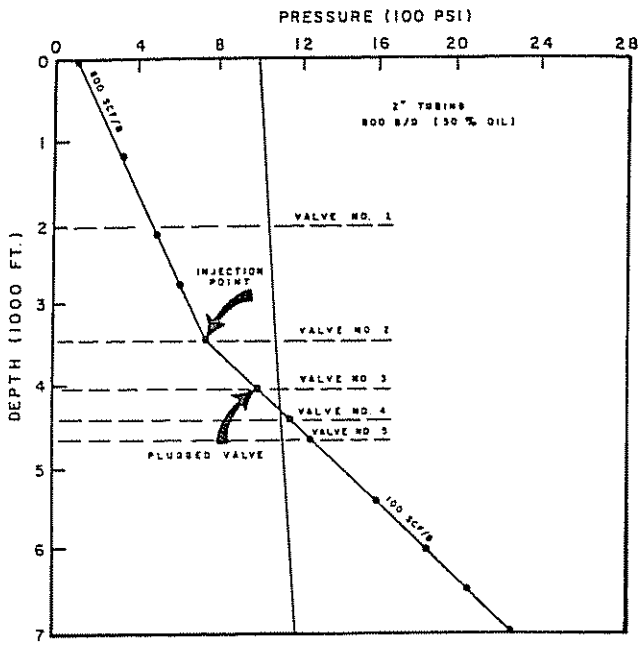


Fig. 3.611-D Flowing Pressure Survey (Gas Should be Entering Valve No. 3—Check Valve No. 3)

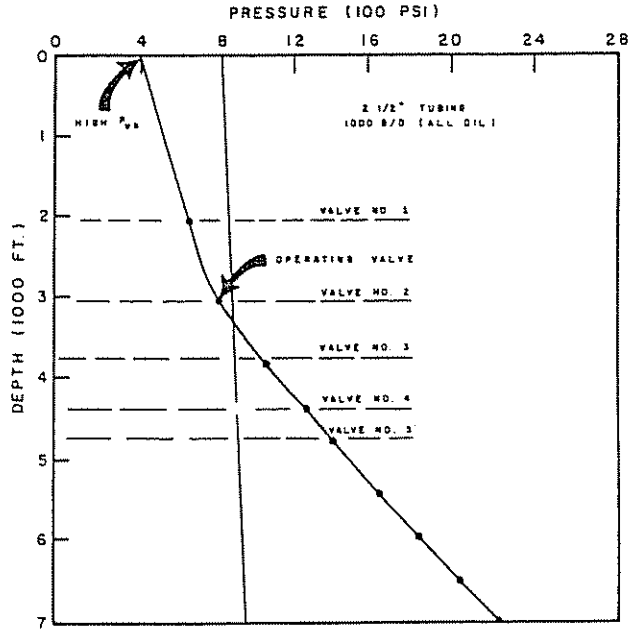


Fig. 3.611-F Flowing Pressure Survey (High Surface Tubing Pressure Prevents Operating from Lower Valves, and Increasing Rate)

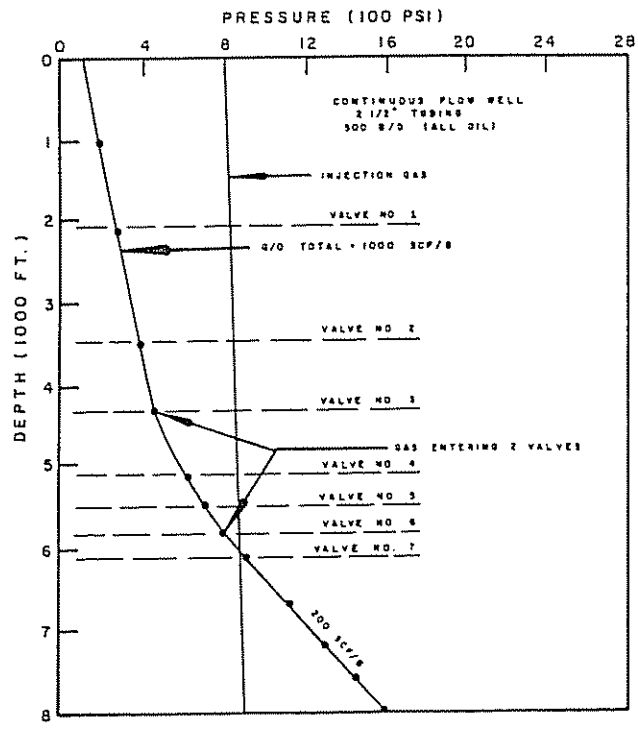


Fig. 3.611-E Flowing Pressure Survey (Gas Entering at Valves No. 3 and No. 6. Valve No. 3 Should Be Pulled)

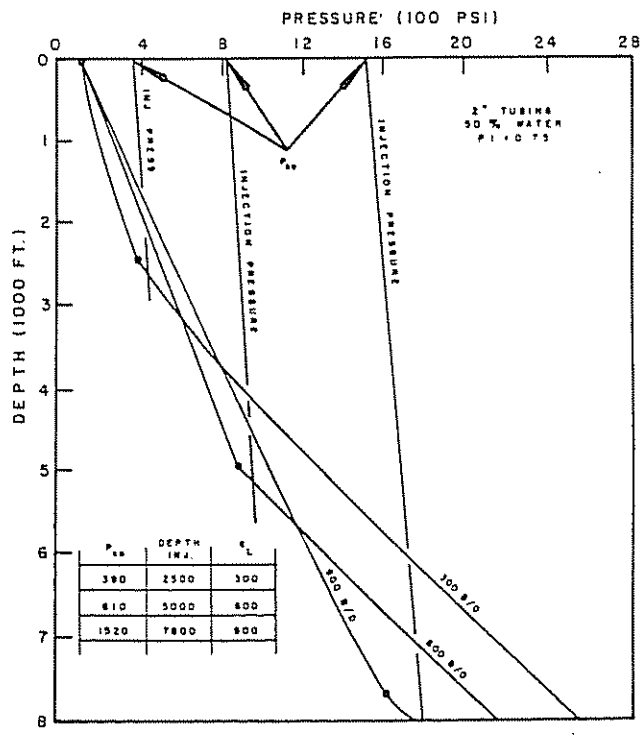


Fig. 3.611-G Effect of Surface Injection Gas Pressure

operating valve. It might be possible to reach Valve 4 by temporarily lowering the tubing wellhead pressure or increasing the injection gas pressure or both. Once Valve 4 has been reached, operations can be maintained from that point.

Figure 3.611E:

This survey shows gas entering through more than one gas lift valve. This is evidenced by the change in gradient occurring at Valve 3 and Valve 6. Although the operation is not too bad, it would be more efficient if all the gas were entering the lowermost valve possible. If Valve 3 could be repaired, it is likely Valve 7 would be reached and increase the production and decrease the injection gas-liquid ratio.

Figure 3.611F:
This pressure survey confirms that the operating valve is fairly shallow (Valve 2 at 3000 ft) due to the high wellhead pressure giving only 400 psi "ΔP" between surface casing and tubing pressure. The operation appears to be performing satisfactorily under present conditions, but the high wellhead pressure should be investigated.
Figure 3.611G:
This figure shows the effect of injection surface operating gas pressure. The effect of more operating pressure becomes more pronounced for higher productivity wells. Even in this case for a PI = 0.75, the rate increases from 600 to 900 B/D for a change in operating pressure from 810 to 1520 psi. There are many installations whereby an increase in production rate can only be achieved by increasing the injection pressure.

3.6323 Field cases of flowing pressure surveys

Several field cases of flowing pressure surveys will be illustrated in this section and discussions as to possible improvement will be made. Figures 3.612A through 3.612H show typical flowing pressure surveys.
Figure 3.612A:
Refer to Figure 3.612A which shows a well making 1000 B/D of oil and water (90% water). From all surface indications the well was performing satisfactorily. However, from the flowing pressure survey it was immediately evident that a change in valve spacing should greatly increase total fluid production. Note that the fluid level in the casing lacks only a few feet of uncovering the last valve with the available line pressure. Since this well had a PI of 10 or greater, the

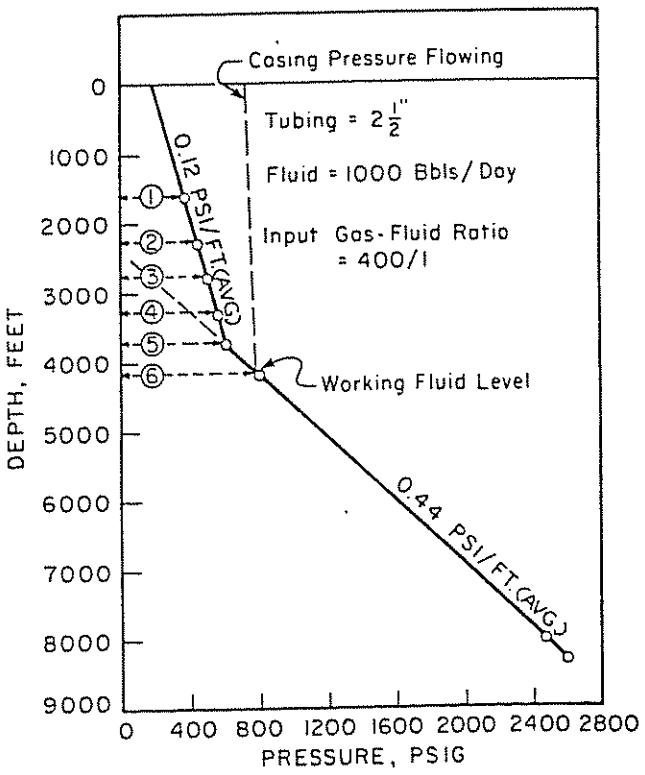


Fig. 3.612-A Flowing Pressure Survey

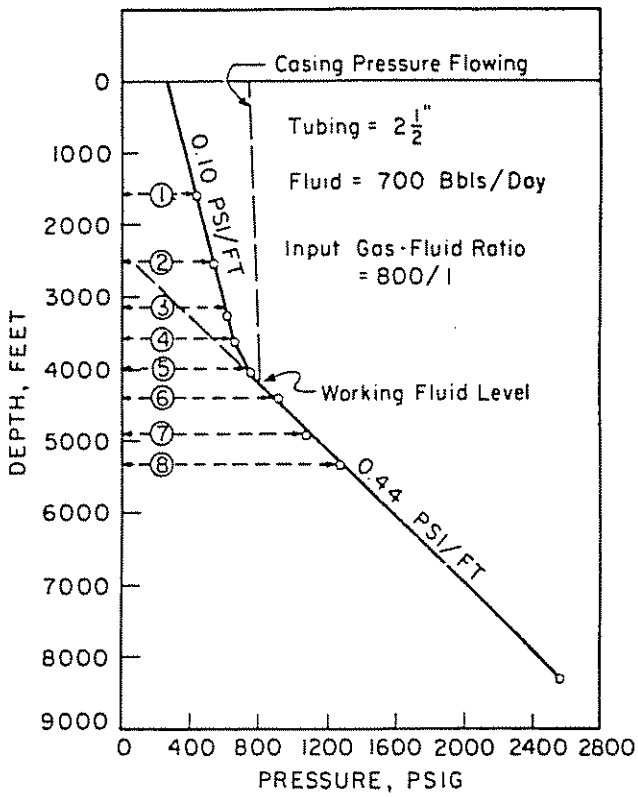


Fig. 3.612-B Flowing Pressure Survey (after Brown, et al,[1] Courtesy West Texas Oil Lifting Short Course)

valves were re-spaced by placing the last valve at a position whereby it would be operated (approximately 60 feet back up the string). This chart (Fig. 3.612A) indicates the possibility of injecting more gas to lighten the gradient such that the lowermost valve could be uncovered. No additional gas was available. Another alternative would be to increase the injection pressure and/or lower the tubing pressure. None of these alternatives were possible, thereby presenting the necessity of re-spacing the valves.

By checking the static fluid level, it was possible to relocate the uppermost two valves as the last two valves in the string and to space them three and two tubing joints apart, respectively. The production rate on this well was increased to 1800 B/D. Since this was an active water drive with the field showing very little drop in pressure with time, the spacing was satisfactory for 1½-2 years.

It is unusual to space valves as close as two or three tubing joints. However, for high productivity water drive wells, being able to inject gas 60-90 ft lower can mean a tremendous increase in production. For example, if the PI is 10 and gas is injected 90 feet lower, changing the gradient from 0.45 psi/ft to 0.15 psi/ft, it changes the flowing BHP by (0.45 − 0.15)(90) = 27 psi. For a PI of 10 this is (10)(27) = 270 B/D.
Figure 3.612B:
Figure 3.612B shows a well where two or three gas lift valves are admitting gas having valve interference. Comparing the actual flowing gradient to what the flowing gradient above the point of gas injection should be indicates too much gas is being injected. This was

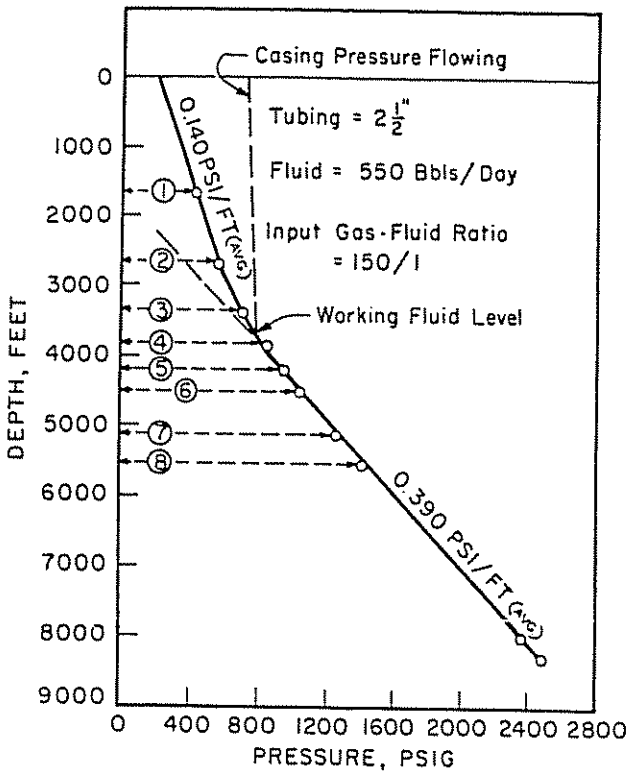


Fig. 3.612-C Flowing Pressure Survey

also confirmed by a measurement of the input injection GLR of 800/1 compared to 400/1 for the well of Figure 3.612A. There seems to be no need for respacing valves on this well, but a need for repairing Valves 3 and 4 is evident. Valves 6, 7 and 8 could be grouped closer to the annulus working fluid level (point to which the available casing pressure depresses the fluid in the annulus).

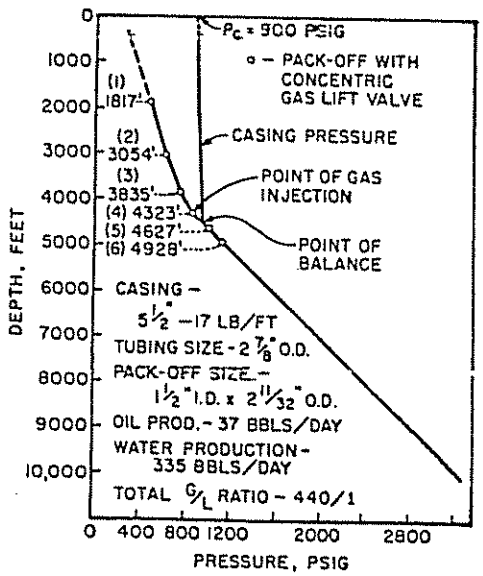


Fig. 3.612-D Flowing Pressure Survey for a Southern Louisiana Well (after Brown, et al,[1] Courtesy West Texas Oil Lifting Short Course)

Figure 3.612C:

Figure 3.612C shows a gas lift installation where too many valves have been run in the well. Four valves would be enough to take care of this well under present conditions. However, if this were a well in which water percentages were expected to increase considerably thereby resulting in a lower annulus working fluid level, the use of lower valves would be justified. An input GLR of 150/1 indicates a very efficient installation.

A closer look at Figures 3.612B and 3.612C reveals valves being spaced closer together until approximately 4000 ft, where the spacing again increases. As noted in Figure 3.612C, there is a greater distance between Valves 6 and 7 than between 5 and 6. The reasoning behind this spacing was a switch from continuous flow valves to intermittent valves. At approximately 4000 ft, the depth to which good continuous flow operations would continue to function had been reached. Continued operation revealed very little drop in BHP with an increase in water production, so the lowermost valves were eventually brought in closer. An original lack of good productivity information on these wells resulted in this type of spacing.

Figure 3.612D:

Figure 3.612D shows a flowing pressure survey as run on a continuous flow well with pack-off gas lift valves. These were 1½" ID by 2 11/32 in. OD pack-off assemblies.

Valve 5 is almost uncovered by gas. As a result of this survey, the casing pressure was increased slightly while the tubing pressure was lowered. The production immediately increased from 372 to 650 B/D. This indi-

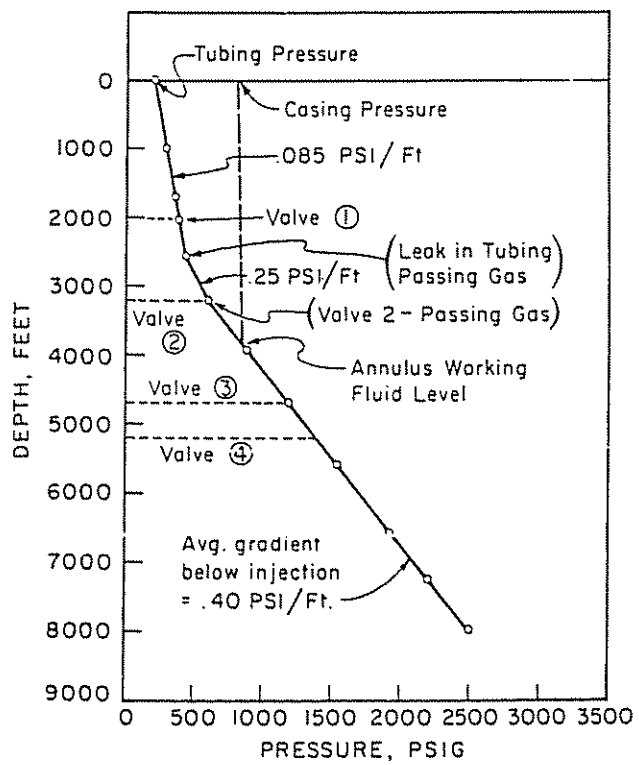


Fig. 3.612-E Tubing Leak Located by Flowing Pressure Survey (after Brown, et al,[1] Courtesy West Texas Oil Lifting Short Course)

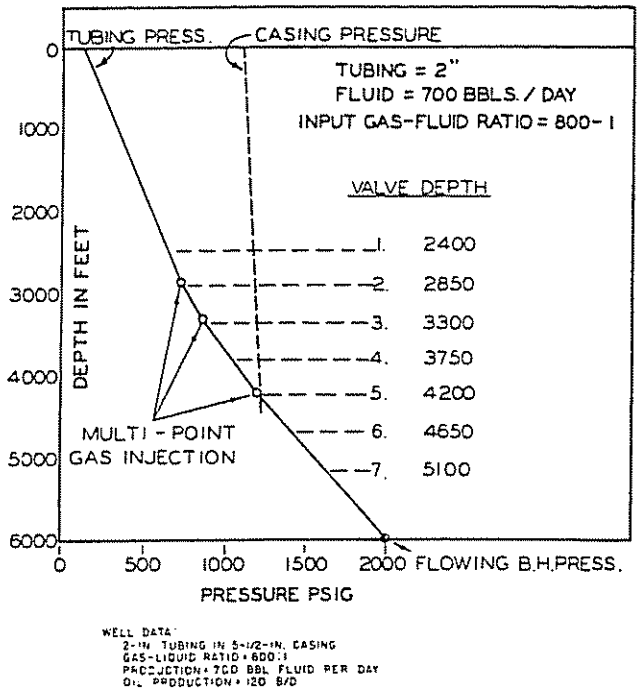


Fig. 3.612-F Tubing Leaks Located by Flowing Pressure Survey

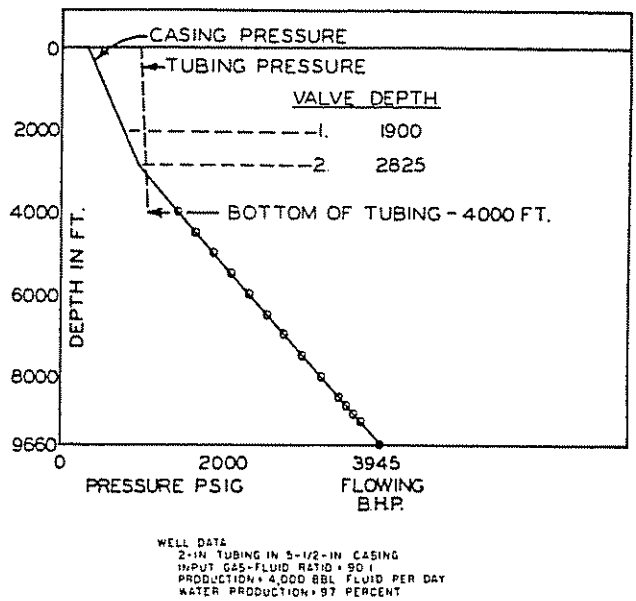


Fig. 3.612-H Pressure Survey of Casing Flow Gas Lift Well

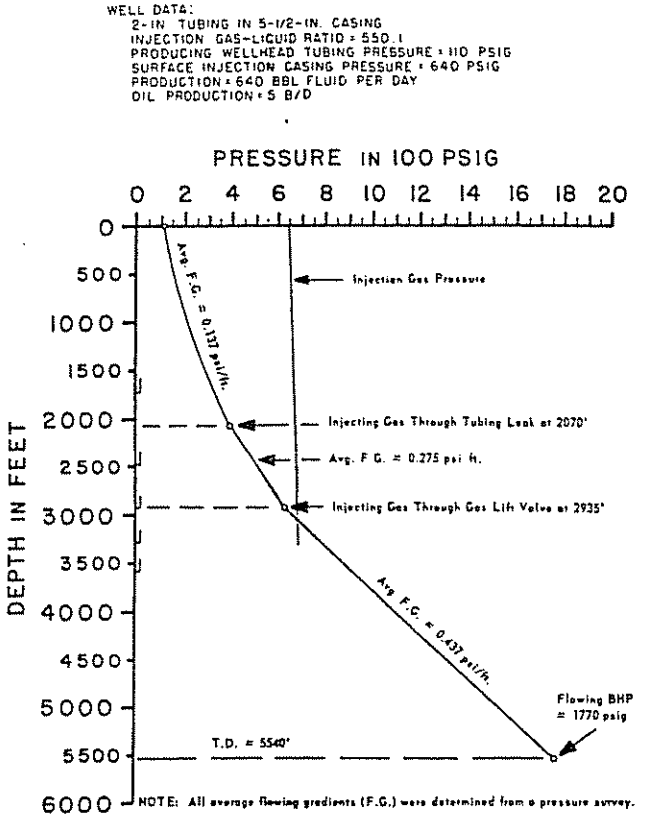


Fig. 3.612-G Locating a Tubing Leak from a Pressure Survey in a Continuous Flow Gas Lift Well

cated that Valve 5 and possibly Valve 6 were then working.

Figure 3.612E:

In running flowing pressure surveys in continuous flow wells, it is sometimes found that a leak in the tubing string exists rather than a leaking or faulty gas lift valve. This is noted by an abrupt change in the gradient at the hole in the tubing as in Figure 3.612E. Of course, leaking gas lift valves can be detected in the same manner.

Figure 3.612F:

This figure also shows tubing leaks immediately above valves 2 and 3, which could possibly be leaking.

Figure 3.612G:

Figure 3.612G is another example of locating a tubing leak between valves 1 and 2. The leak is at 2070 ft.

Figure 3.612H:

Figure 3.612H shows a flowing pressure survey for an annular flow well. The survey has been conducted in the tubing string (run open-ended to 4000 ft in a 9660 ft well). The flowing pressure survey is obtained in the casing only from 4000 ft to total depth. The fluid level is obtained in the tubing and the approximate depth of injection can be obtained.

3.633 Flowing temperature surveys—continuous flow wells

Normally, flowing temperature surveys are run in conjunction with flowing pressure surveys with those few exceptions where only the temperature recording gage is available or the pressure gage fails to show reliable results. In most cases, the operating gas lift valve can be located with a good temperature survey as well as leaking valves or holes in the tubing. The cooling effect of the expanding gas can be detected fairly easily except for very high flow rate wells.

For wells flowing in excess of 6000 B/D in 4" tubing, detecting any cooling effect due to gas expansion

becomes very difficult. Stops should be made above and below each gas lift valve as well as between valves to make certain that a good temperature traverse can be established. Figures 3.613A through 3.613D show typical temperature surveys in continuous flow wells as given by Chirinos.[2]

Figure 3.613A:

Figure 3.613A shows a flowing temperature survey with numerous stops. Gas expansion and the resultant cooling effect show Valve 3 is the operating valve. However, Valve 2 is passing some gas. As a matter of interest, the flowing pressure survey failed to show Valve 2 as passing any gas. Another important factor is noted at the surface. The temperature in the top (lubricator) is 90°F. However, some 50 ft below the surface the temperature is about 117°F. It would be a serious error to assume a straight line temperature gradient between 90°F. at the surface and the bottom hole temperature of 156°F. Therefore, in conducting temperature surveys, one stop should always be made some 30 to 100 ft below the surface, perhaps deeper in off-shore wells standing in water.

Figure 3.613B:

This survey clearly shows that all injection gas is passing through Valve 3. The small ΔT indicates a very small amount of gas passing; this is verified by an injection G/L of 100 scf/B.

Figure 3.613C:

Figure 3.613C shows a flowing temperature survey on a fairly high volume well (4275 B/D in 4" tubing). In this case selecting the operating gas lift valve is difficult. As the rates become higher, there is less cooling effect of the expanding gas; for very high rates it is difficult to select the operating valve with a temperature survey only.

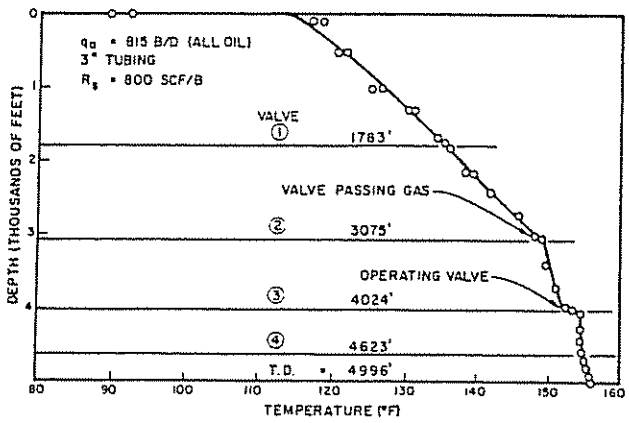


Fig. 3.613-A Flowing Temperature Survey

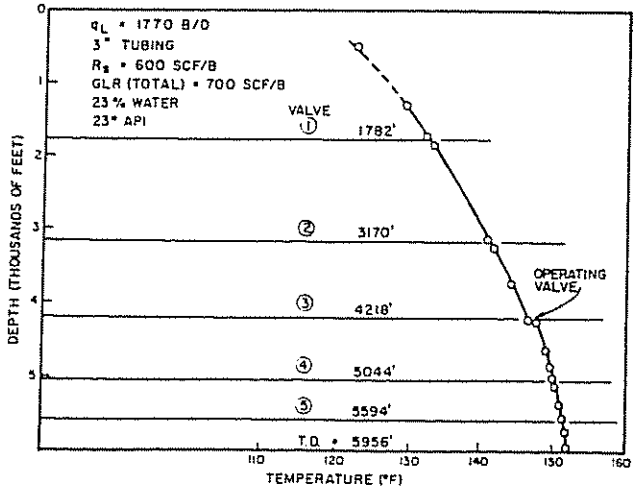


Fig. 3.613-B Flowing Temperature Survey

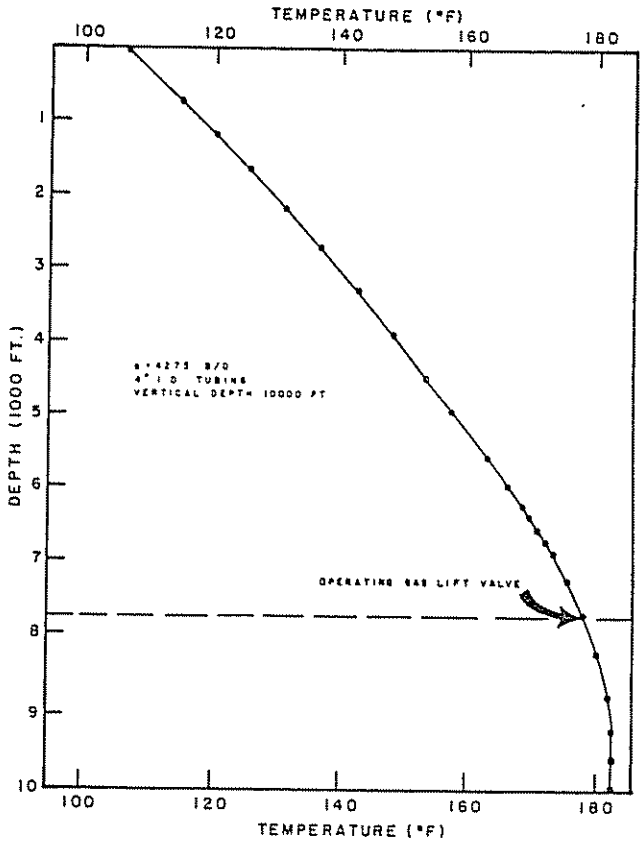


Fig. 3.613-C Flowing Temperature Survey

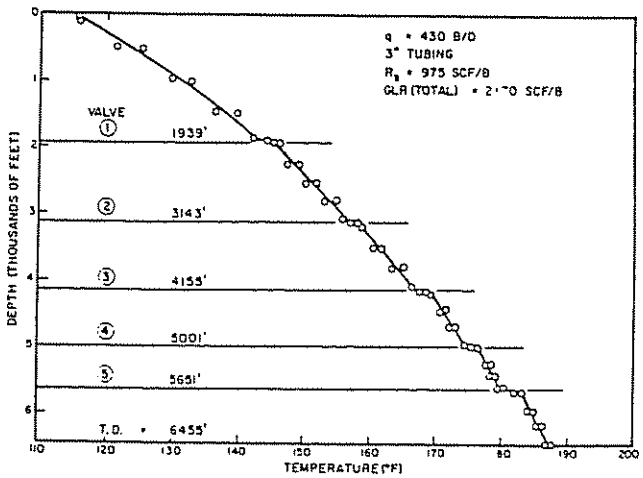


Fig. 3.613-D Flowing Temperature Survey

Figure 3.613D:

Figure 3.613D shows a well making 430 B/D in 3" tubing; all five gas lift valves appear to be passing some gas. The ΔT at Valve 5 shows it is passing most of the gas. The corresponding flowing pressure survey showed only Valve 5 to be passing gas, pointing out the advantage of running both a pressure and temperature survey at the same time.

3.634 Combination flowing pressure and flowing temperature surveys (for continuous flow wells)

It is more beneficial to run both a flowing temperature and a flowing pressure survey in the well at the same time. There are numerous cases where one gives more information than the other. This requires a slightly longer combination recorder gage unit and perhaps a longer lubricator at the surface. However, the same running procedure holds for the combination gage. Typical recordings are shown in Figures 3.614A through 3.614P.

Figure 3.614A:

Figure 3.614A shows a combination survey whereby most of the gas appears to be entering Valve 1. This information is obtained from the temperature survey only, since the pressure survey shows only a slight break at this depth. The pressure survey shows that gas could be entering Valve 3 at 5700 ft. The well could be improved by pulling Valve 1 and repairing it.

Figure 3.614B:

This combination survey must rely on the pressure survey to show Valve 3 to be the operating valve as shown by a change in the flowing pressure traverse. However, the temperature survey shows Valve 2 is passing some gas which is not detectable by the pressure survey. Again, this illustrates the importance of both surveys.

Figure 3.614C:

This combination survey shows Valve 4 is the operating gas lift valve as noted from the pressure survey only. Valve 5 is practically uncovered by gas and should be the operating valve. A temporary lowering of the wellhead pressure or raising of the casing pressure might start gas through Valve 5. Once gas is passing through the valve it is possible to easily maintain operations from that point after, restoring the well to its original conditions.

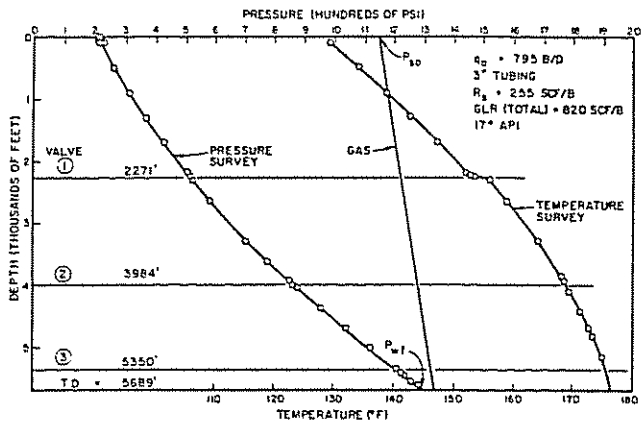


Fig. 3.614-A Combination Flowing Pressure Temperature Survey

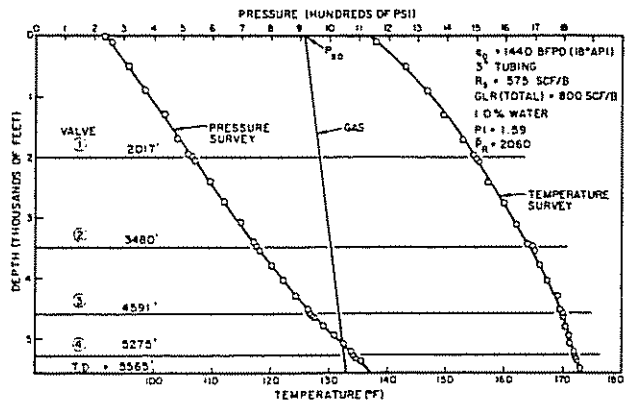


Fig. 3.614-B Combination Flowing Pressure Flowing Temperature Survey

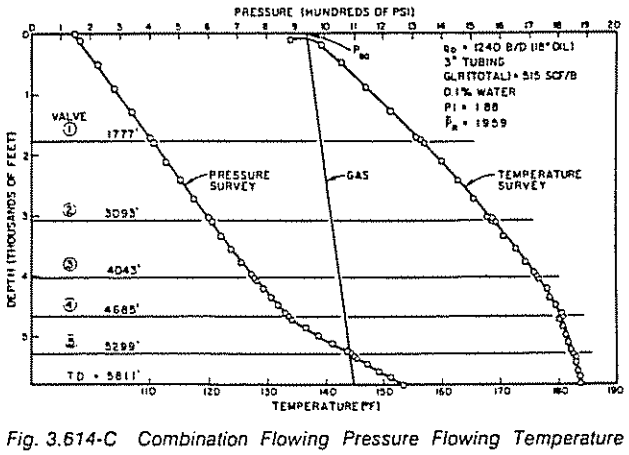


Fig. 3.614-C Combination Flowing Pressure Flowing Temperature Survey

Figure 3.614D:

Figure 3.614D shows a combination survey in a directional well whereby the operating valve is easily detected by both the flowing pressure and flowing temperature survey as being Valve 7.

Figure 3.614E:

Figure 3.614E is another combination survey showing Valve 7 to be the operating valve. This is quite distinctly shown by both the pressure and temperature survey.

Figure 3.614F:

Figure 3.614F shows a combination survey where the flow rate is sufficiently high that the change in either temperature or pressure at the operating valve makes it difficult to locate. The pressure survey appears to be the most informative in this case, although there is some change in temperature gradient also but no abrupt drop in temperature across the valve.

Figure 3.614G:

This is a good example of all the gas entering the bottom Valve 4 as shown by a decided change in temperature and confirmed by the pressure survey. No other valves are passing gas. Note the difference in temperature with the instrument in the lubricator as compared to being just below ground surface. A difference of 13°F. exists in this instance.

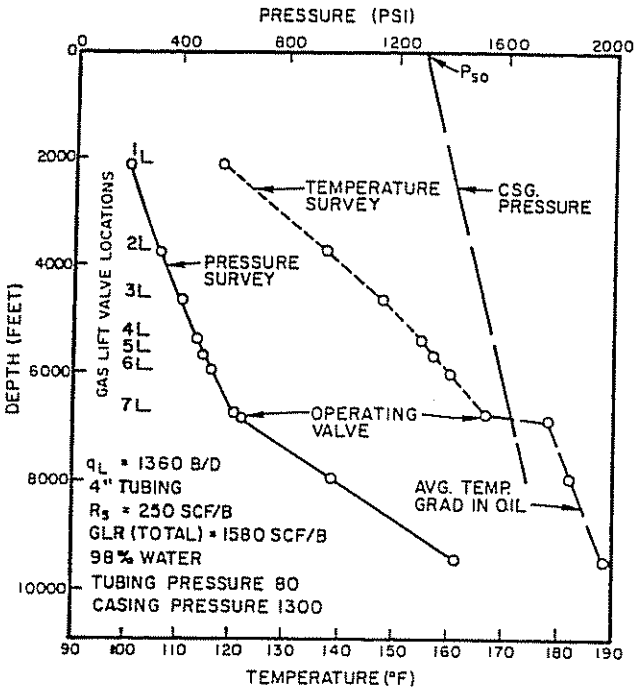


Fig. 3.614-D Combination Flowing Pressure Flowing Temperature Survey

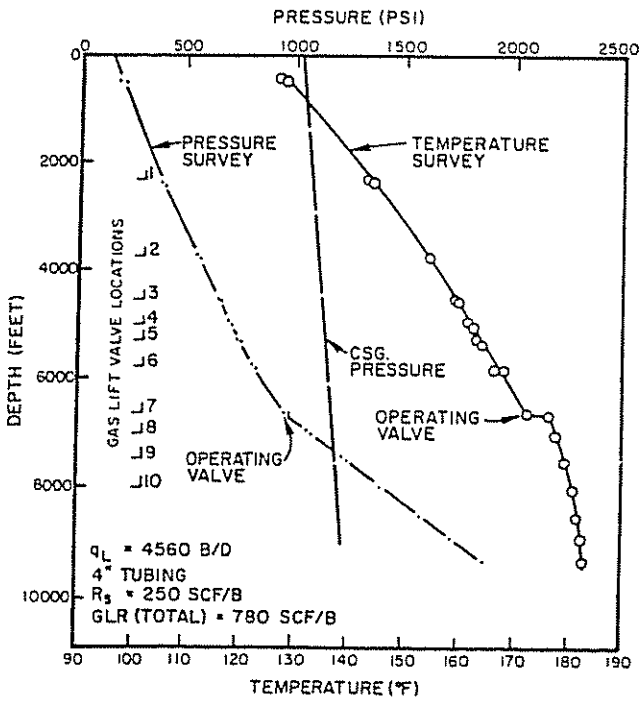


Fig. 3.614-E Combination Flowing Pressure Flowing Temperature Survey

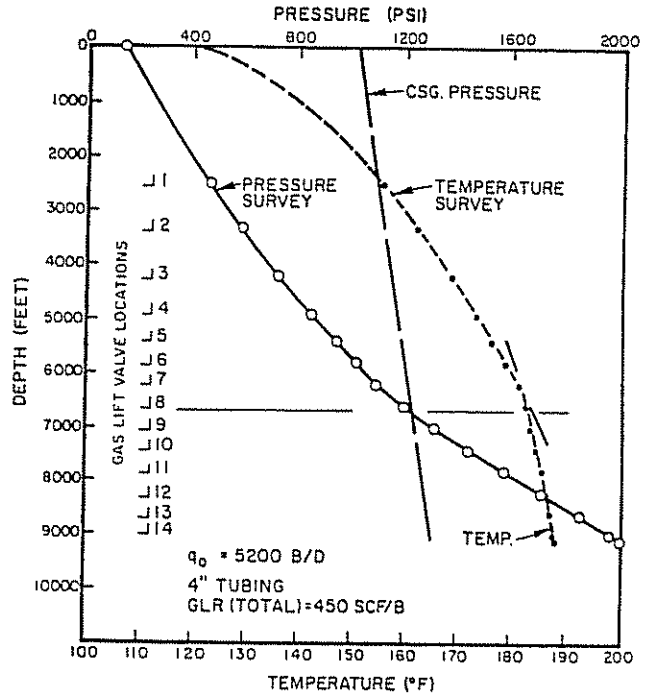


Fig. 3.614-F Combination Flowing Pressure Flowing Temperature Survey

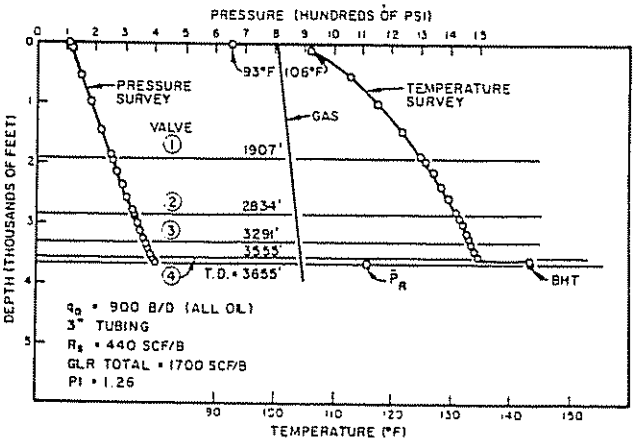


Fig. 3.614-G Combination Flowing Pressure Flowing Temperature Survey

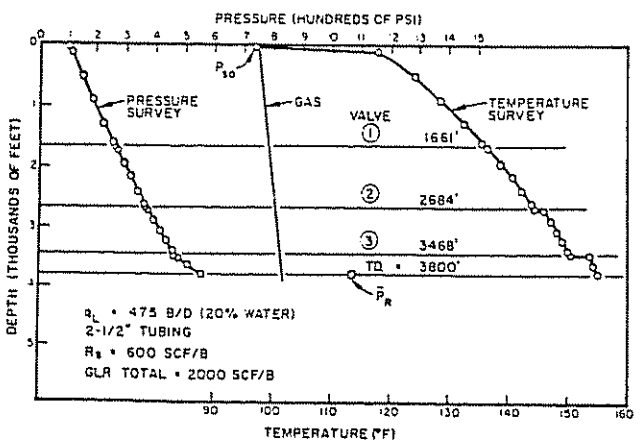


Fig. 3.614-H Combination Flowing Pressure Flowing Temperature Survey

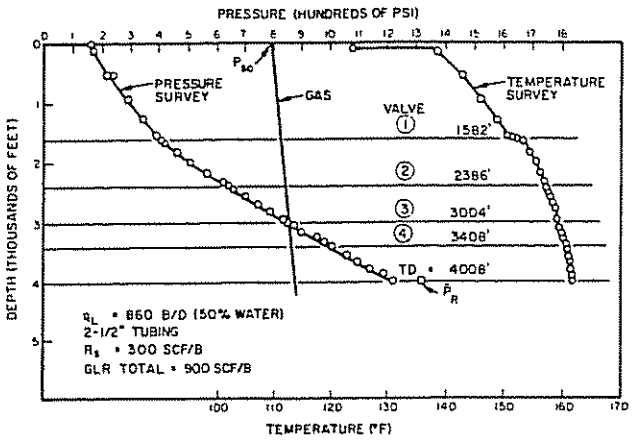


Fig. 3.614-I Combination Flowing Pressure Flowing Temperature Survey

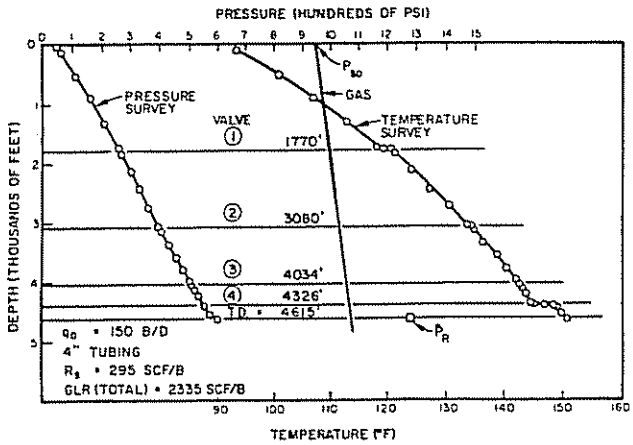


Fig. 3.614-J Combination Flowing Pressure Flowing Temperature Survey

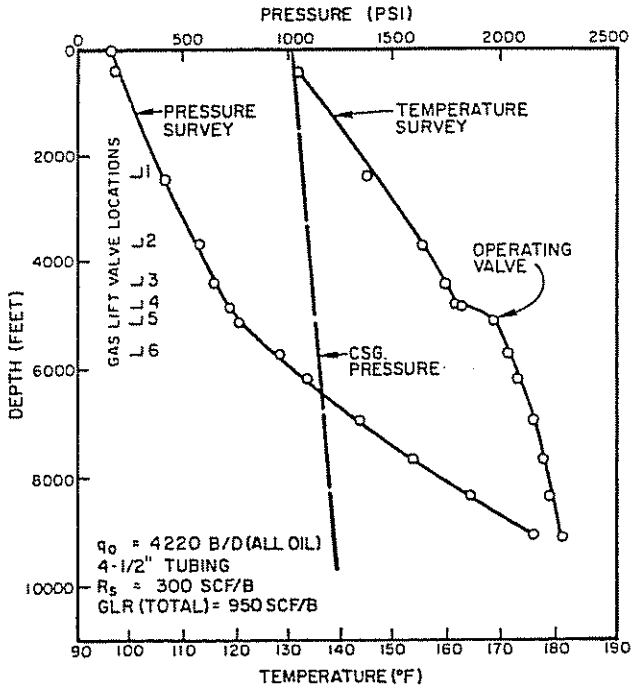


Fig. 3.614-K Combination Flowing Pressure Flowing Temperature Survey

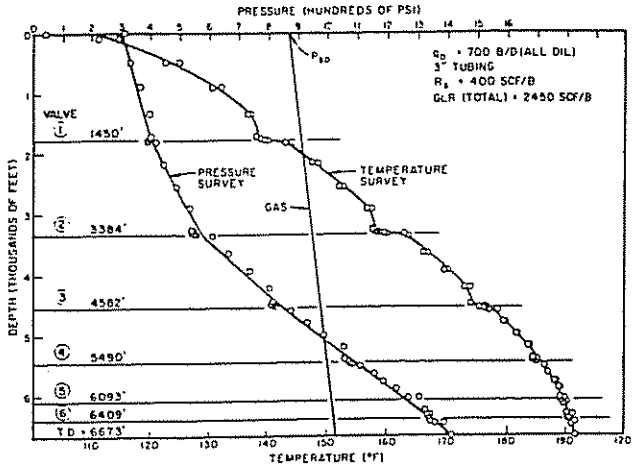


Fig. 3.614-L Combination Flowing Pressure Flowing Temperature Survey

Figure 3.614H:

This is a case where the temperature and pressure survey show Valve 3 to be passing most of the gas. The temperature survey shows Valve 2 to be passing some gas, but the pressure survey fails to confirm this. Again, a large difference in the temperature as recorded in the lubricator and immediately below the surface exists.

Figure 3.614I:

Both the pressure and temperature surveys show Valve 1 to be the operating valve. The pressure survey offers additional information in that sufficient casing pressure is available to open Valves 2 and 3. Either Valve 1 has lost pressure or Valves 2 and 3 are plugged or have failed in the closed position.

Figure 3.614J:

Figure 3.614J is another case where the temperature survey is a better indicator of gas passing through Valves 1 and 2. The high GLR shows too much gas being injected.

Figure 3.614K:

This is a good survey in a high volume well clearly showing Valve 5 to be the operating valve.

Figure 3.614L:

This is a case where both the temperature and pressure survey show Valves 1, 2, and 3 to be passing gas, confirmed by a high injection GLR.

Figure 3.614M:

Figure 3.614M shows a case of circulating gas through the bottom valve. Since the production rate is zero, evidently the formation is plugged or a sand bridge exists in the tubing below Valve 2.

Figure 3.614N:

Figure 3.614N shows Valve 1 to be passing gas with a production rate of zero. Valve 2 must be plugged since sufficient gas pressure exists to open it.

Figure 3.614O:

Figure 3.614O shows Valve 3 to be passing most of the gas by both the temperature and pressure survey. Valve 4 is passing some gas and the temperature survey shows Valves 1 and 2 are passing gas. The pressure survey fails to verify this.

Figure 3.614P:

In Figure 3.614P both the pressure and temperature survey show Valve 3 to be the operating valve. Howev-

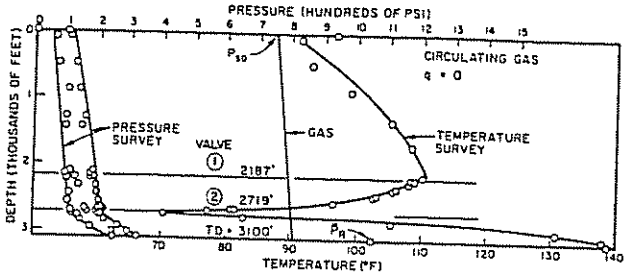


Fig. 3.614-M Combination Flowing Pressure Flowing Temperature Survey

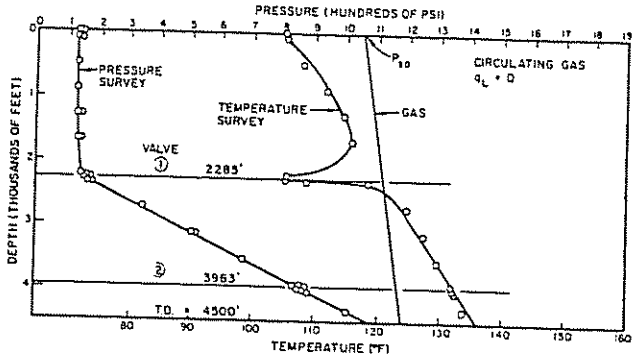


Fig. 3.614-N Combination Flowing Pressure Flowing Temperature Survey

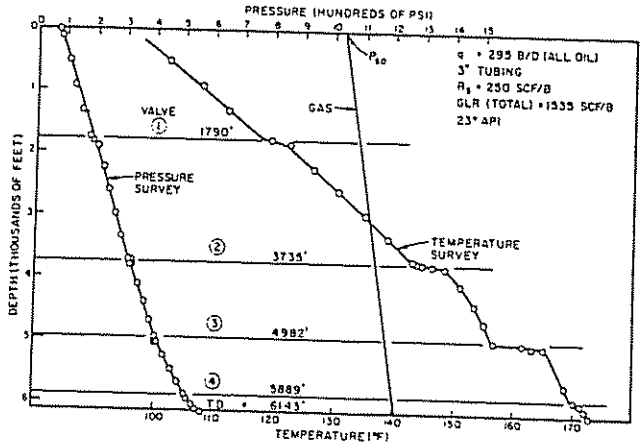


Fig. 3.614-O Combination Flowing Pressure Flowing Temperature Survey

er, gas should be entering Valve 6 since sufficient casing pressure exists to open it.

3.635 Pressure surveys—intermittent lift

3.6351 Introduction

It is more difficult to run a flowing pressure survey on an intermittent well than a continuous flow well. This is particularly true if it is unknown which valve is operating. The danger occurs when a slug of liquid and follow-up gas attempt to come by the bomb. It may blow the pressure bomb up the hole, kink the wire line, and cause a dropping of the bomb. Observe caution in running a bomb into an intermittent well. The following procedure is recommended.

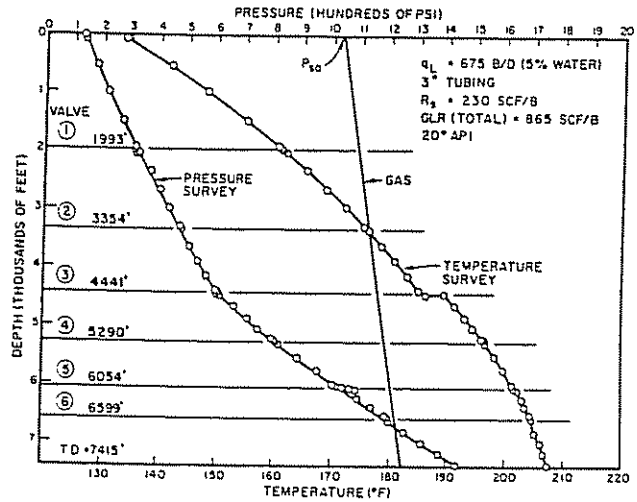


Fig. 3.614-P Combination Flowing Pressure Flowing Temperature Survey

(1) Be sure that the time cycle intermitter on the injection line is closed.
(2) Run the bomb to a position below the intermitting valve, which in most cases will be the bottom valve. At any rate, start from below the bottom valve.
(3) Open the time cycle intermitter and produce the accumulated head of fluid. Use excess gas to clear out additional liquid accumulated while running the bomb to bottom.
(4) Close the intermitter.
(5) Place the intermitter on its standard production cycle.
(6) Allow the bomb to record several kicks.
(7) Pull the bomb at this time and inspect the chart. By checking the flowing BHP, the operating valve can be determined. If it is not the bottom valve, return with a new chart and stop the bomb beneath the operating valve. Repeat the testing procedure for the now known operating valve.
(8) Allow the bomb to remain on bottom long enough to register pressures well beyond the normal slug accumulation and, if possible, long enough to approach maximum static pressure.

Valuable information can be obtained from this survey. Such things as the initial slug load being lifted can be read off the bomb. In addition, the minimum pressure between cycles can be determined. By knowing the location of the operating valve, the portion of the fallback or slippage not clinging to the walls can be measured in pressure. An average flowing BHP can be calculated from the pressure survey chart. This in turn will allow the calculation of a PI.

The pressure bomb, while helping to analyze intermittent installations, may be a very useful tool in determining slippage. The pressure bomb should again be run to bottom and left immediately below the operating valve. The measured head of fluid may then be calculated from the pressure buildup on the bomb, of course considering tubing back pressure and formation volume factor. The actual slug of fluid that reaches the

surface may then be measured in the stock tank and a comparison made between the volume of stock tank oil and the volume of oil converted from the head of pressure and corrected to stock tank conditions.

The operating valve can be determined even though the survey is run while the bomb is on bottom and lift is actually occurring from an upper valve. For example, if a valve is located at 6000 ft and a valve opening pressure at depth is 750 psi, the maximum pressure that can be recorded in the tubing is 750 psi if lift is occurring from that valve. This can only occur when the valve opens. If the tubing pressure recorded exceeds 750 psi, then the operating valve is above the valve at which the bomb is stopped. By knowing the gradient of the inflow fluid, the operating valve can be determined.

Assume that the maximum recorded pressure is 1250 psi at 6000 ft. Evidentally, lift cannot be occurring from the 750 psi valve at 6000 ft. Taking (1250-750) = 500 psi indicates a point back up the hole an equivalent footage of 500 psi. If the fluid gradient is 0.345 psi/ft, this would represent (500/0.345) = 1450 ft. If calculations are correct, there should be a gas lift valve at (6000-1450) = 4550 ft. If this is verified, the bomb should be run back to slightly below this valve and the test repeated.

3.6352 Hypothetical pressure surveys

Figure 3.615A:

Figure 3.615A shows a hypothetical case of an intermittent flowing pressure survey. Several things are to be noted from this survey. It appears to be a good installation since the operation is from the bottom valve. The cycle time is 25 minutes. The tubing load is 440 psi and 625 psi is placed underneath the slug when the valve opens. Recovery is excellent as noted by a low minimum pressure of 90 psi.

Figure 3.615B:

Figure 3.615B shows a pressure survey where the bomb is run below the bottom valve (below 5950 ft.), yet the next valve above at 4950 ft is the operating valve. It is fairly easy to determine since 1000 ft × .40 = 400 psi additional pressure from Valve 2 to Valve 3. For better information, the bomb should now be run some 30 ft below Valve 2 at 4950 ft and the tests repeated.

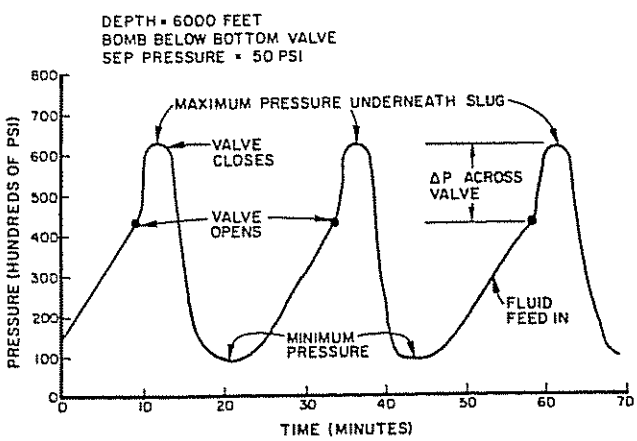


Fig. 3.615-A Typical Pressure Survey Intermittent Lift Standard Installation

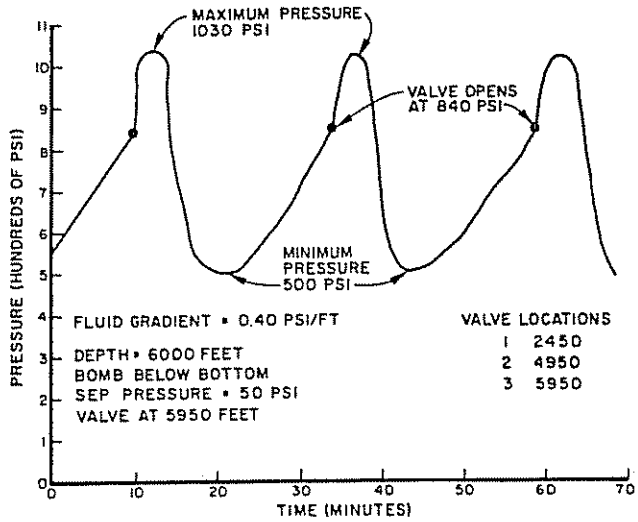


Fig. 3.615-B Intermittent Pressure Recording—Operating Valve One Valve above Bomb Position

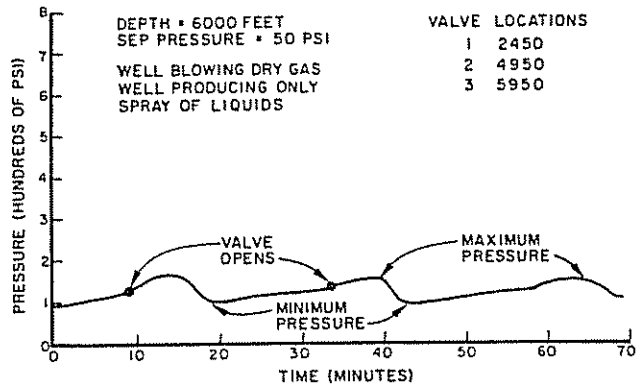


Fig. 3.615-C Intermittent Flowing Pressure Survey—Blowing Dry Gas

Figure 3.615C:

Figure 3.615C is a case whereby a well is producing no liquids, probably due to a sand bridge in the tubing since the bomb is placed slightly below the bottom valve.

Figures 3.615D and 3.615E:

Figures 3.615D and 3.615E illustrate the method of calculating a weighted average BHP. Details are given in Section 3.4362. Also Figure 3.615D shows a well without a standing valve as compared to Figure 3.615E which incorporates a standing valve. Note that a lower flowing pressure; hence, greater production can be maintained with a standing valve.

3.6353 Field cases of intermittent pressure surveys

Several actual field cases are given in the following discussions as given by Winkler.[3]

Figure 3.616A:

Figure 3.616A shows a flowing pressure survey on a standard intermittent installation. Although the cycle time is 6 hours, a shorter time would not increase the production because the fill-in of fluids follows a straight line between cycles. Excellent recovery is

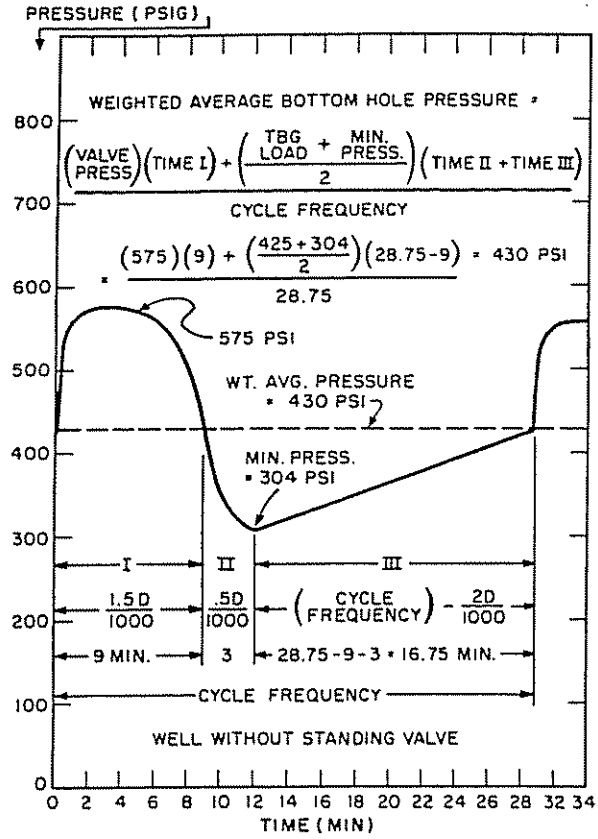


Fig. 3.615-D Time Periods and Weighted Average Flowing BHP for a Typical Intermittent Gas Lift Cycle (after Beadle, et al,[7] Courtesy Jour. Pet. Tech.)

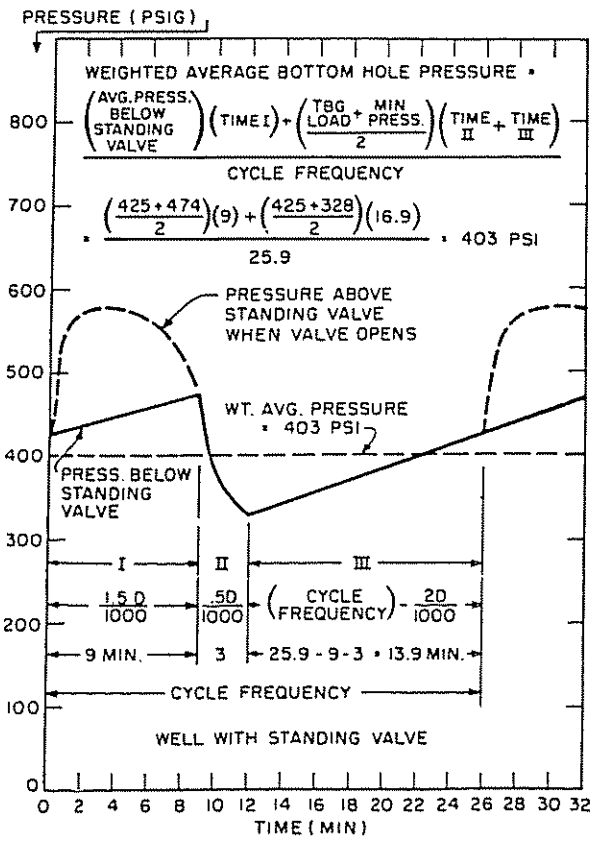


Fig. 3.615-E Weighted Average Flowing BHP with Standing Valve (after Beadle, et al,[7] Courtesy Jour. Pet. Tech.)

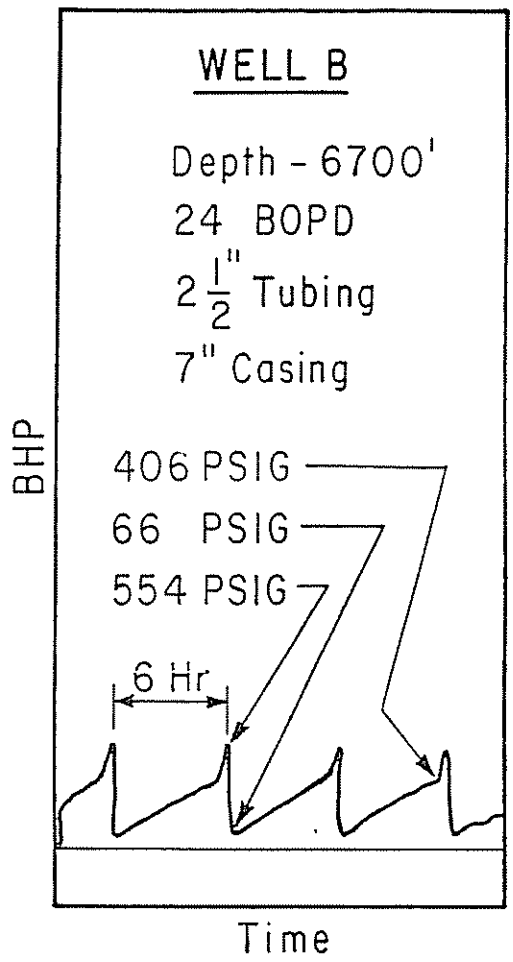


Fig. 3.616-A Flowing Pressure Survey Intermittent Installation (after Winkler, Courtesy West Texas Oil Lifting Short Course)

obtained as noted by the low minimum pressure of 66 psi.

Figure 3.616B:

Figure 3.616B shows a flowing survey in an intermittent chamber installation. This is a low volume well producing 42 B/D from 6750 ft and appears to be operating very efficiently.

Figure 3.616C:

Figure 3.616C shows a flowing survey run on a special automatic vent chamber installation as described in Section 3.3.[4]

A pressure instrument was lowered with a wire line into the 1.900"-OD vent tubing to 8,230 ft. The minimum pressure while the chamber was filling was 145 psi (surface pressure plus gas weight was 75 psi). As soon as the chamber vented, there was a pressure surge exceeding 900 psi which declined to less than 200 psi as the chamber refilled. As seen in Figure 3.616 C, there was a good drop in wellbore pressure from that which prevailed during continuous flow and before the AVC system was installed.

Figure 3.616D:[3]

Figure 3.616D shows one cycle of an intermittent survey for a well with a standard tubing installation. There exists one unexplainable kick after the valve closes and nearing the minimum pressure. This could represent another well kicking into the same separator, particularly if the flowline check were not holding.

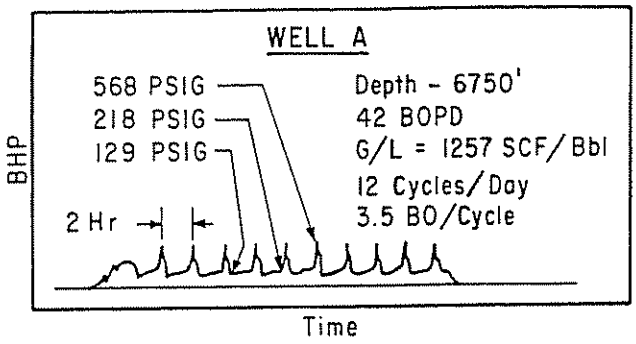


Fig. 3.616-B Flowing Pressure Survey Intermittent Chamber Installation (after Winkler, Courtesy West Texas Oil Lifting Short Course)

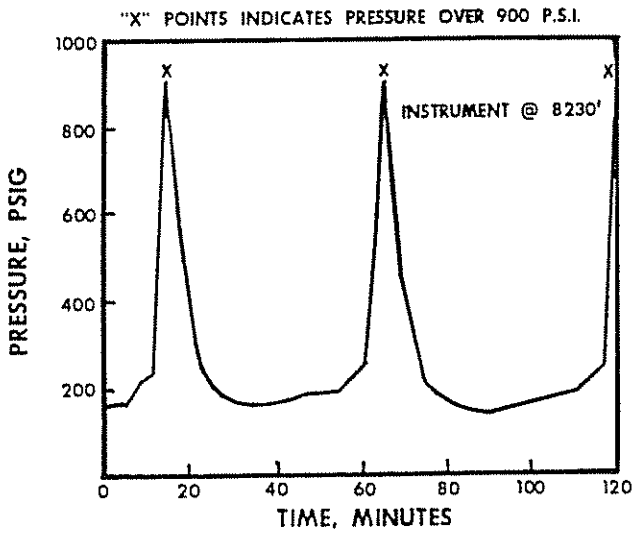


Fig. 3.616-C Subsurface Pressure Survey, Special Chamber (after DeMoss)

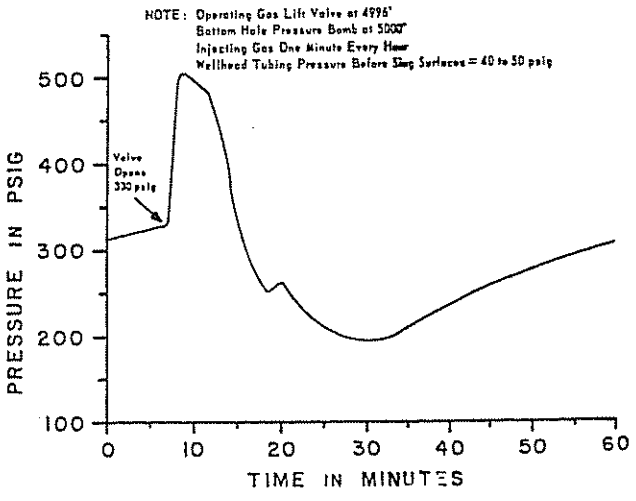


Fig. 3.616-D Pressure Survey in Intermitting Well (after Winkler)

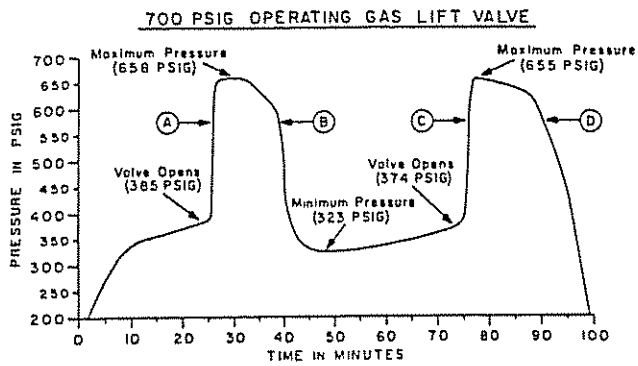


Fig. 3.616-E Flowing Pressure Survey in Intermittent Well (after Winkler)

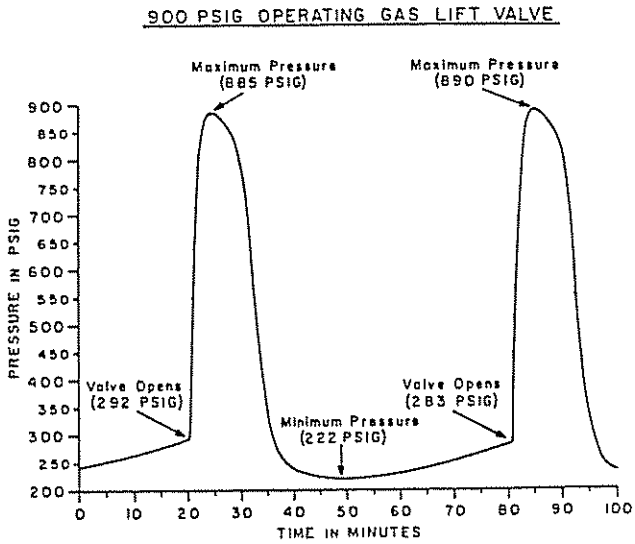


Fig. 3.616-F Pressure Surveys above Standing Valve at 6580 ft. in a Chamber Installation (after Winkler)

Figure 3.616E:[3]

Figure 3.616E shows a typical survey in an intermittent well with the following noted:

(1) The valve opens initially on a tubing load of 385 psig.
(2) Line A shows pressure increase as casing pressure increases underneath the slug.
(3) Maximum pressure underneath the slug reaches 658 psig.
(4) Valve closes at top of line B and pressure drops rapidly.
(5) A minimum pressure of 323 psig occurs.

Figure 3.616F:[3]

Figure 3.616F shows an intermittent survey on a chamber installation. The valve opens at a pressure of 292 psig and increases to a maximum of 885 psig when the gas lift valve opens. This is due to the liquid load being U-tubed from the chamber into the tubing. A minimum pressure of 222 psig could indicate relatively inefficient operations in that a differential liquid load of $292 - 222 = 72$ psig is being removed each cycle. Depending upon casing size, this may or may not be a good recovery each cycle.

3.636 Well sounding devices

3.6361 Introduction

A quick method for determining the liquid level in either a continuous or intermittent flow well is by sounding the well. This is an acoustic device that is economical to use since several wells can be sounded in one day. This method (by firing a cartridge at the surface) utilizes the principle of a sound wave traveling down the tubing string. It will detect all couplings, gas lift valves, and liquid level in the annulus.

Exercise caution in analyzing a well by this method. For a well with a packer, the liquid level may be a misleading indicator of the operating gas lift valve. The well may have originally unloaded to the bottom valve and later moved up the hole due to increased liquid feed-in or a faulty valve. Although the acoustic device would show the well unloaded to the last valve, it is possible that a valve higher in the string would actually be the operating valve. Many wells originally placed on lift will require additional drawdown in the first stages of lift, thereby unloading the annular space of fluid. These same wells, after several days of production from a lower valve, may then operate from a valve higher up the string. Brown and Canalizo offer the following additional comments:[5]

> The acoustic device would be very well suited for continuous flow installations without a packer. The annular working fluid level could be easily detected in this manner. Therefore, a valve losing pressure and causing the fluid to rise could be easily detected. It would be more satisfactory to take periodic soundings and, in particular, one sounding when the well is performing satisfactorily. This would allow for comparisons of changing fluid levels.
>
> Also for intermittent wells with a packer, where it is known that the well should be operating from the bottom valve, this could be an immediate check. If the fluid level has moved back up the string, a valve pressure loss could be suspected. A packer is normally run on an intermittent well and the lowest fluid level attained during the artificial lift period would be the level determined with the sounding device.

Figures 3.617A and 3.617B show the types of guns used for acoustic surveys. Figure 3.617A is the most commonly used cartridge type gun and 3.617B is the gas gun.

The following discussion is given by "Echometer":[6]

> A gun-microphone assembly consists of a gun and a microphone together. A gun assembly generates a pressure pulse only. A microphone assembly converts a pressure pulse into an electrical pulse.
>
> The gun emits a single pressure pulse when desired which travels through the gas down the casing annulus and is reflected by collars and the fluid level. The reflected pressure waves deflect the microphone and generate electrical signals. These electrical signals are fed to the amplifier-recorder. A "good" pressure pulse is necessary for suitable reflected signals. If the initially generated pressure pulse is weak, then the reflected signals will be weak. Two types of guns are available. The more common gun utilizes blanks. The other type utilizes compressed gas and is called a gas gun.
>
> Two important factors must be considered when selecting the type of gun. First, the gun should generate a pressure wave that results in the collars being counted as far down the well as possible. Extensive studies were performed which indicated that 70 cubic inches of compressed gas liberated into a low pressure casing annulus creates a "good" pressure pulse, and a greater gas volume does not count collars appreciably further down a well. The 45 caliber blank creates about 70 cubic inches of gas. The effectiveness, convenience, availability and inexpensiveness of the 45 caliber blank has been responsible for making the 45 caliber blank the most popular. Second, the gun must generate a pressure pulse of sufficient strength to cause the reflected pressure wave from the fluid level to be readily distinguishable. The fluid level reflected pressure pulse is stronger when larger volumes of gas are discharged into the casing annulus. If a strong fluid level response is recorded with a small 45 caliber blank (70 cubic inches of gas), then a larger blank (or greater volume of gas) does not improve performance because no additional collars are counted. On deep, slim hole, lower pressure wells, the reflected fluid level pressure wave is weak and a larger initial pressure pulse may be necessary. The 10 gauge blank contains 6 times as much black powder as the 45 caliber blank, generates about 400 cubic inches of gas, and results in approximately 3 times greater reflected fluid level pressure waves. Very little difference in collar response is noted however.
>
> The need of a large initial pressure pulse becomes obvious in a deep (the sound must travel a far distance), slim hole (the sound travels a restricted path), lower pressure (compressed gas carries the pressure pulse better) well. Gas guns have been developed to

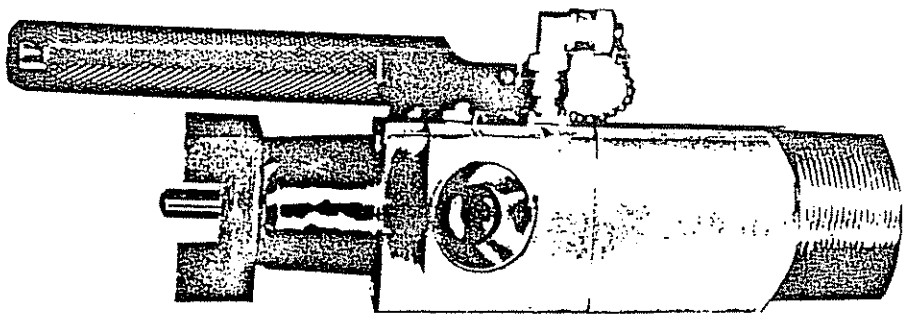


Fig. 3.617-A Cartridge Type Microphone Assemoly (after Echometer)

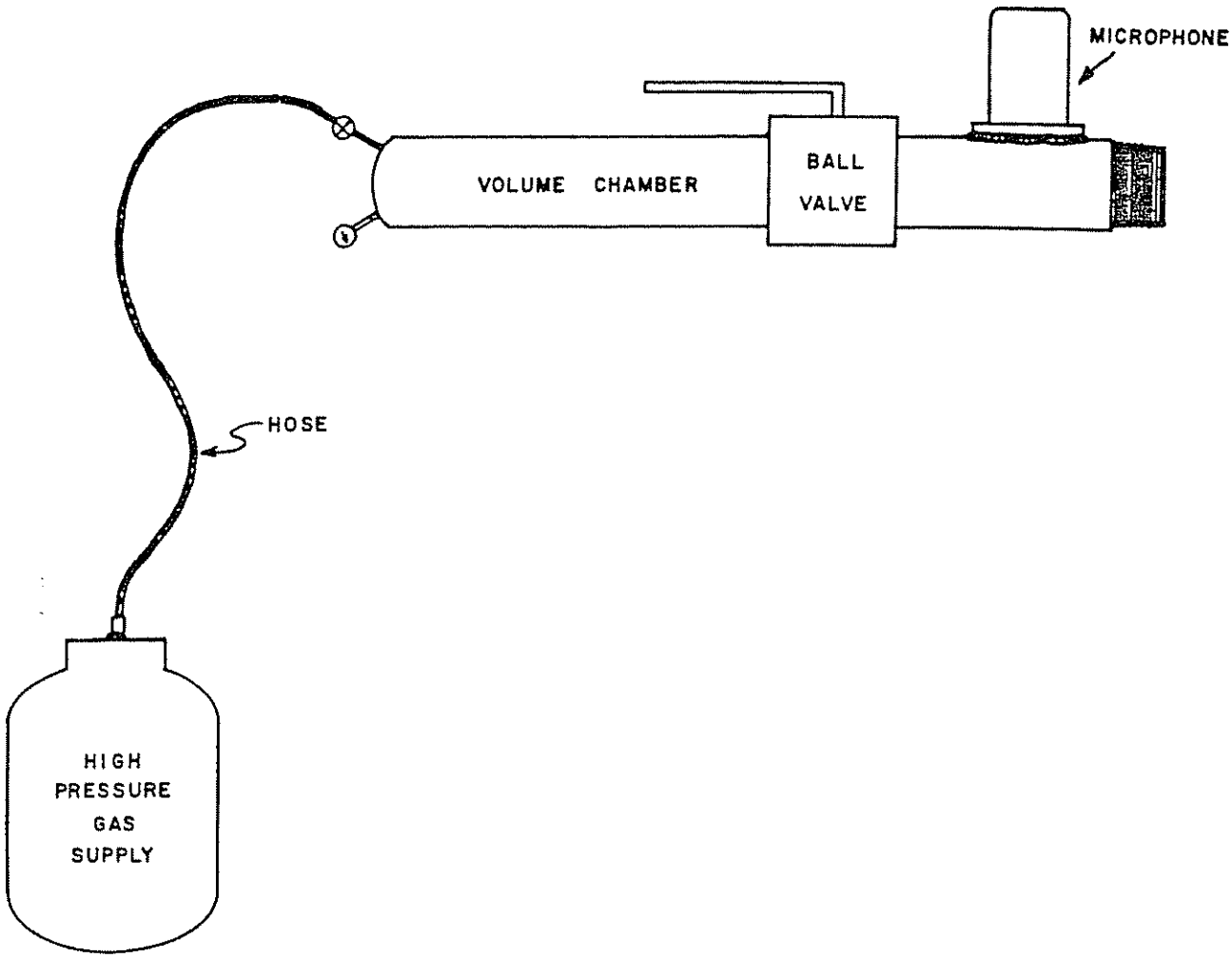


Fig. 3.617-B Gas Gun Microphone Assembly (after Echometer)

supply these large volumes of gas. A gas gun can be built from a 2" nipple, 2 feet long, with a ball valve for the rapid release of the gas. A typical gas gun is shown in Figure 3.617B.

The gas gun normally consists of a ball valve for the rapid release of gas from the volume chamber, a volume chamber which has a volume of approximately 75 cubic inches (or 1 quart), a pressure gage on the volume chamber for measuring the amount of gas charge, a hose to fill the volume chamber from a high pressure gas supply, and a high pressure gas supply.

In normal operation, the gas gun volume chamber is charged to 100 PSI above casing pressure. However, a lower pressure may be sufficient, or a higher pressure may be required for satisfactory results.

Operating instructions for the gas gun:

(1) Connect gas gun (and microphone) to well
(2) Open valve to well
(3) Open gas gun ball valve and measure well casing pressure
(4) Close ball valve
(5) Open valve between volume chamber and high pressure gas supply to increase pressure in volume chamber to 100 psi above casing pressure. Note, a lower differential pressure may be sufficient or a higher differential may be required for some wells.
(6) Close valve between hose to high pressure gas supply and volume chamber
(7) Open ball valve rapidly to generate a pressure pulse

Figure 3.617C shows how upper and lower couplings can be accented as well as the liquid level.

3.6362 Field cases of acoustic surveys

Figure 3.617D:

Figure 3.617D shows a typical good acoustic survey on a pumping well. Initially, large kicks are recorded for each coupling and then dampen out. The liquid level is easily detected in this case although difficulty in detecting each coupling immediately above the liquid level is experienced. By using a set of dividers to obtain distance between several joints near the surface where the recordings are good, it is then easy to count total joints. The latest and most sophisticated equipment has automatic counters, etc.

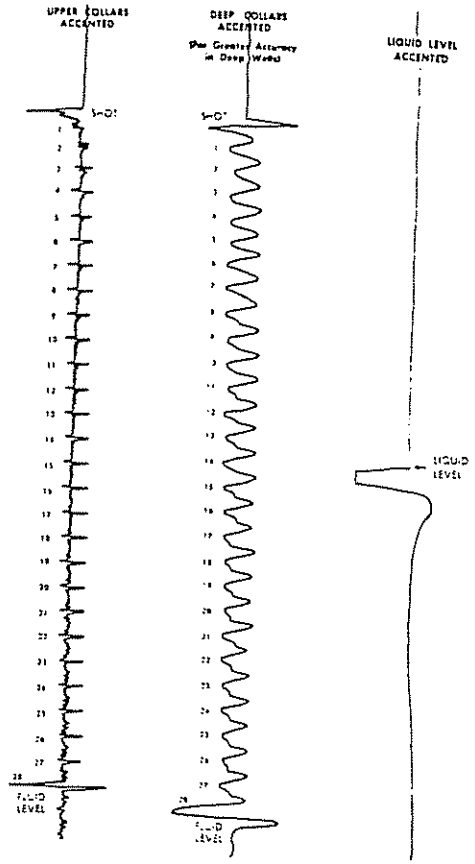


Fig. 3.617-C Typical Acoustic Recordings (after Echometer)

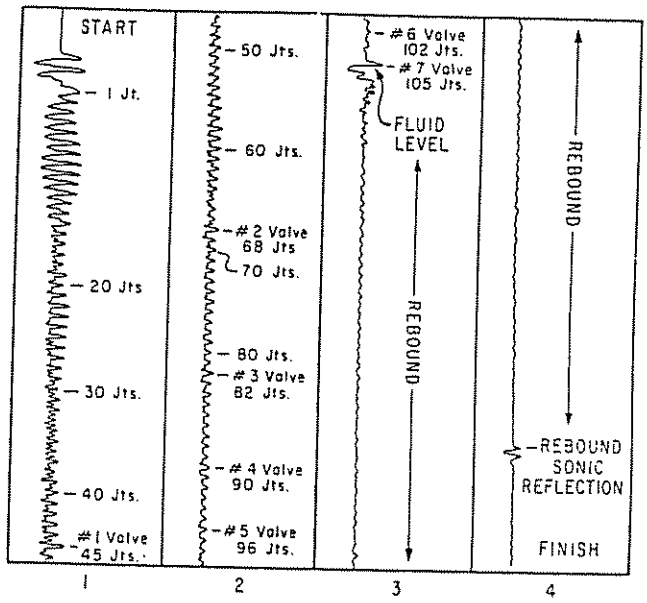


Fig. 3.617-E Typical Acoustic Survey of Gas Lift Well (Courtesy API[5])

Figure 3.617E:

Figure 3.617E shows a typical survey to locate each gas lift valve as well as pick up the fluid level at Valve 7. This clearly shows that liquid has been unloaded from the annulus to Valve 7, but does not confirm that Valve 7 is the operating valve.

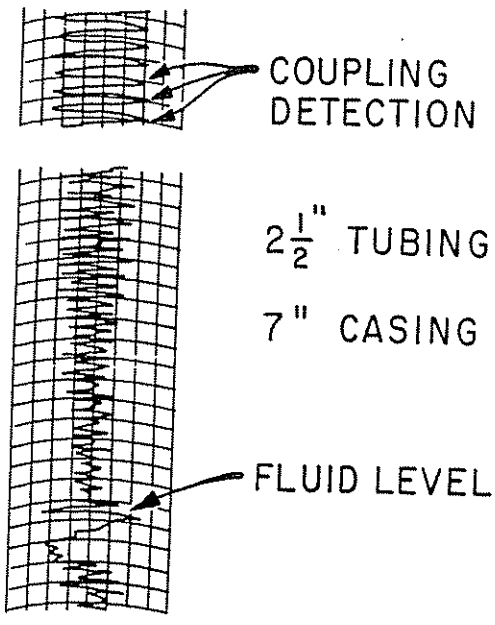


Fig. 3.617-D Typical Acoustic Survey on Pumping Well

3.637 Surface recordings of casing and tubing pressures

3.6371 Introduction

A gas lift installation should not be considered complete until a two-pen pressure recording instrument has been installed to make a daily recording of the tubing and casing pressure. However, if for economic reasons the installation cannot be justified, a portable recording instrument should be made available for spot checking individual wells.

A recording should be made while the well is performing properly, so comparisons may be made if any trouble develops. By careful study of the tubing and casing pressures, many troubles may be interpreted and corrected without the expense of pressure surveys or tubing jobs. An improvement by adjusting surface controls can be made in many instances.

These charts can be used for either continuous or intermittent flow gas lift. If the recorder is placed on the well as part of the initial installation, the unloading of the well can be watched and, in most cases, each valve uncovered in the unloading process can be noted. The following information can be checked and/or obtained from recording charts:

(a) *Leaking tubing, packer, or gas lift valve:* noted on intermittent lift by a drop in casing pressure between cycles.

(b) *Surface restriction:* noted by high pressure build-up during intermittent kick or by high tubing pressure on a continuous flow well.

(c) *Freezing:* noted by failure to inject gas and no flu-

id recovery. Tubing pressure will settle back to separator pressure.

(d) *Well sanded up or plugged:* chart will not show slug buildup on intermittent lift and will record low on continuous lift.

(e) *Cycles too fast on intermittent lift:* noted by a fast drop in casing pressure and small tubing kicks.

(f) *Cycles too far apart if on intermittent lift:* noted by a slow drop in casing pressure and large broken tubing slugs or no good slug surfacing.

(g) *Changing valve or alternating valve operation:* noted by a lowering and raising of the casing pressure.

(h) *Loss of operating valve dome pressure:* casing pressure recording will be considerably less than the calculated surface operating pressure of any valve installed.
Note: This may indicate a leaky pack-off in pack-off gas lift valve installations or a leaky packer on an intermittent installation.

(i) *Shallow casing leak:* casing pressure recording may indicate the same as a leaking valve, hole in tubing, or loss of dome pressure, but there is no indication of gas flow from the tubing.

(j) *A well may start flowing under its own source of power:* if the injection gas was controlled by a tubing shutoff device, this would be noted on a recording chart.

The installation of these recorders should not be directly on the Christmas tree. Vibrations from produced fluids will generally cause trouble. A flexible hose connection and a mounting of the recorder away from the Christmas tree or flowline will ensure best results.

3.6372 Continuous flow recorder charts

3.63721 Hypothetical charts

Several hypothetical cases are presented to show typical cases that arise in the use of two-pen pressure recorder charts for continuous flow.

Figure 3.618A(1):

Figure 3.618A(1) shows a high rate producing oil well with no problems. With 1300 psi surface casing pressure, it is producing 1800 B/D against 300 psi wellhead pressure. An injection GLR of 600 scf/B indicates a good efficient operation. A possible improvement would be to lower the wellhead pressure.

Figure 3.618A(2):

Figure 3.618A(2) shows the same well as Figure 3.618A(1), only the casing pressure is lowered to 1000 psi and the rate has dropped to 1300 B/D. This shows that an upper valve has lost pressure, causing the casing pressure to lower to 1000 psi and to decrease the rate. The wellhead pressure lowered slightly due to the decreased rate. The GLR increased to 1000 scf/B.

Figures 3.618B(1) and 3.618B(2):

Figure 3.618B(1) shows P_{wh} to be 400 psi and a reduced production to 600 B/D. By lowering P_{wh} to 150 psi as noted in Figure 3.618B(2), the rate doubled to 1200 B/D.

Figures 3.618C(1) and 3.618C(2):

Figure 3.618C(1) shows a case whereby a bad valve or leak caused the casing pressure to lower to 675 psi and the wellhead pressure increased to 350 psi. The

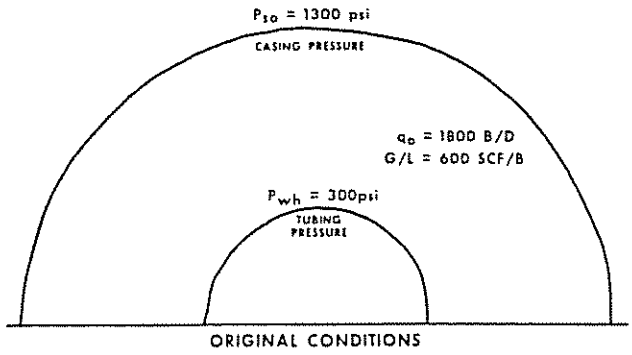


Fig. 3.618-A (1) Continuous Flow Well

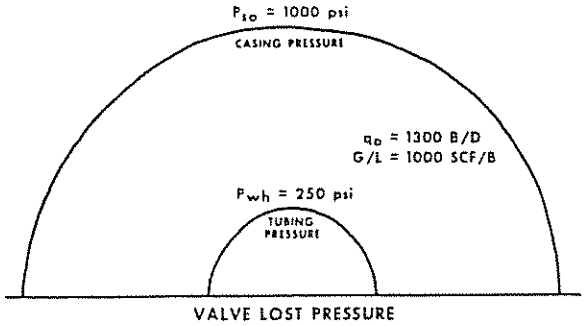


Fig. 3.618-A (2) Continuous Flow Well

CONTINUOUS FLOW WELL

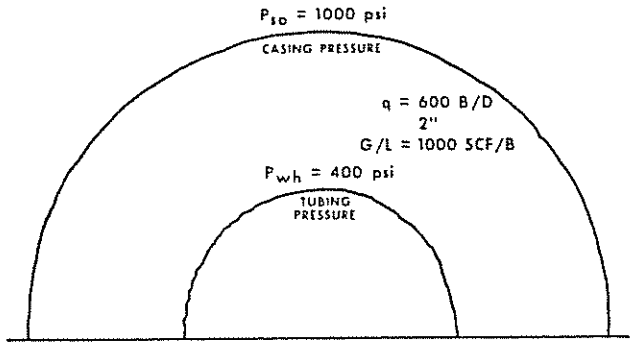


Fig. 3.618-B (1) High Wellhead Pressure

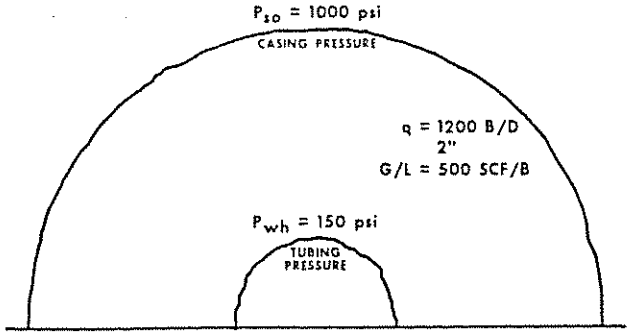


Fig. 3.618-B (2) Effect of Lower P_{wh}

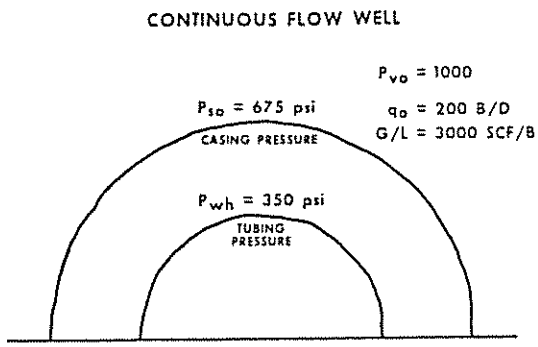


Fig. 3.618-C (1) Bad Gas Lift Valve—Lost Pressure or Leaking Seat

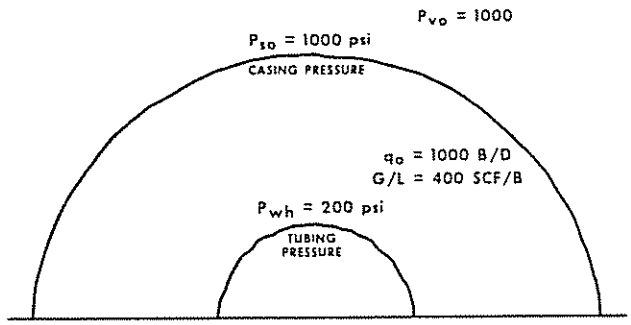


Fig. 3.618-C (2) Pulled Gas Lift Valve and Repaired

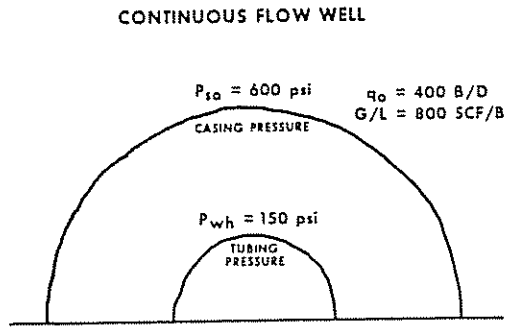


Fig. 3.618-D (1) System Pressure Too Low

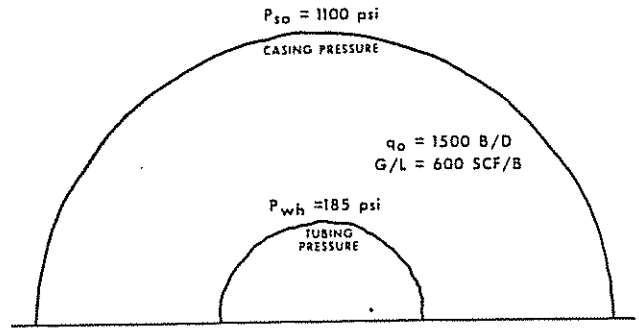


Fig. 3.618-D (2) Changed System Pressure to 1100 PSI

rate decreased to 200 B/D for a high GLR of 3000 scf/B.

Figure 3.618C(2) shows the results after repair to 3.618C(1). The operating pressure is now 1000 psi, P_{wh} = 200 psi, the rate increases to 1000 B/D, and the GLR decreases to 400 scf/B.

Figures 3.618D(1) and 3.618D(2):

Figure 3.618D(1) shows a case whereby the operating gas pressure is low (600 psi). The rate is 400 B/D. By increasing the pressure to 1100 psi, as noted in Figure 3.618D(2), the rate was increased to 1500 B/D.

Figures 3.618E(1) and 3.618E(2):

Figure 3.618E(1) shows a well with a production rate of zero. The casing pressure has increased to 1400 psi due to a low tubing pressure acting on the seat area of the valve. Figure 3.618E(2) shows that a sand bridge was removed, allowing the well to produce 1200 B/D. The casing pressure has lowered to 1100 psi due to increased tubing pressure. (This was a small O.D. valve with a large seat area.)

Figures 3.618F(1) and 3.618F(2):

Figure 3.618F(1) shows that a hole in the tubing or possibly Valve 1 is passing gas at a low casing pressure of 450 psi. The low production rate and high GLR confirm this.

Figure 3.618F(2) shows that a tubing leak existed and was repaired. The casing pressure is now 1200 psi and the rate has increased to 3000 B/D for a GLR of only 500 scf/B.

Figures 3.618G(1) and 3.618G(2):

Figure 3.618G(1) shows what can happen if trying to continuous flow with a choked intermittent valve. In the open position the valve serves as an orifice only. The choke eventually passes enough gas to lower the casing pressure and close the valve. After casing and

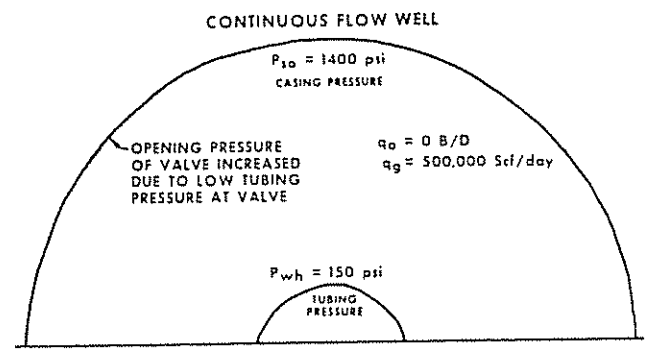


Fig. 3.618-E (1) Well Sanded Up

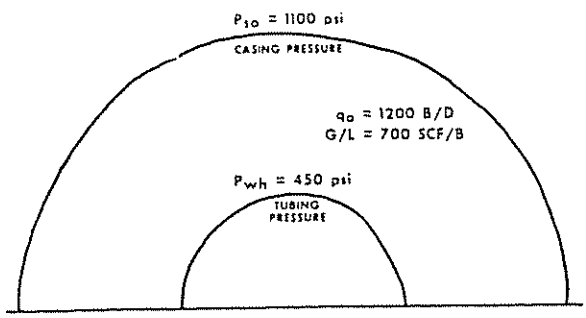


Fig. 3.618-E (2) Sand Bridge Removed

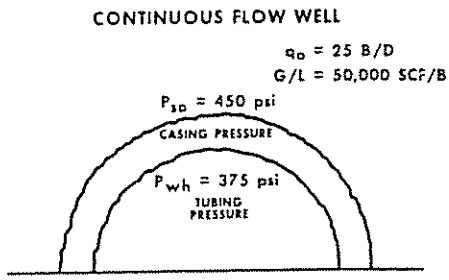


Fig. 3.618-F (1) Hole in Tubing Near Surface Blowing Gas Around

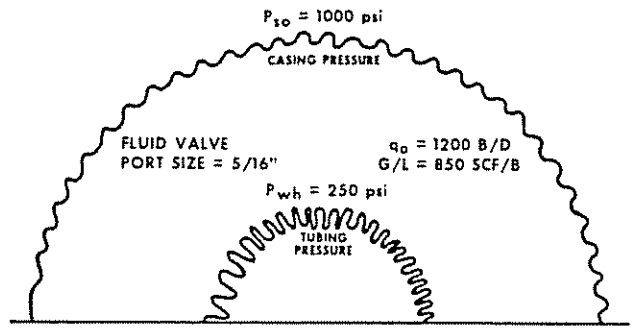


Fig. 3.618-H (1) Trying to Continuous Flow with Choke Size Too Large —Fluid Operated Valve

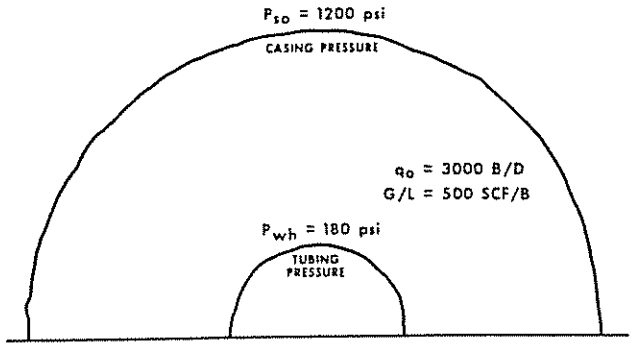


Fig. 3.618-F (2) Repaired Tubing Leak

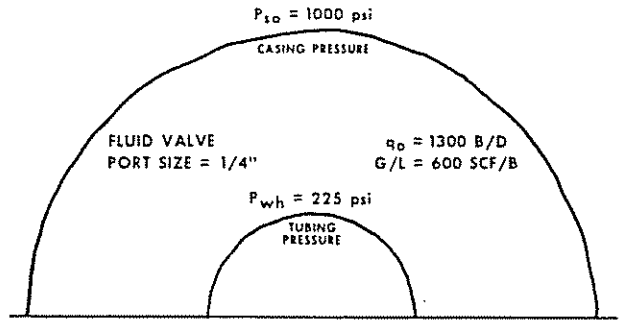


Fig. 3.618-H (2) Pulled Valve and Reduced Port Size to ¼"

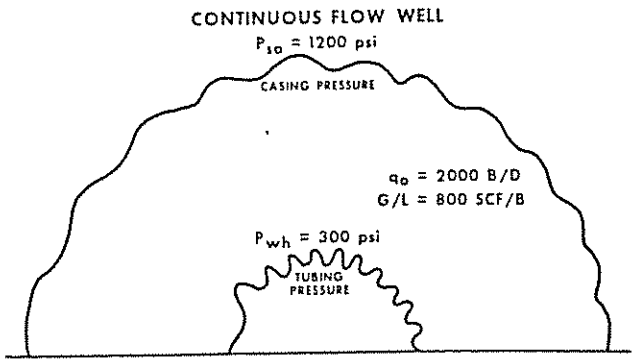


Fig. 3.618-G (1) Trying to Continuous Flow with Choked Intermittent Valve

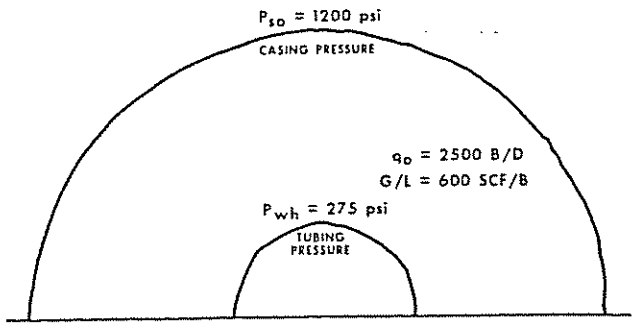


Fig. 3.618-G (2) Replaced Intermittent Valve with Continuous Flow Valve

tubing pressure build-up, it again opens. An intermittent cycle is set up.

Figure 3.618G(2) shows the results after replacing the intermittent valve of Figure 3.618G(1) with a continuous flow valve.

Figures 3.618H(1) and 3.618H(2):

Figure 3.618H(1) shows a case where the choke or seat size is too large on a continuous flow fluid operated valve. An intermittent action is induced. It could possibly be removed on the same valve by increasing the injection gas volume or raising the tubing pressure or both. However, as noted in Figure 3.618H(2), the valve is pulled and its port size is reduced to ¼" from $^5/_{16}$". The behavior improved although there was no appreciable increase in flow rate.

Figures 3.618I(1) and 3.618I(2):

Figure 3.618I(1) shows that a variable line pressure can cause sufficient variation in the casing pressure to set up a surging valve action to cause heading on the tubing.

Figure 3.618I(2) shows how to eliminate this by installing a regulator upstream from the injection choke to maintain a constant operating casing pressure. This smoothed out the operation and resulted in a slight increase in production.

Figures 3.618J and 3.618K:

Figure 3.618J shows a valve with too large a port; hence, it tries to close, causing a heading action.

Figure 3.618K shows another valve whereby the port size is too large.

Figure 3.618L:

Figure 3.618L shows a good continuous flow operation.

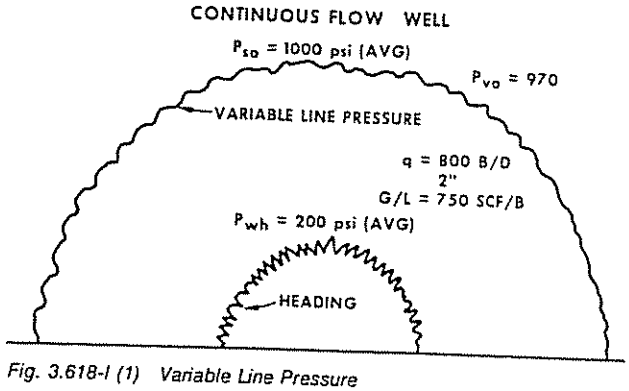


Fig. 3.618-I (1) Variable Line Pressure

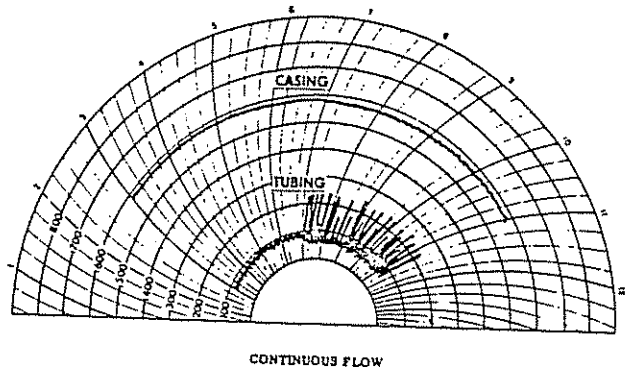


Fig. 3.618-K Intermittent Valve Action. The slight valve opening and closing is a result of the surface casing pressure being near the closing pressure of the valve. Severe periodic opening and closing is apparent from 7:00 on because of the reduced and irregular casing pressure. The port size is too large or the gas volume is too low or both.

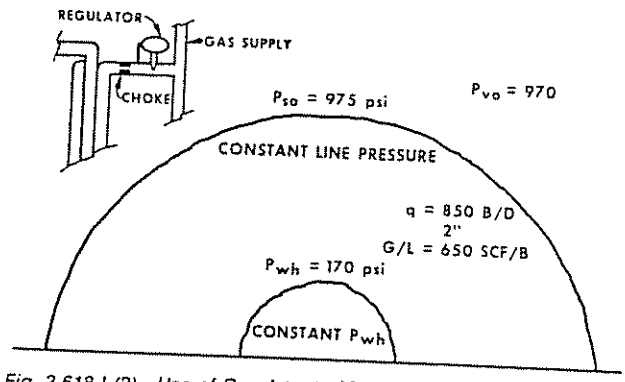


Fig. 3.618-I (2) Use of Regulator to Maintain Constant Supply Pressure Upstream from Choke

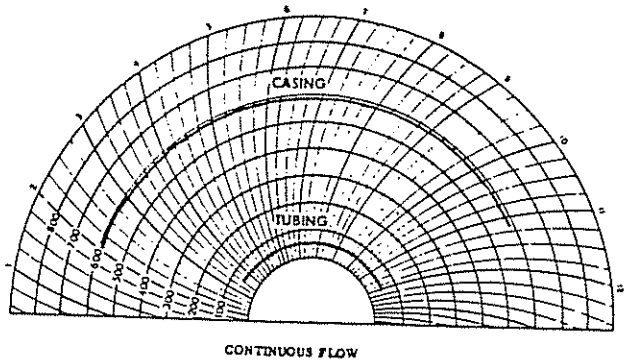


Fig. 3.618-L Good Operation. Note the uniform tubing and casing pressure and the low surface tubing back pressure. High productivity well.

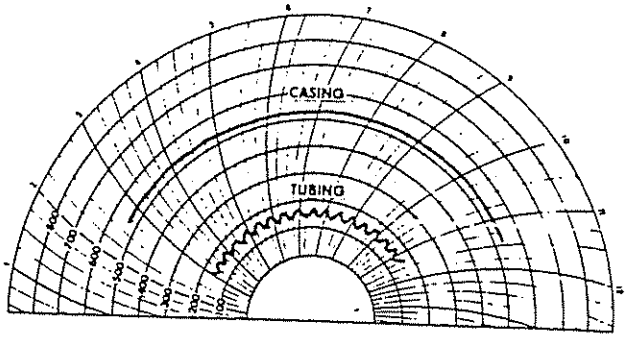


Fig. 3.618-J Valve Port Size Too Large. The wavy tubing pressure line indicates valve opening and closing. This is a high productivity well and the valve action, while undesirable, is not serious. The valve has a large port with the input gas being controlled by means of a casing choke. A smaller port in the valve would eliminate this condition.

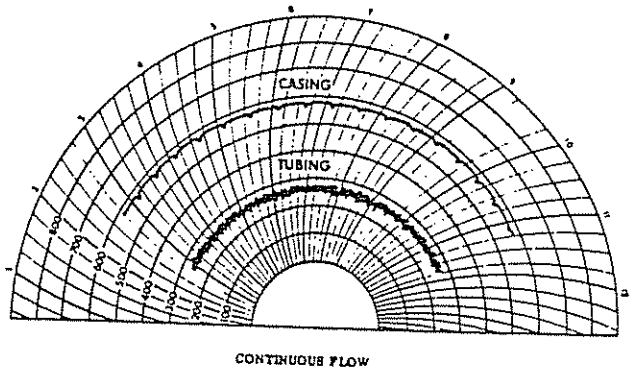


Fig. 3.618-M Illustrating the Effect of High Tubing Back Pressure and Intermittent Well Interference

The periodic drawdown in casing pressure shown on this chart was caused by a nearby intermitting well which had an operating casing pressure considerably less than the line pressure. This situation can be improved by:

1. Increasing the efficiency of the other wells so that they require less gas;

2. Increasing the casing gas supply line pressure;

3.Increasing the storage capacity of the injection system;

4. Redesigning the valve string to operate at a lower surface injection pressure.

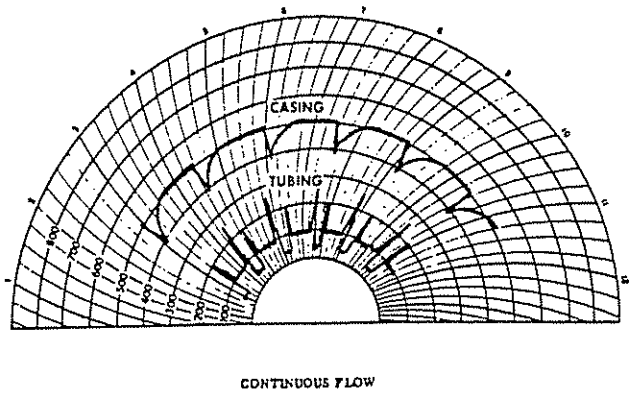


Fig. 3.618-N Tubing Leaks; Parted Tubing String; Gas Periodically Blowing Around the Parted Tubing. A small surface casing choke was used to produce the above chart. When the well was pulled several "corrosion" holes were in the tubing above 4000 feet, the depth at which the tubing had parted. The chart indicates that the well was flowing continuously through the holes in the tubing and would periodically (about 1½-hour intervals) blow around the parted tubing. This is shown by the sharp drop in casing pressure which temporarily stopped the production by aeration until the casing pressure built up again.

Figure 3.618M:
Figure 3.618M shows high back pressure and the effect of an intermittent well taking gas every thirty minutes.

Figure 3.618N:
Figure 3.618N shows a well blowing around parted tubing in periodic intervals.

3.63722 Field cases of two-pen surface recorder charts for continuous flow

Figures 3.619(1) through 3.619(27) show field examples of two-pen pressure recorder charts. Explanations are offered on the chart itself; hence, no discussion is given here.

3.6373 Intermittent flow recorder charts

3.63731 Hypothetical charts

Several hypothetical cases are presented to illustrate various operations (good and bad) for intermittent lift.
Figures 3.620(1) and 3.620(2):
Both figures show good intermittent lift utilizing a time cycle controller to inject the gas in cycles. Some of the good characteristics are:
(a) No decrease in gas pressure between cycles.
(b) Reasonably thin peak on casing showing fast valve action.
(c) Good peak on tubing indicating good liquid recovery, although a surface restriction will also appear to be a good liquid head.

Figures 3.620(3) through 3.620(11) show additional hypothetical cases with the explanations included on the charts.

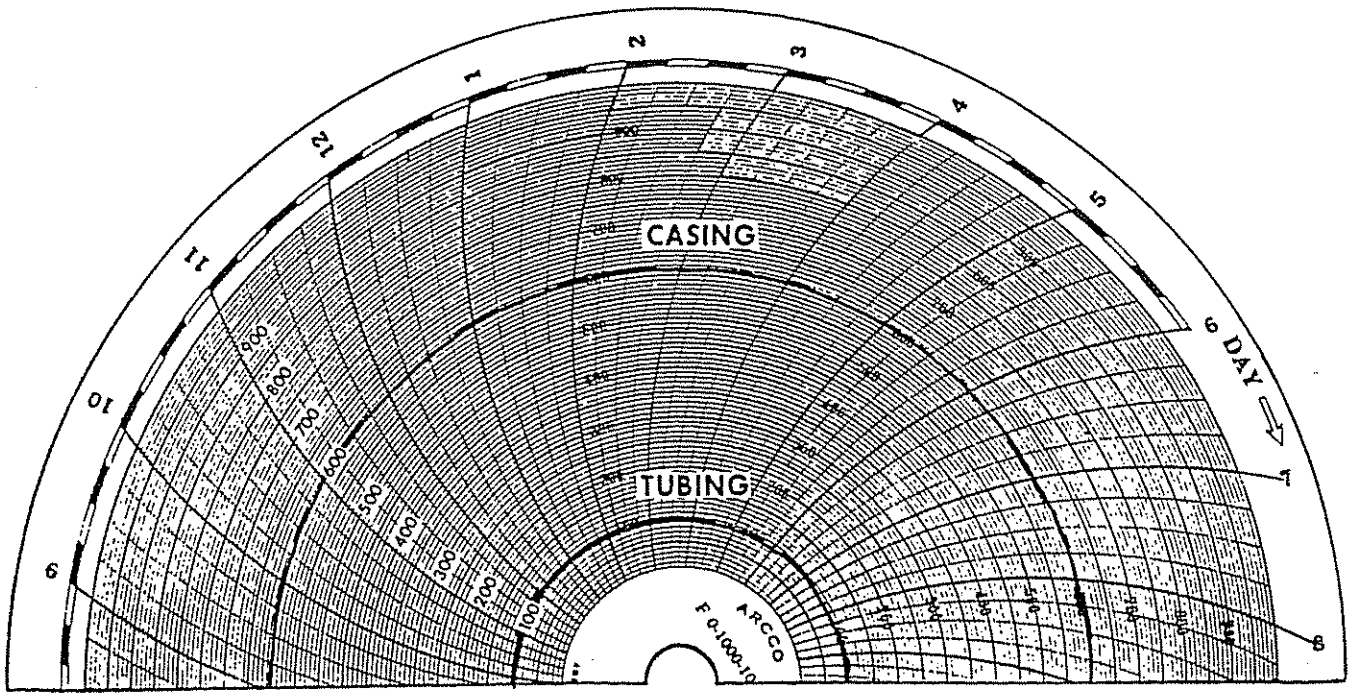


Fig. 3.619 (1) Recorder Chart—Operation: Continuous Flow; Trouble: None; Remarks: Good Operation

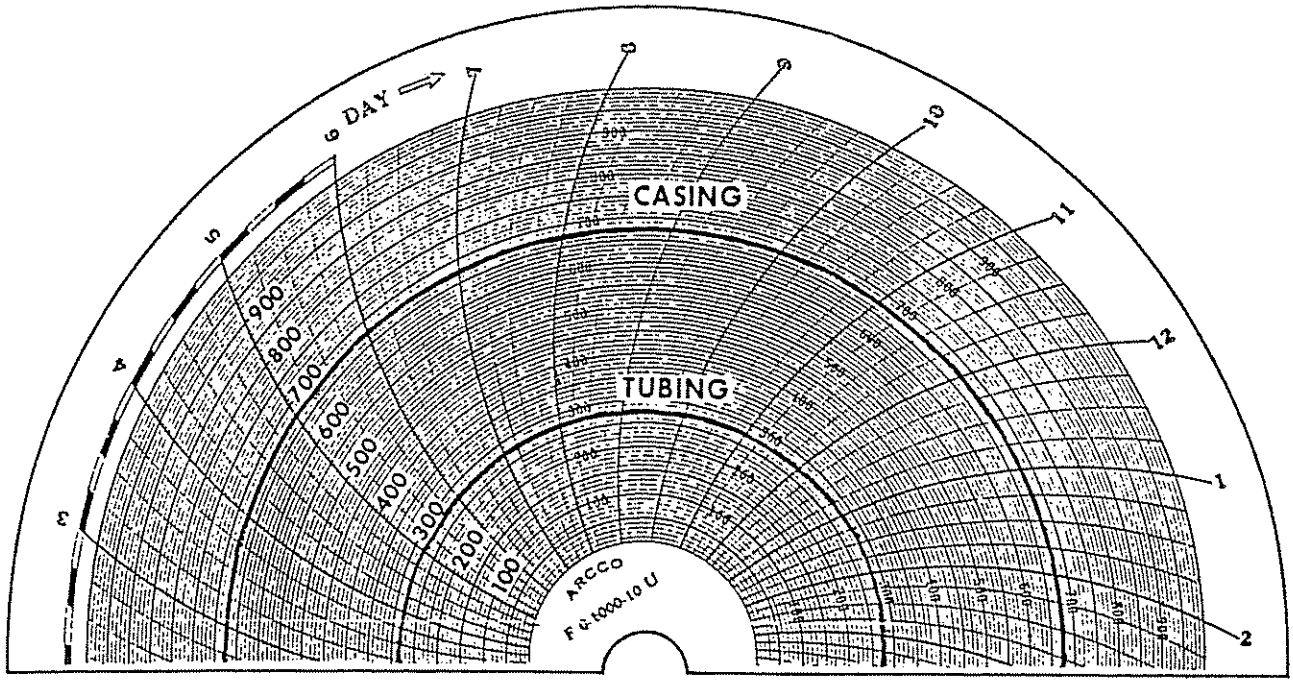


Fig. 3.619 (2) Recorder Chart—Operation: Continuous Flow (400 B/D; Trouble: High Flowing Surface Tubing Pressure; Remarks: With a Tubing Pressure of 275 PSI and a Casing Pressure of 675 PSI, a Differential Lift Pressure of 400 PSI Is All That Is Left; Therefore, Continuous Flow Depth Is Restricted to About 3500 ft. or Less. The Back Pressure Should Be Removed.

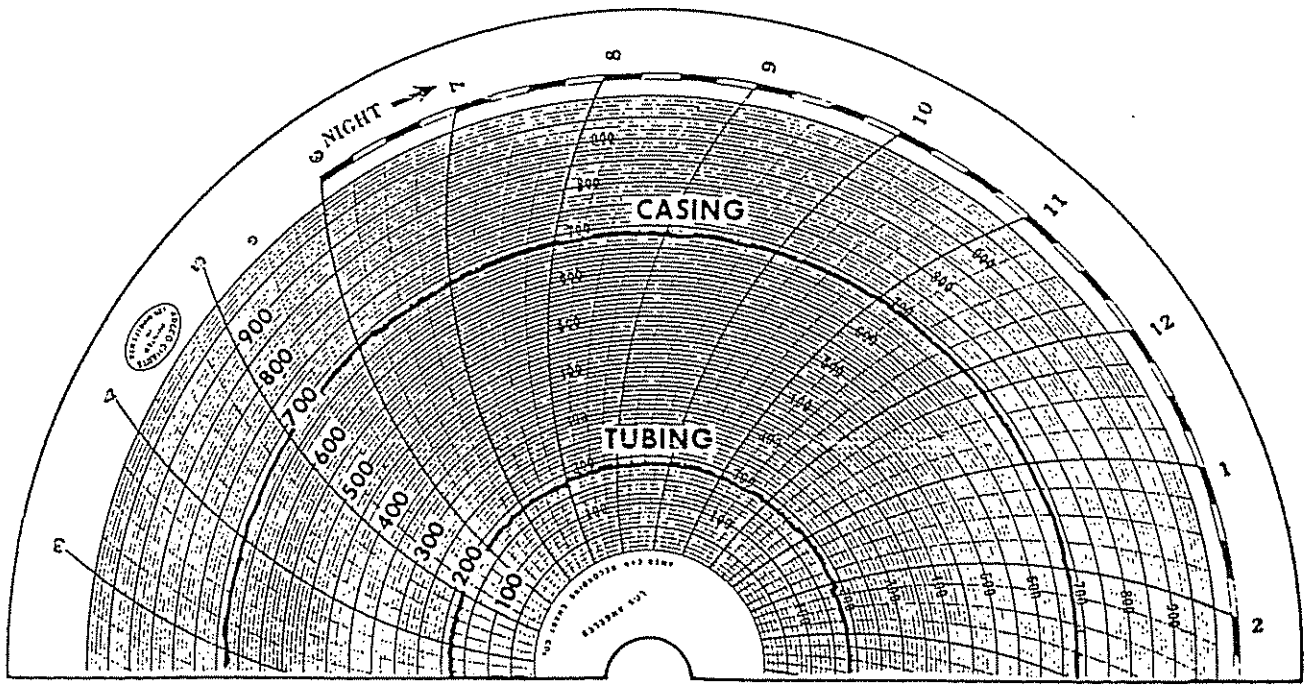


Fig. 3.619 (3) Recorder Chart—Operation: Continuous Flow (640 B/D); Trouble: None; Remarks: This is the Same Well As Noted in Fig. 3.619 (2) Except the Back Pressure Has Been Lowered to 180 PSI from 275 PSI—the Production Increased from 400 B/D to 640 B/D

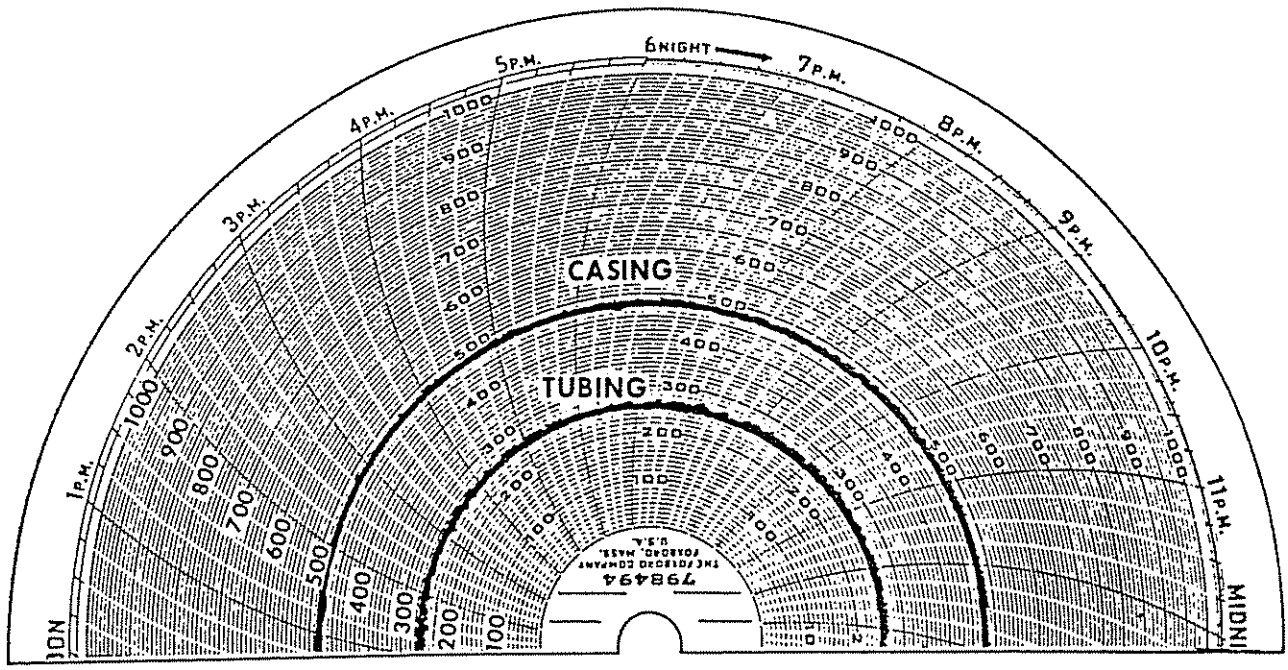


Fig. 3.619 (4) Recorder Chart—Operation: Continuous Flow; Trouble: Low Production; Remarks: System should be investigated for excessive back pressure. Gas injection by choke at surface.

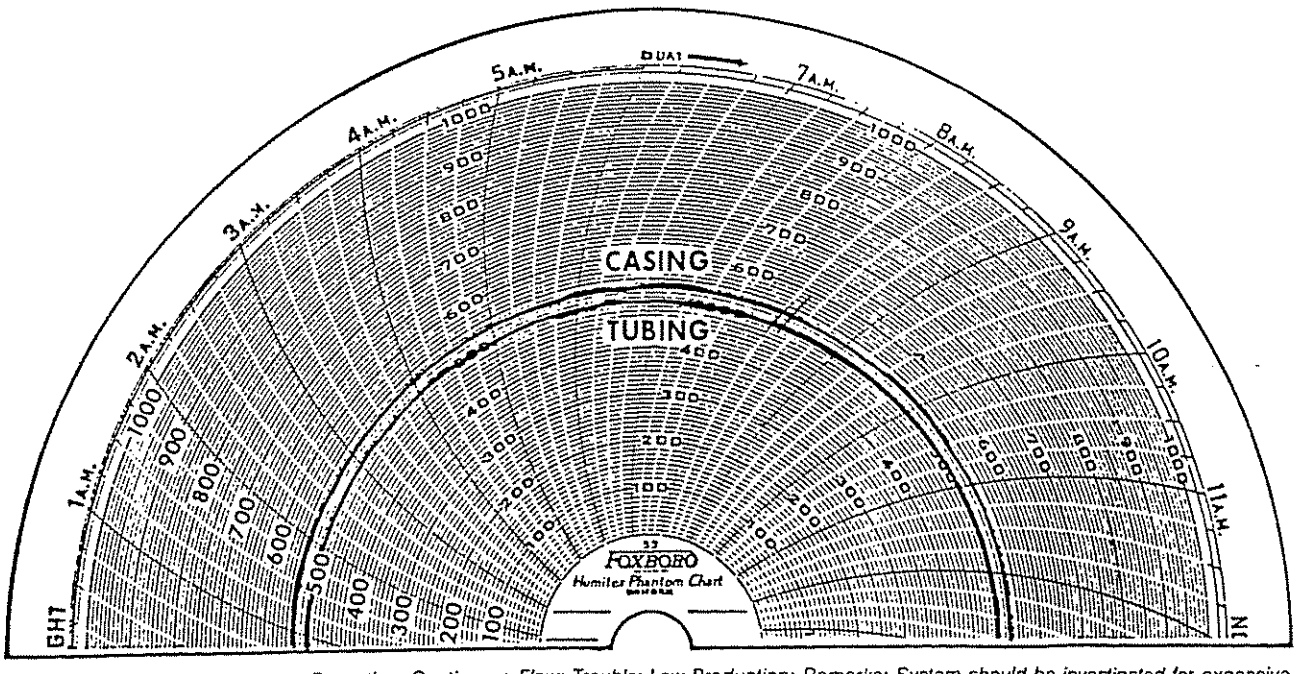


Fig. 3.619 (5) Recorder Chart—Operation: Continuous Flow; Trouble: Low Production; Remarks: System should be investigated for excessive back pressure. Gas injection by choke at surface.

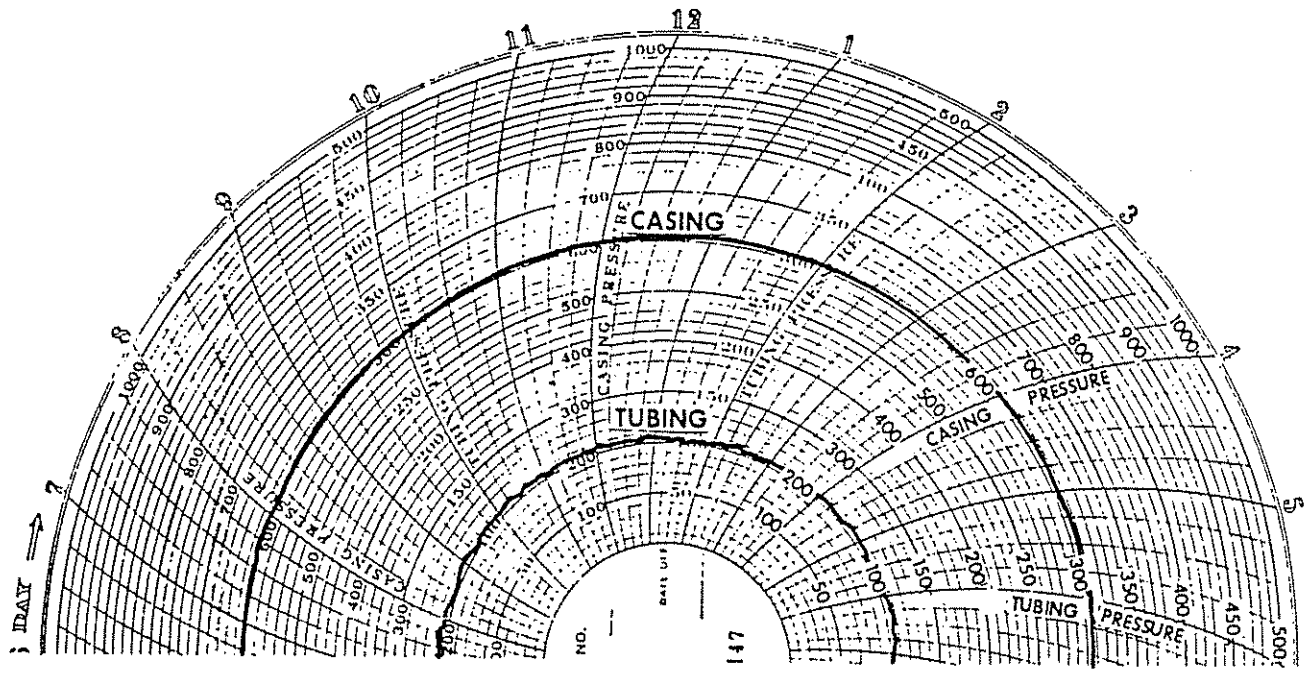


Fig. 3.619 (6) Chart of Continuous Flow Well. Suface Gas Injection Rate Is in Adjustment So That the Pressure Is Constant and the Rate of Production of Fluid Is Satisfactory.

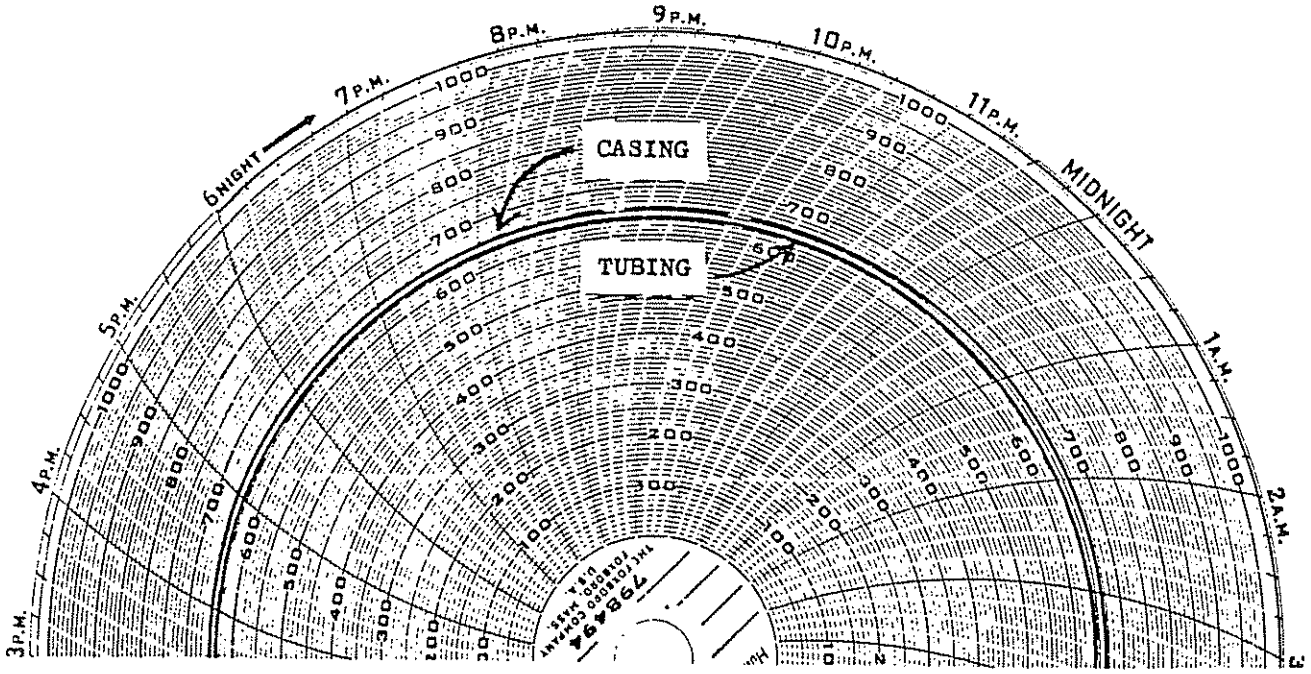


Fig. 3.619 (7) Operation: Continuous Flow, Casing Choke Control. Type of Well: High Productivity, High Bottom-hole Pressure. Trouble: Well Is Closed in. Recommendation: Check to see for what reason it is closed in. Type of valves: Pressure-Operated. Remarks: On checking, it was noted that the well had produced its monthly allowable, and had been closed in. This can hurt some oil wells. It is better to cut the daily production and produce the well constantly.

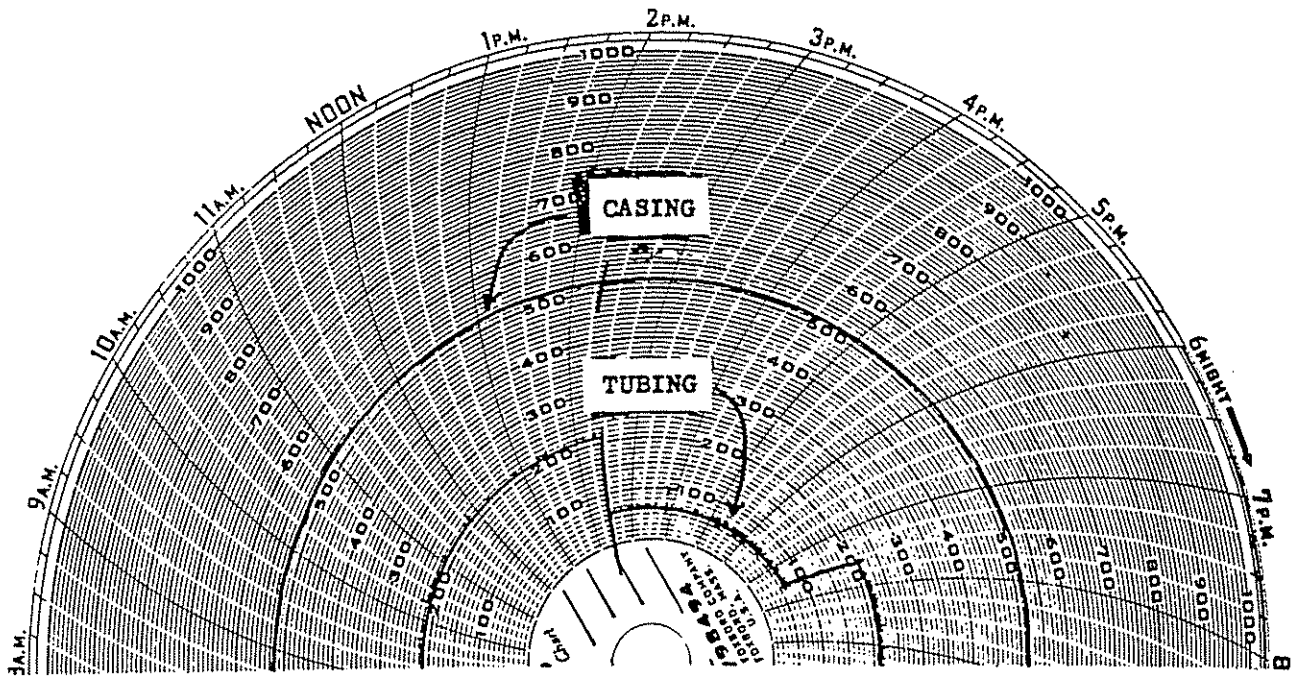


Fig. 3.619 (8) *Operation: Continuous Flow, Casing Choke Control. Type of Well: High Productivity, High Bottom-hole Pressure. Trouble: Well is being tested in test separator. Recommendation: Remove high normal back pressure, or test against same high back pressure for accurate flow test. Remarks: It would be impossible to have an accurate production test on the well under the above conditions.*

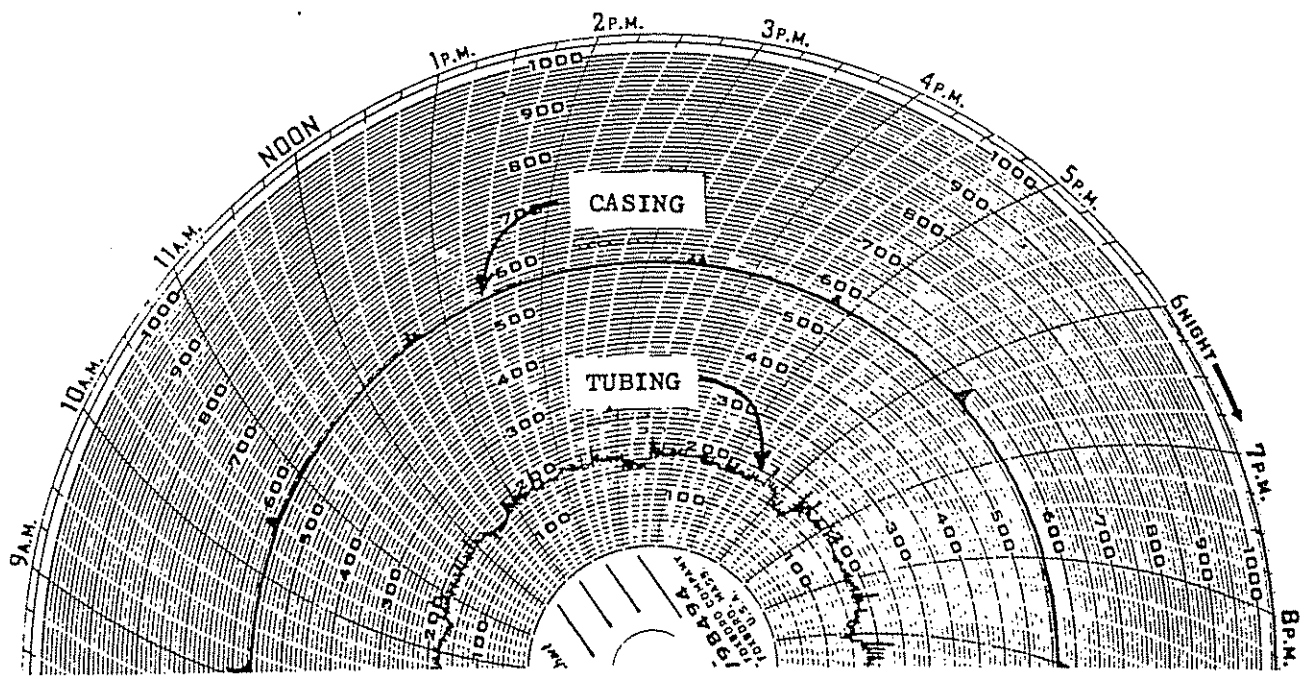


Fig. 3.619 (9) *Operation: Continuous Flow, Tubing Control. Type of Well: High Productivity, High Bottom-hole Pressure. Trouble: Well is flowing, but loads up with water periodically. Recommendation: Operating satisfactory. Remarks: The tubing control element is set to inject gas into the well at 160 psig. It can be noted by the rise in casing pressure opposite the drop in tubing pressure.*

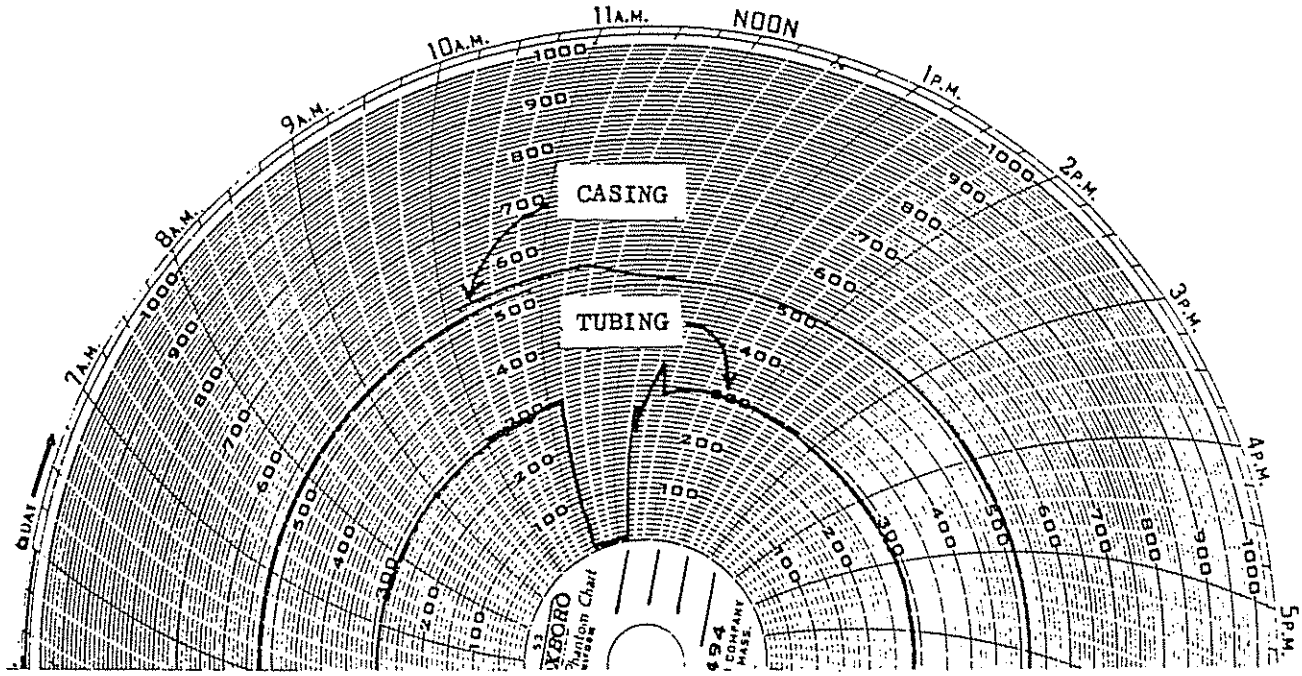


Fig. 3.619 (10) *Operation: Continuous Flow, Casing Choke Control. Type of Well: High Productivity, High Bottom-hole Pressure. Trouble: Well was closed in to repair flow line. Recommendation: Note Type of Gas-Lift Valves: Pressure-Operated. Remarks: When the master valve was opened the tubing pressure was at 250 psig. Flow immediately started but the pressure fell off to 210 psig at the peak of "U"-tube. As the gas cleared through the gas-lift valve the tubing pressure increased to a maximum of 345 psig, then fell off and finally stabilized at 285 psig.*

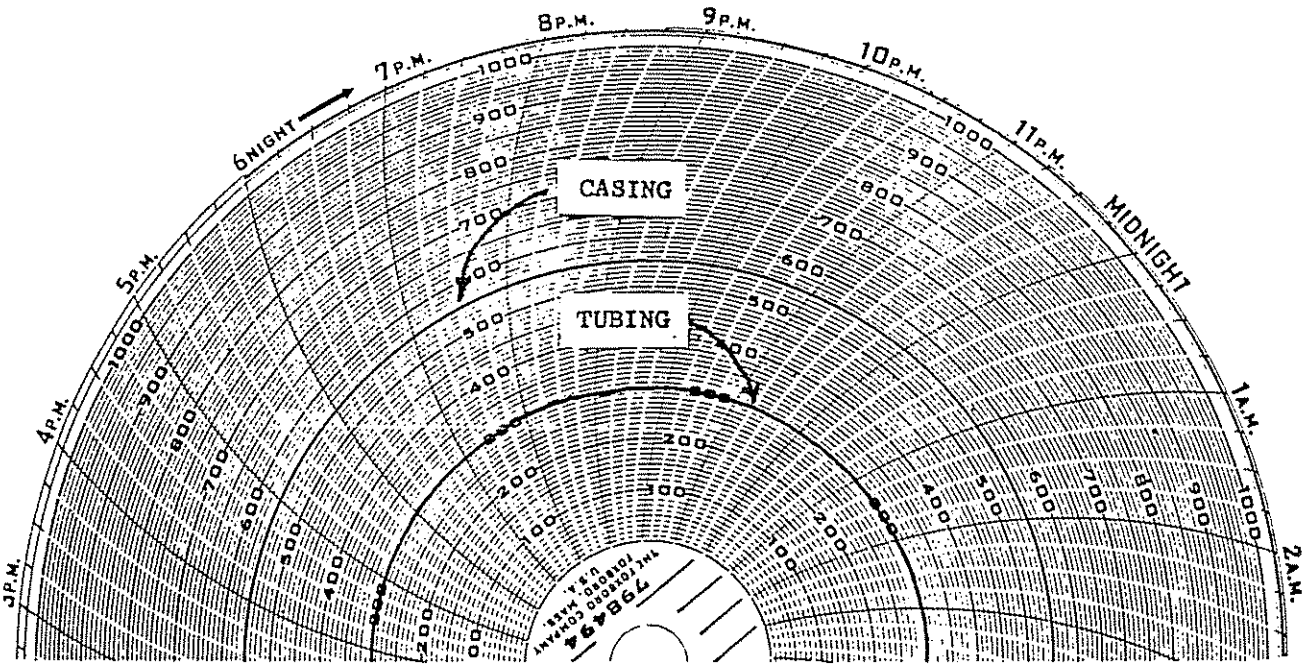


Fig. 3.619 (11) *Operation: Continuous Flow. Type of Well: High Productivity, High Bottom-Hole Pressure. Trouble: None, well is flowing. Recommendation: Leave well alone. Type of Gas-Lift-Valves: Pressure-Operated. Remarks: Well is flowing; no gas is being injected.*

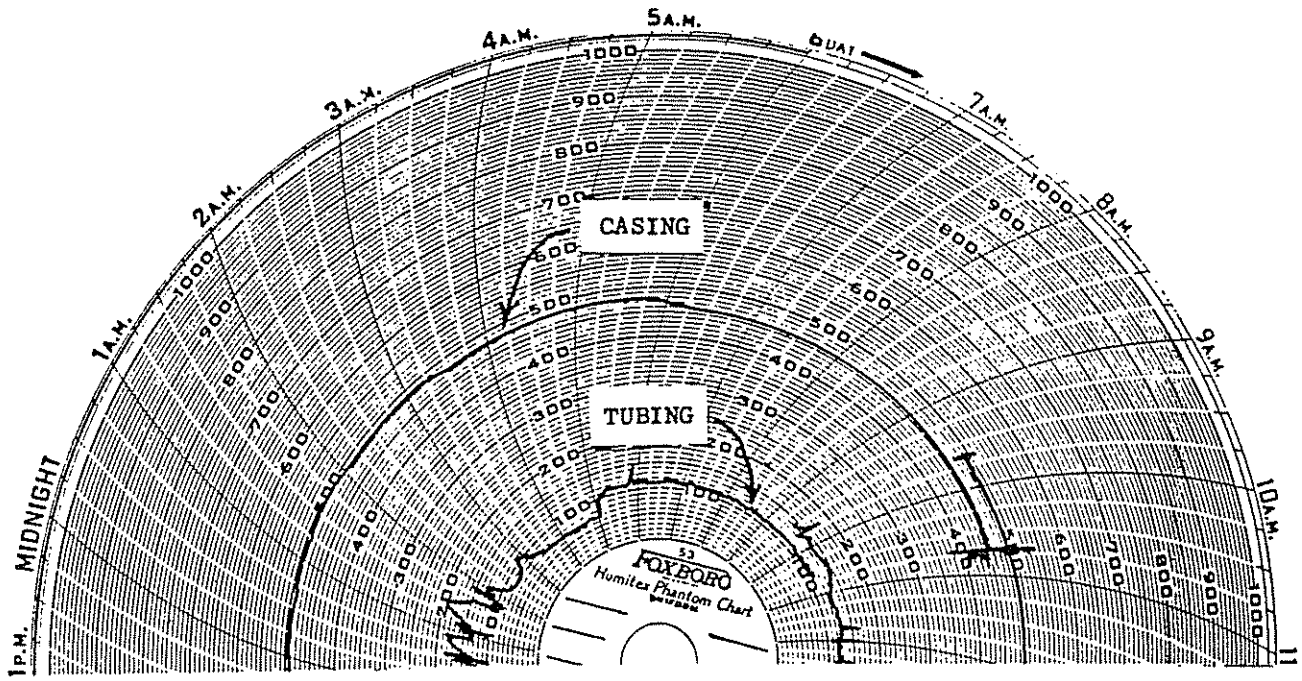


Fig. 3.619 (12) *Operation: Continuous Flow, Casing Choke Control. Type of Well: High Productivity, High Bottom-Hole Pressure. Trouble: Choke on Gas Line Froze. Recommendation: A gas heater might be installed ahead of the choke, or a jacket might be welded around the choke to permit the hot flowline fluids to pass over it, or the well might be placed on intermittent injection. Type of Gas-Lift Valves: Pressure-operated.*

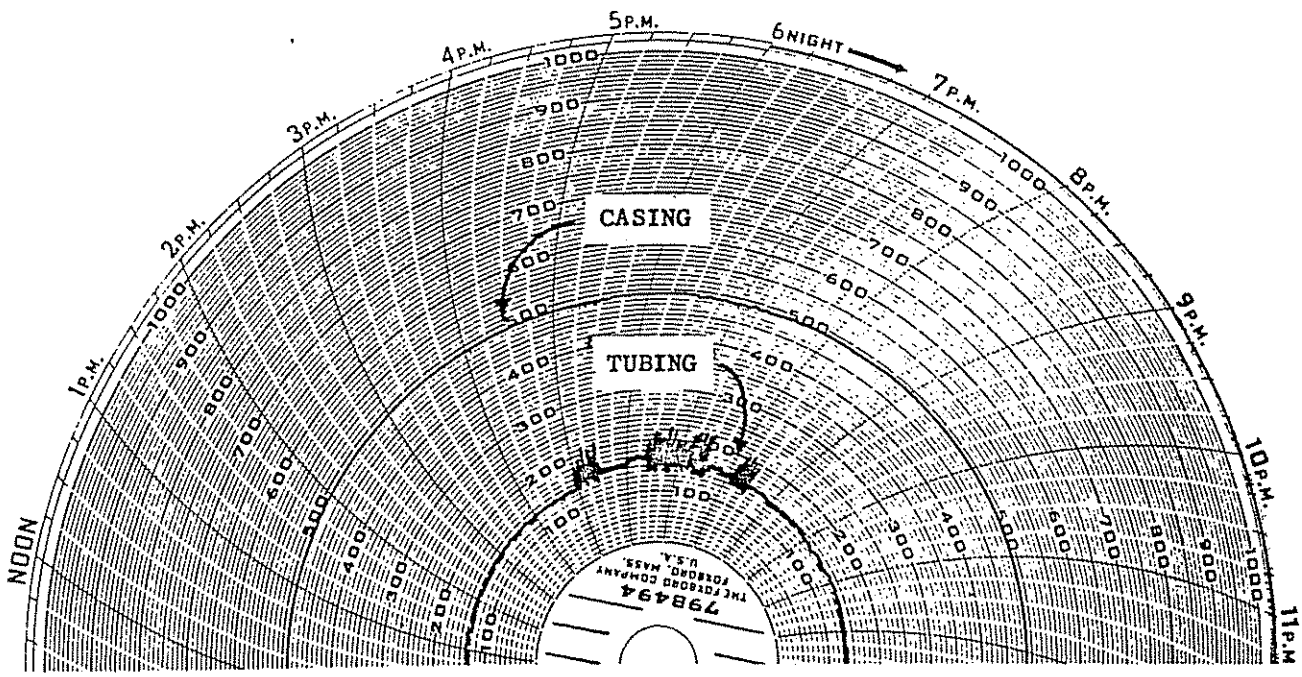


Fig. 3.619 (13) *Operation: Continuous Flow, Casing Choke Control. Type of Well: High Productivity, High Bottom-Hole Pressure. Trouble: Not serious, well is "heading." Recommendation: Check to see if system-gas pressure fluctuates. Type of Gas-Lift Valves: Pressure-operated. Remarks: Reasonably good operation. Well has a tendency to "head," which could be caused by erratic valve operation, or a fluctuating system pressure.*

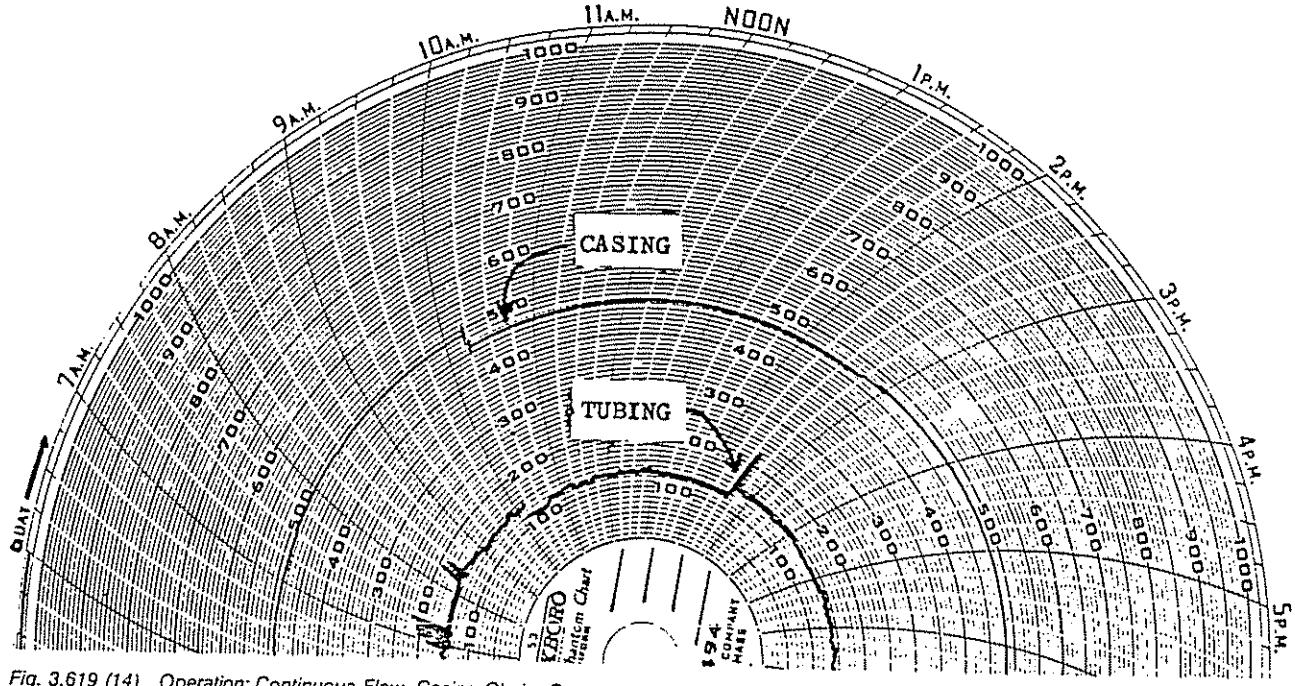


Fig. 3.619 (14) *Operation: Continuous Flow, Casing Choke Control. Type of Well: High Productivity, High Bottom-Hole Pressure. Trouble: None. Recommendation: Leave well alone. Type of Gas-Lift Valves: Pressure-operated. Remarks: The well had been shut in overnight, and the gas had been turned on shortly before the chart was changed. The casing pressure was at 460 psig at the beginning at 10:15 a.m. There was a gradual pressure rise to 468 psig due to fluid temperature increase affecting valve. At 2:45 p.m. the casing pressure increased to 480 psig and a "kick" can be noted on the tubing pressure. This was due to an upper valve becoming the operating valve. At 10:00 a.m. the next morning the casing pressure had increased to 490 psig due to temperature effect.*

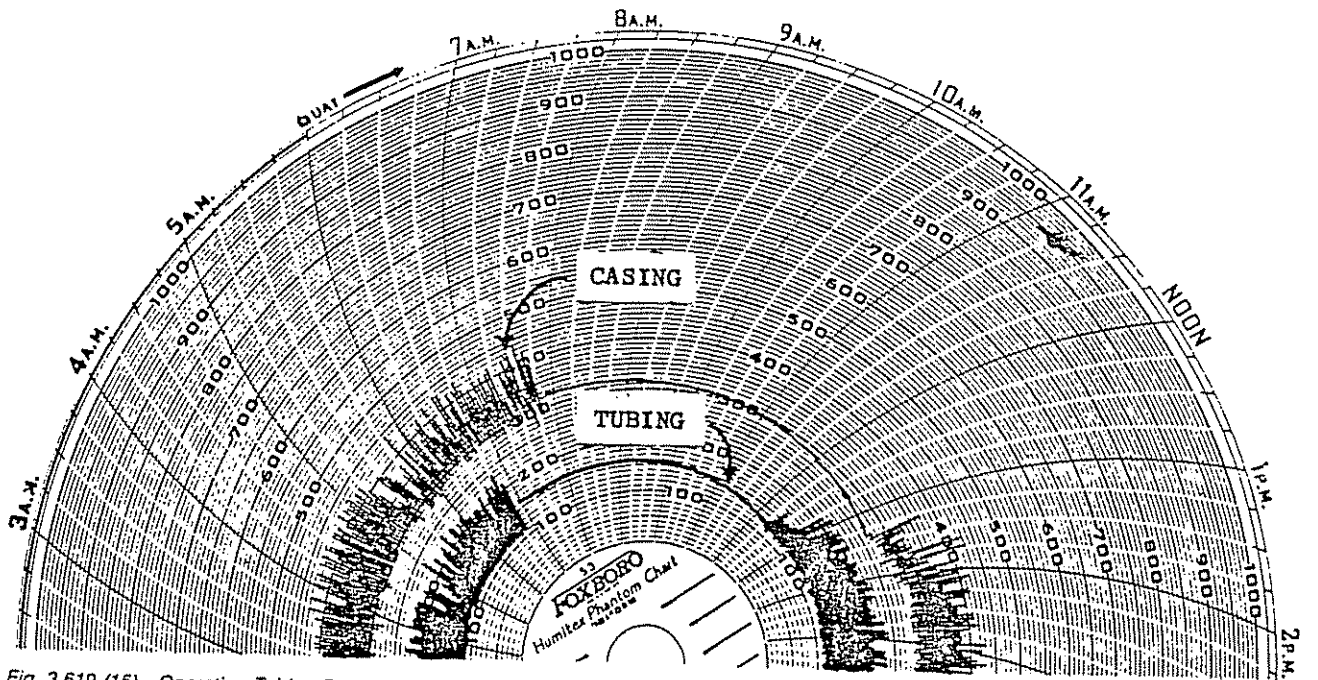


Fig. 3.619 (15) *Operation: Tubing Control without Choke in Gas Line. Type of Well: High Productivity, High Bottom-Hole Pressure. Trouble: Supply gas was closed to well. Well was off production 5½ hours. Type of Gas-Lift Valves: Pressure-operated. Remarks: Work was being done on the gas system. Lift might be improved by adding a choke in gas line.*

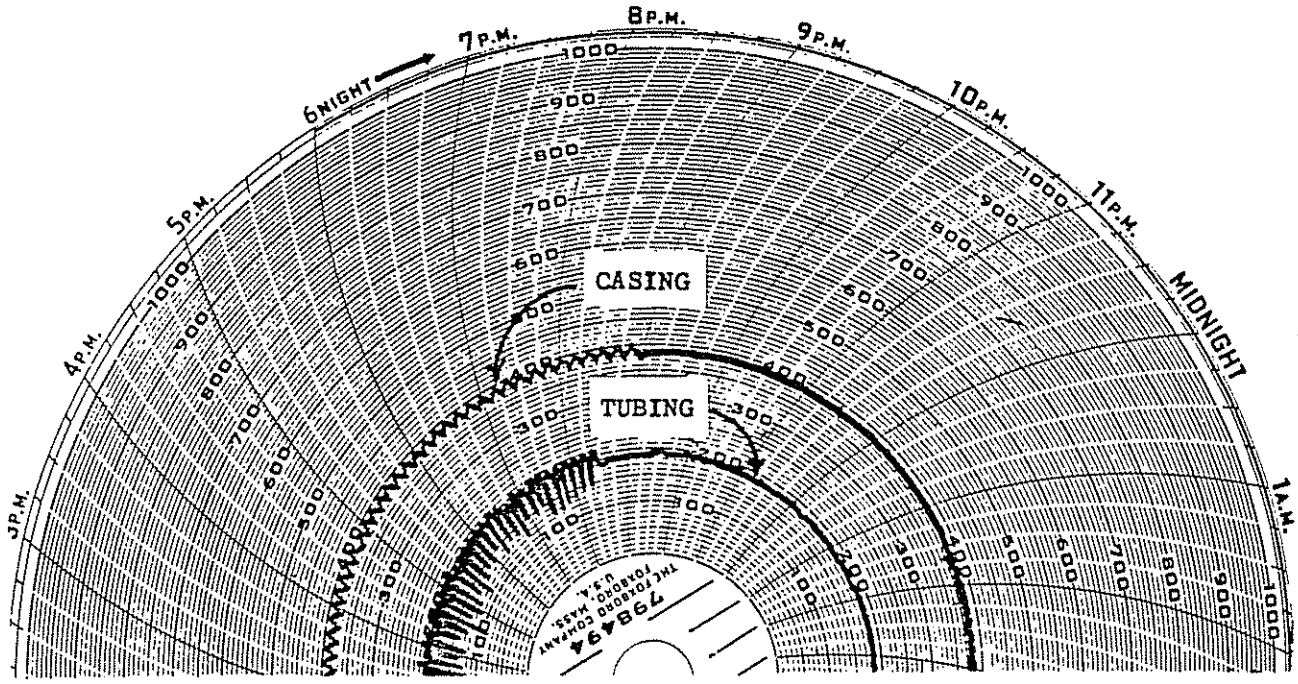


Fig. 3.619 (16) *Operation: Intermittent Injection vs. Continuous Injection. Type of Well: Borderline Production Rate. Trouble: Inadequate production. Type of Gas Lift Valves: Pressure-operated. Remarks: Comparison of intermittent to continuous flow to determine most efficient production rate. Recommendations: An intermittent and continuous flow production comparison.*

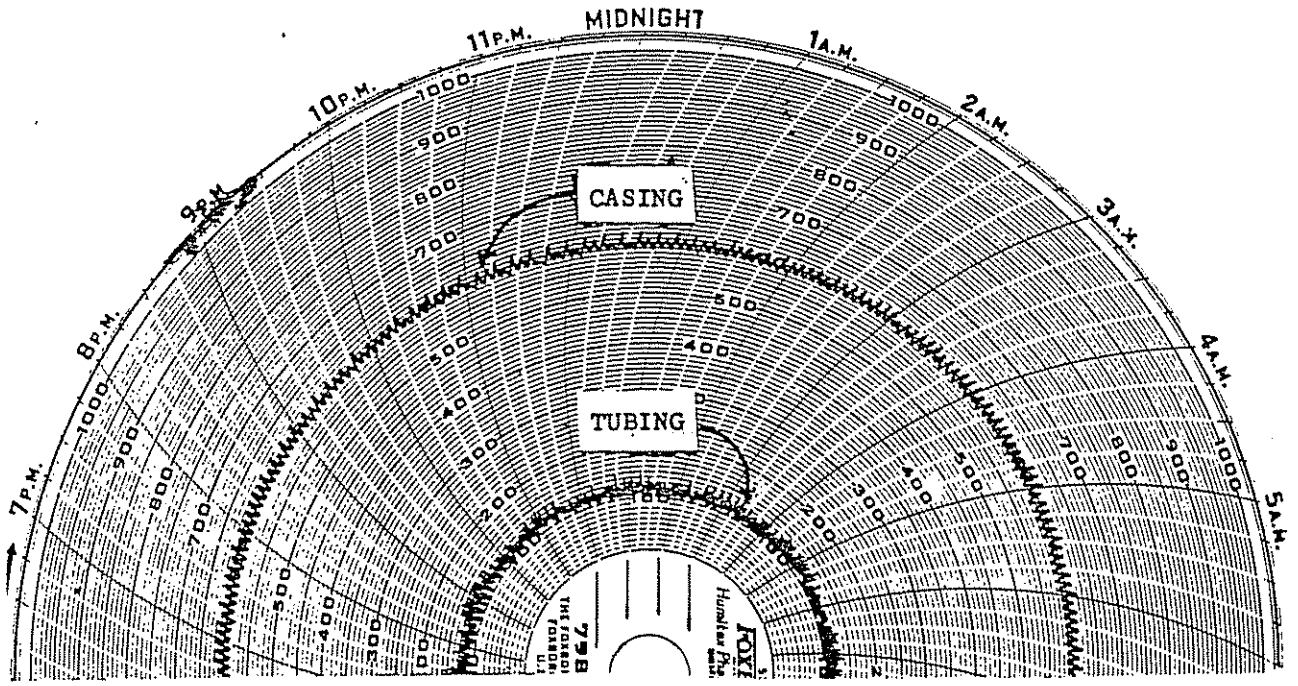


Fig. 3.619 (17) *Operation: Intermittent Continuous Flow, Time Cycle Control. Type of Well: High Productivity, High Bottom-Hole Pressure. Trouble: None. Recommendations: Leave well alone. Type of Gas Lift Valves: Pressure intermittent valves (choked). Remarks: Well is injecting 30 seconds every 5 minutes. Well was first placed on Tubing Control, the fluid was heavily emulsified and the well was not making its production. After placing it on Time Cycle Control the emulsification stopped and the well made its production.*

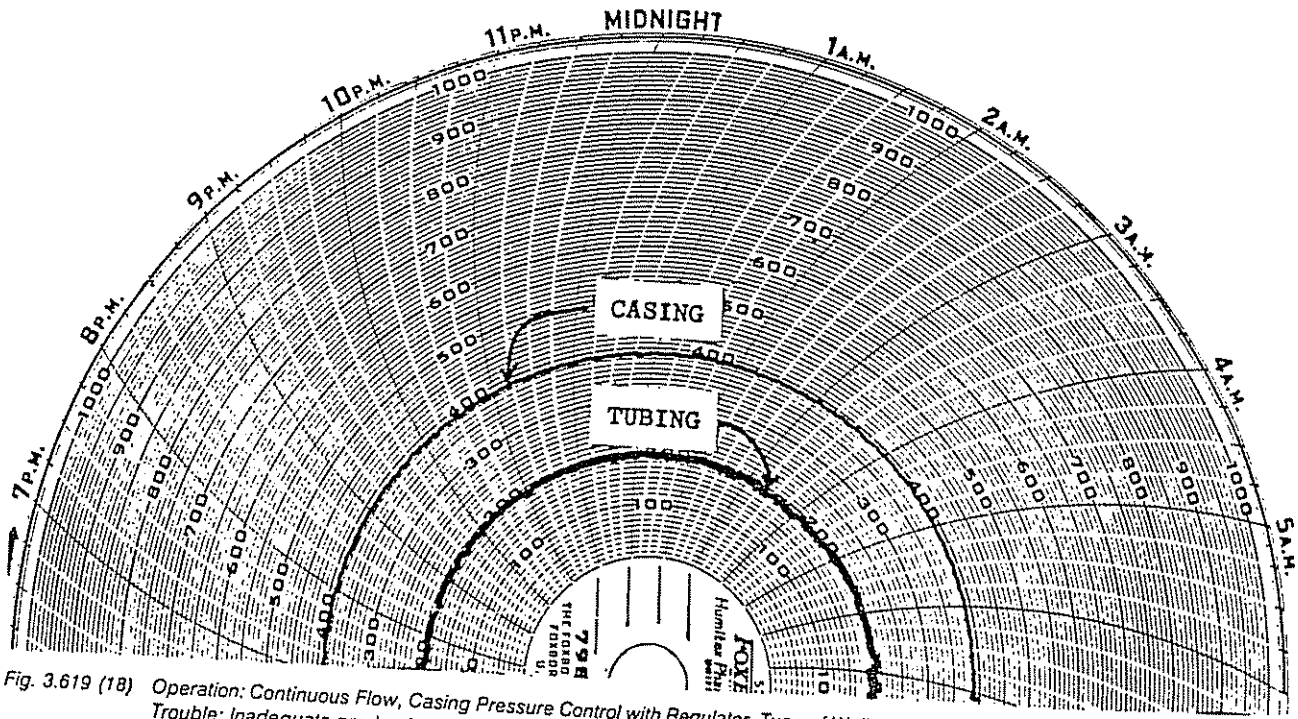


Fig. 3.619 (18) *Operation: Continuous Flow, Casing Pressure Control with Regulator. Type of Well: High Productivity, High Bottom-Hole Pressure. Trouble: Inadequate production. Recommendations: Reduce back pressure. Type of Gas Lift Valves: Pressure operated valves. Remarks: Excessive back pressure may be due to one or more of the following: (1) Choke in flow line; (2) Restriction in flow line (paraffin, sand, mashed, etc.); (3) Flow line too small or too long; (4) Separator pressure too high; (5) Too many sharp bends in flow line; (6) Highly emulsified fluid; (7) Excessive input gas.*

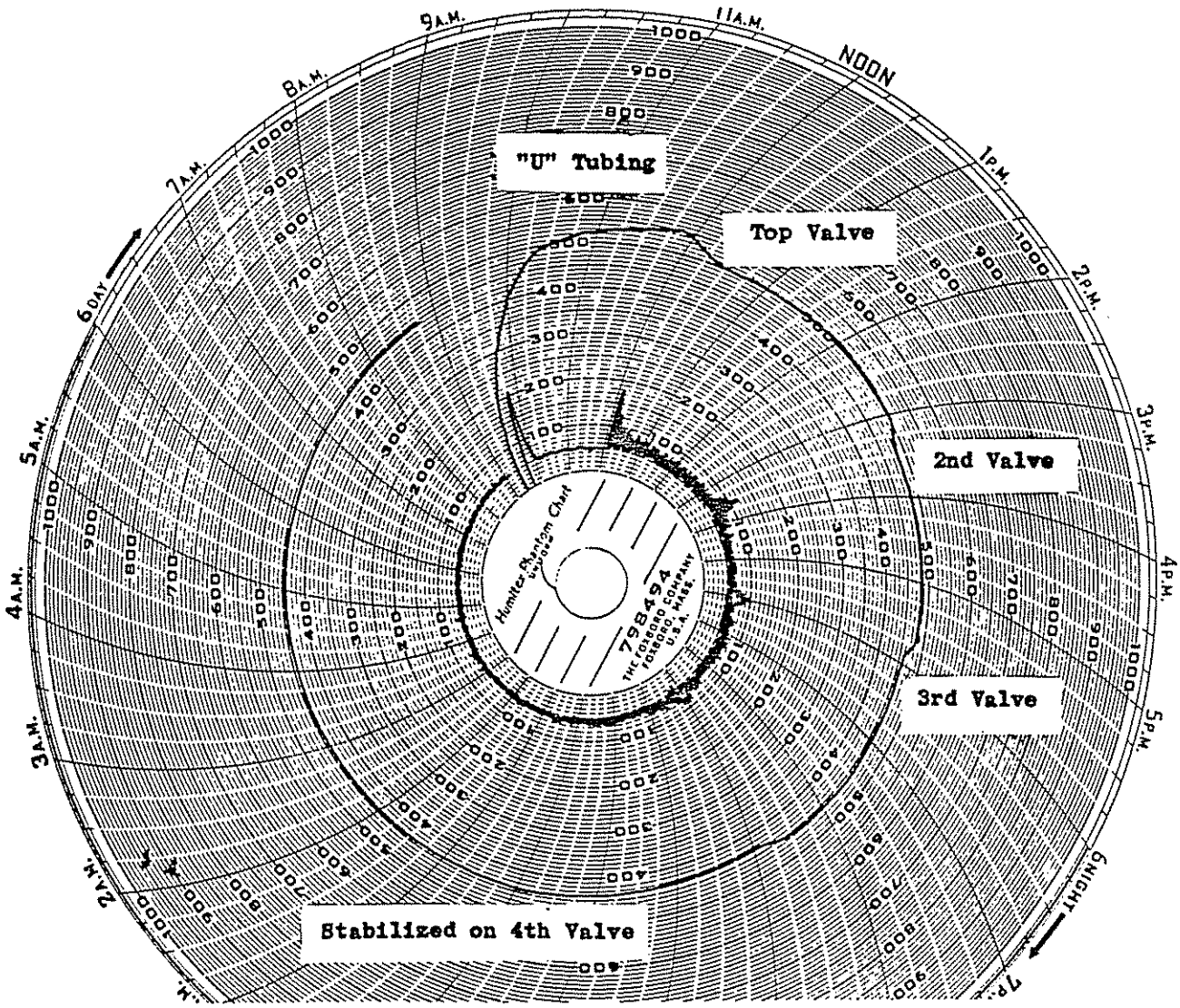


Fig. 3.619 (19) *Unloading Continuous-Flow Well. A choke was used on the gas line to control the gas volume into the casing-tubing annulus. When the gas was first turned on, an immediate surge of fluid returned from the tubing as the well was completely full of salt water. When the liquid volume displaced in the annulus stabilized to the gas-volume rate of the injection gas, the tubing pressure remained at 50 psig until the top valve was uncovered and gas entered the tubing. A surge in tubing pressure is noted as each valve is uncovered. The well finally stabilized on the 4th valve.*

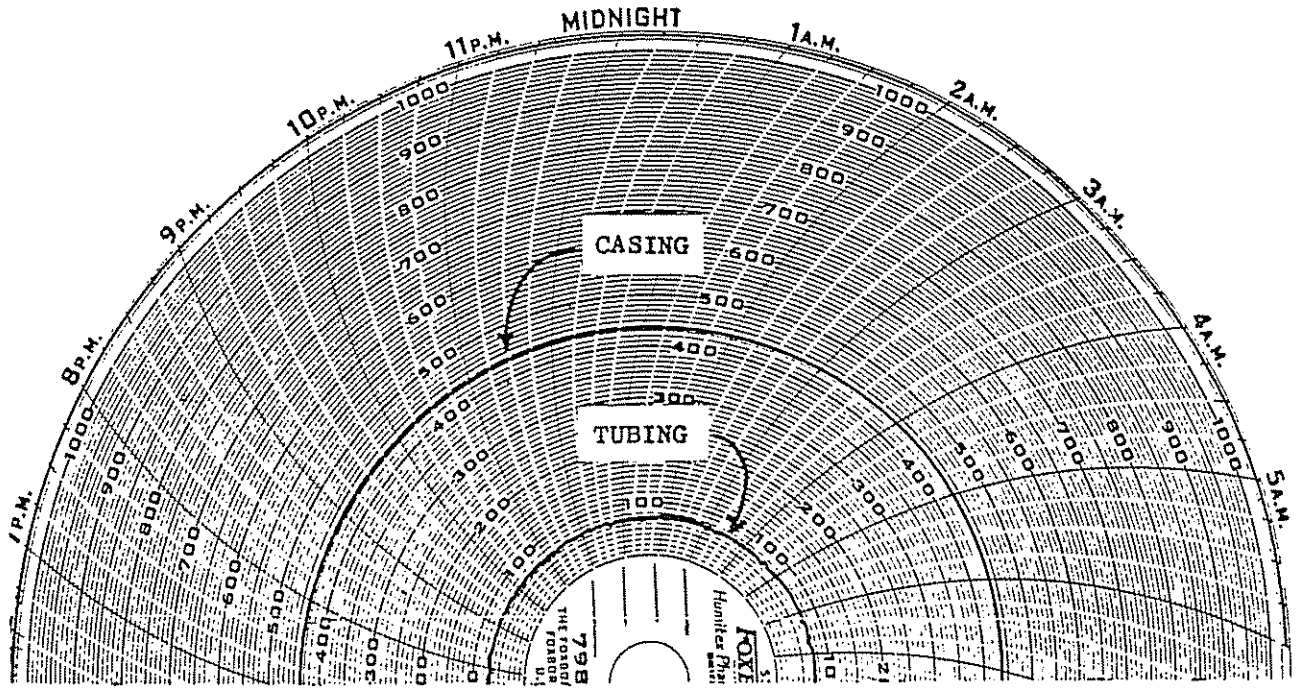


Fig. 3.619 (20) *Operation: Continuous Flow, Casing Choke Control. Type of Well: High Productivity, High Bottom-Hole Pressure. Trouble: None. Recommendation: Leave well alone. Type of Gas Lift Valves: Pressure-operated valves. Remarks: Good continuous flow operation. Well has a high working fluid level. Note the low back pressure effect. Well producing 2100 bbls. of fluid per day, 95% water, from water drive reservoir, through 2½-inch tubing.*

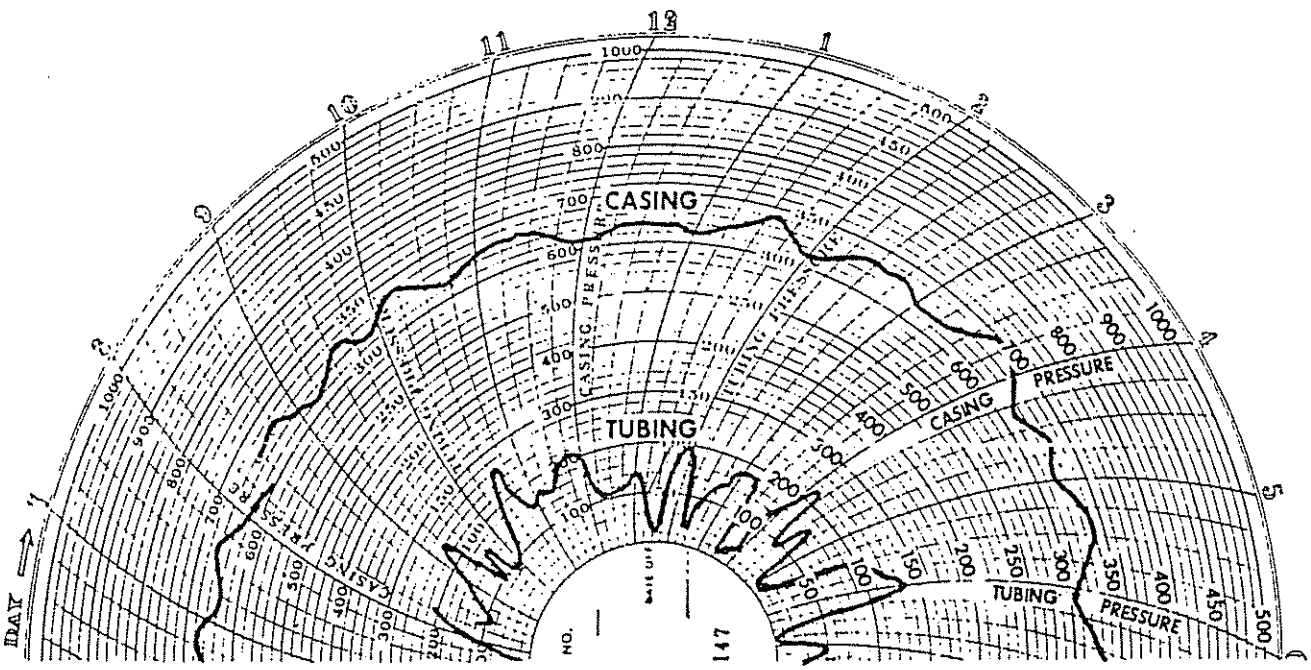


Fig. 3.619 (21) *Continuous-Flow Chart. Indication: Erratic condition of flow indicated by wide variations in the pressure on both the casing and the tubing. If this well is capable of producing sufficient fluid to maintain a steady gradient in the tubing, adjustment of volume and/or pressure of input gas should correct this situation.*

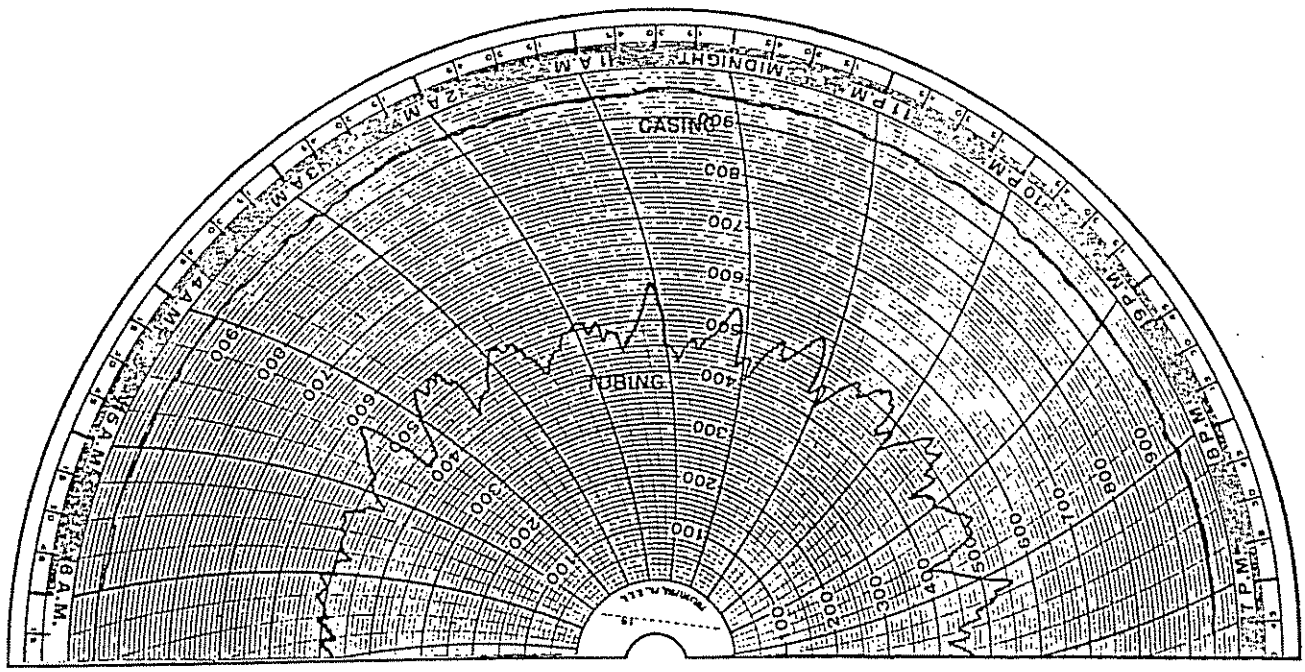


Fig. 3.619 (22) *Recorder Chart—Operation: continuous flow, choke control of injection gas. Trouble: high tubing pressure and varying system pressure. Remarks: Investigate for long flowline and reason for a possible varying pressure at the separator—possibly trying to continuous flow from intermittent valve.*

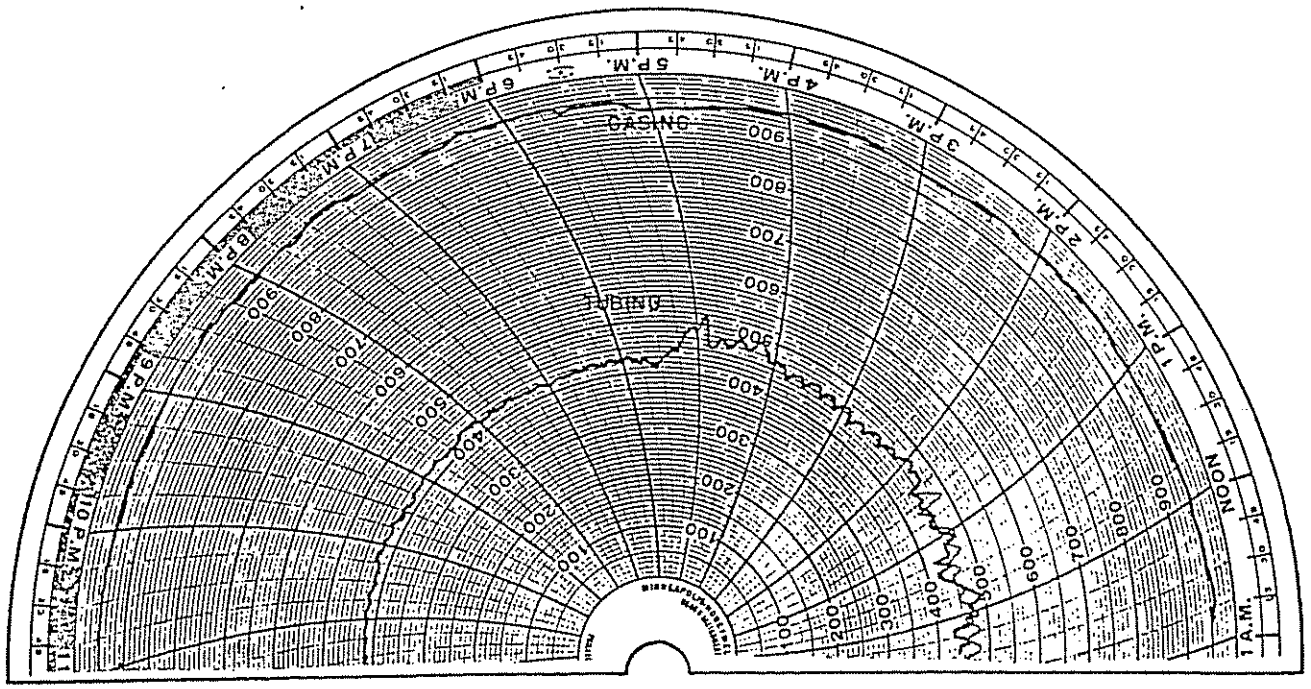


Fig. 3.619 (23) *Recorder Chart—Operation: continuous flow. Trouble: high tubing pressure causing low production of 118 BID from 9300 ft. Remarks: Should remove back pressure for greater production—increased tubing pressure shows an intermittent lift well being placed in the common separator system.*

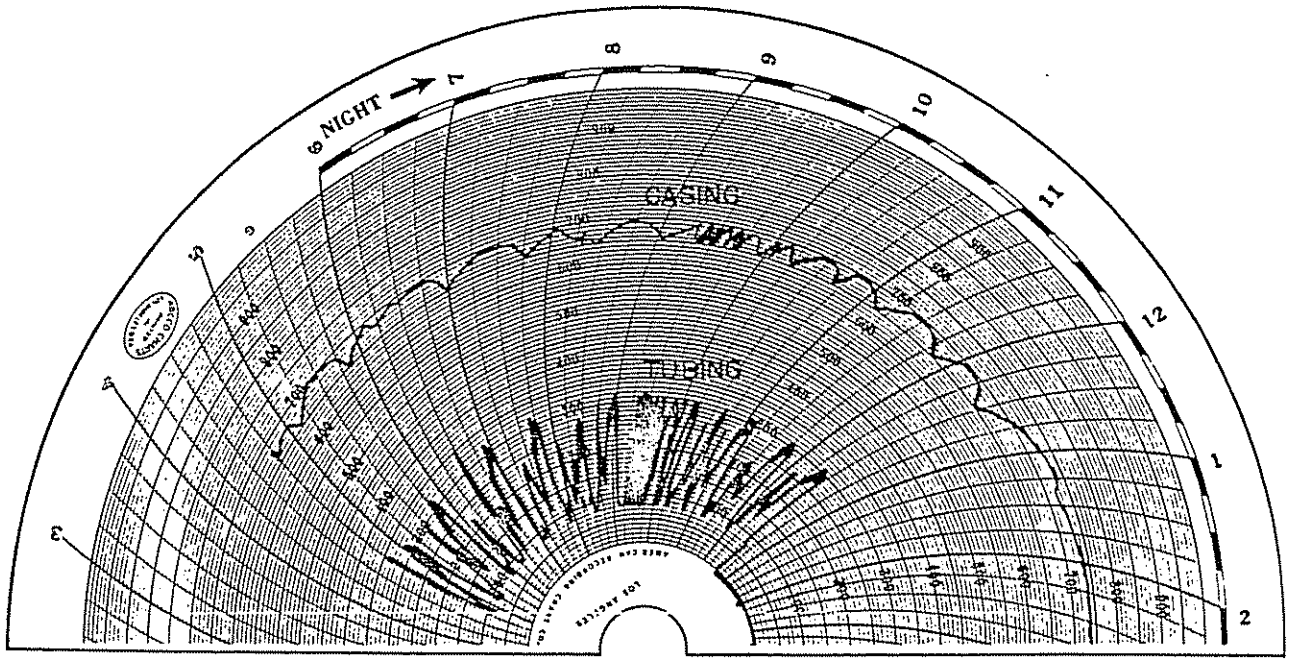


Fig. 3.619 (24) Recorder Chart—Operation: continuous flow (500 B/D; GLR = 400/1; ½-in. surface choke). Trouble: Variable surface injection pressure and trying to continuous flow with intermittent valve. Remarks: The surface line pressure should be stabilized and the intermittent valves replaced with continuous flow valves.

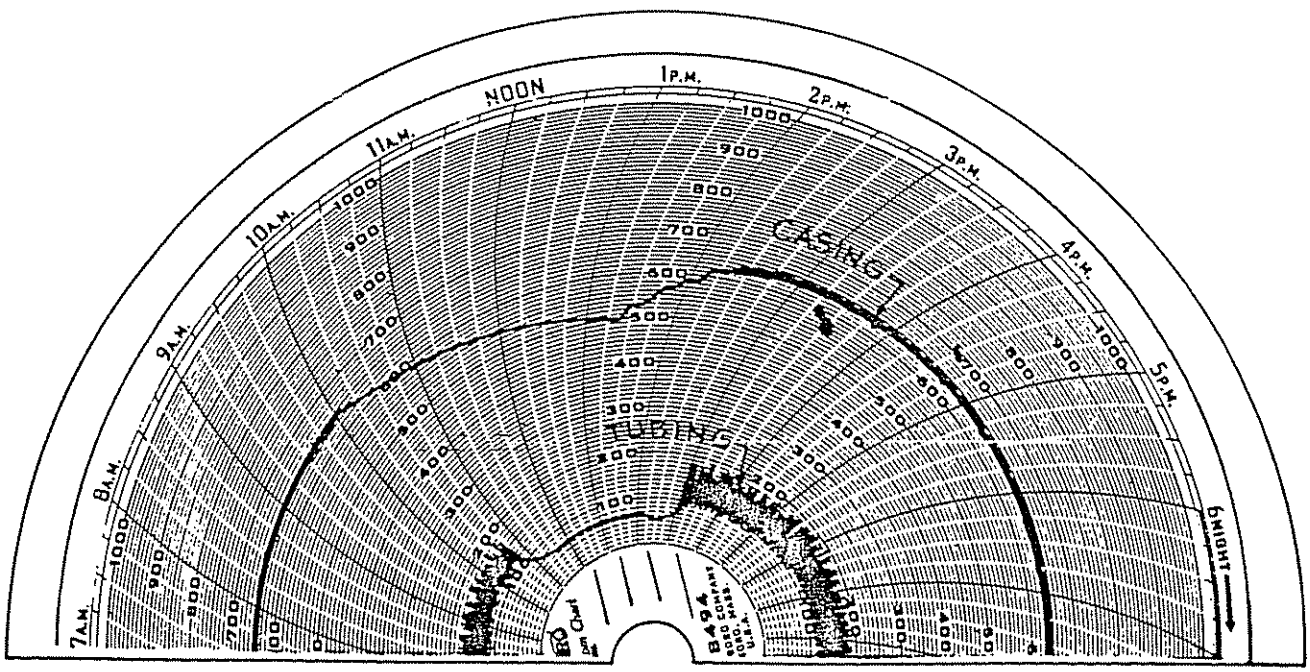


Fig. 3.619 (25) Recorder Chart—Operation: continuous flow, tubing-controlled injection. Trouble: Freezing in injection gas line. Remarks: Freezing of injection gas line and subsequent lowering of casing pressure indicated a leaking valve (not serious). Line thawed out and well again stabilized.

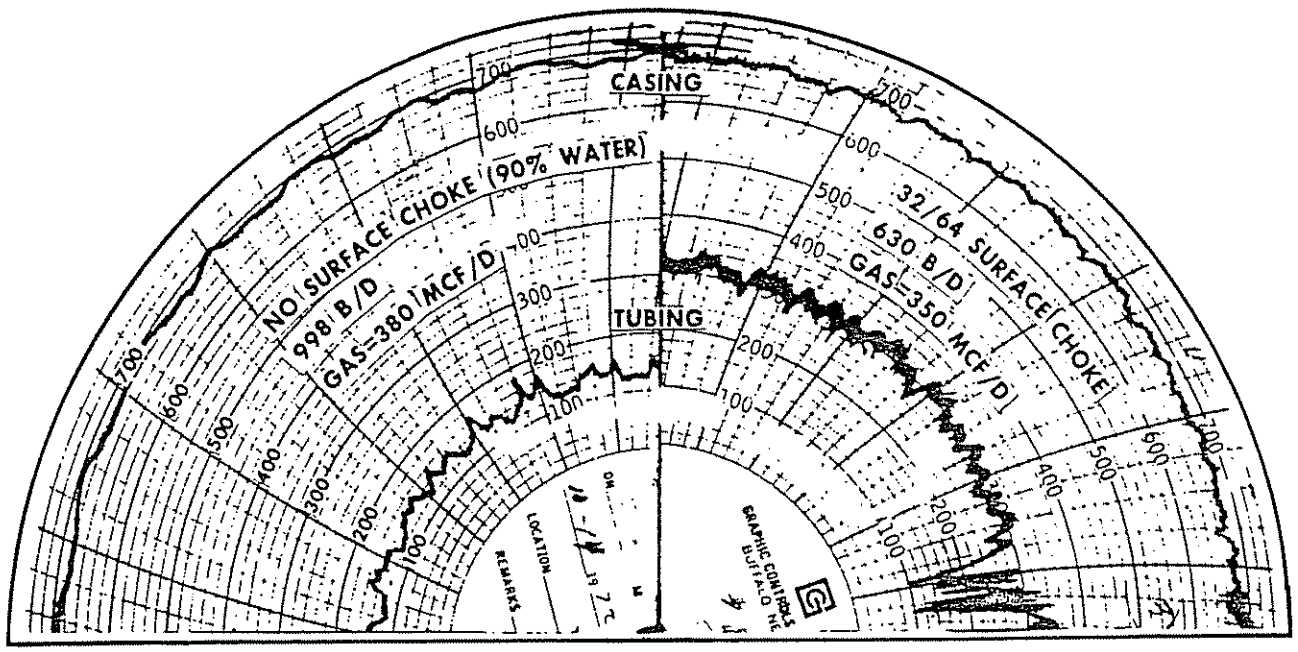


Fig. 3.619 (26) *Effect of Surface Choke for Continuous Flow. Well produces 998 B/D with no choke and 630 B/D with a ½-in. choke at the surface.*

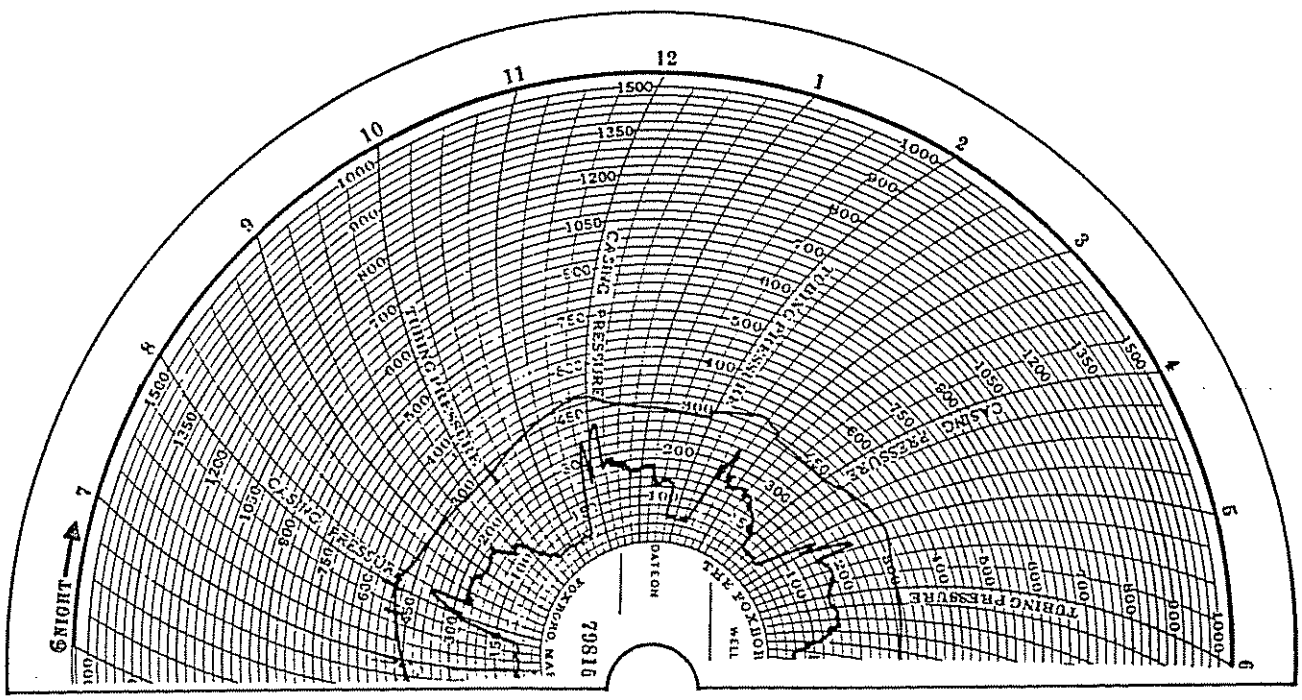


Fig. 3.619 (27) *Recorder Chart—Operation: continuous flow. Trouble: Trying to continuous flow from choked fluid-operated valve; gas lift valve opening and closing giving intermittent operation. Remarks: Increase gas volume and try to lower another valve.*

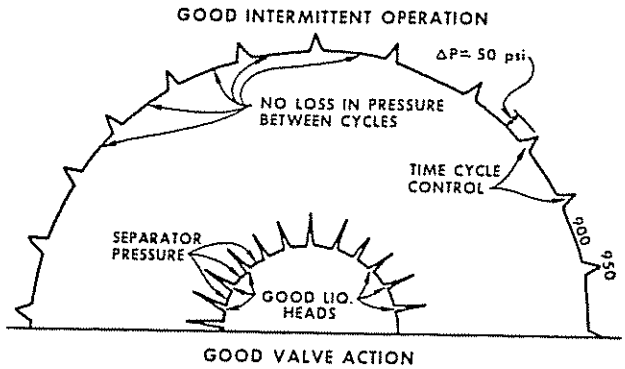


Fig. 3.620 (1) Recorder chart Showing Characteristics of Good Intermittent Operation

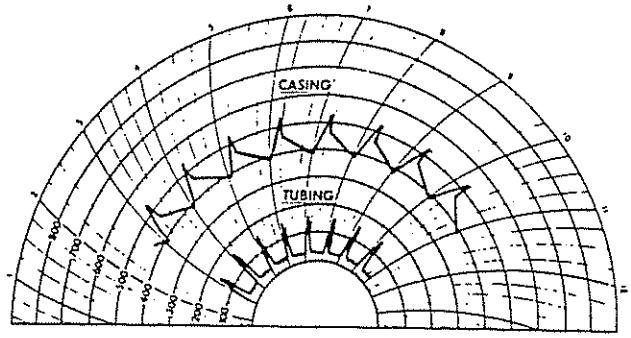


Fig. 3.620 (4) Intermittent Flow - Leak. The leak is evident from the casing pressure. It continues to draw down until the next injection cycle begins. The low thick tubing kicks are indications of gas being injected into the tubing through an upper valve.

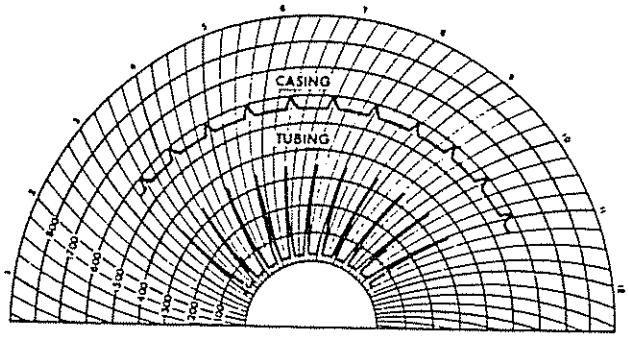


Fig. 3.620 (2) Intermittent Flow—Good Operation. Note on the casing pressure recording the rapid build-up and drawdown and the constant pressure between cycles. This is characteristic of good valve operation. Note on the tubing pressure recording the thin sharp kicks, and the low tubing back pressure. These are characteristic of good slug recovery at the surface.

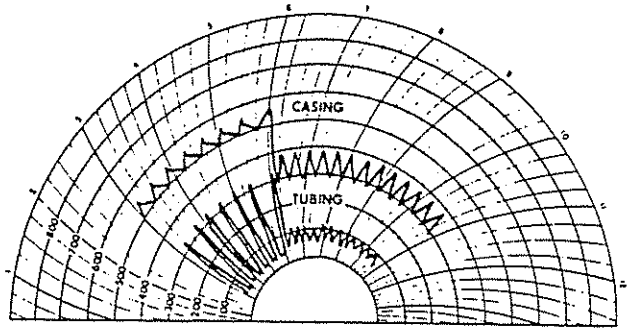


Fig. 3.620 (5) Intermittent Flow - Valve Open. The chart recording between 3:00 and 6:00 shows fair operation as evidenced by the tubing pressure recording. A leak in the string is evident during this time, however, by the continued decrease in casing pressure between cycles. At 6 o'clock a valve began sticking open as shown by the sharp drop in casing pressure, the lack of any valve action thereafter, and the lack of fluid recovery as shown by the tubing pressure recording. Sand, drilling mud, or trash in the annulus could cause this condition.

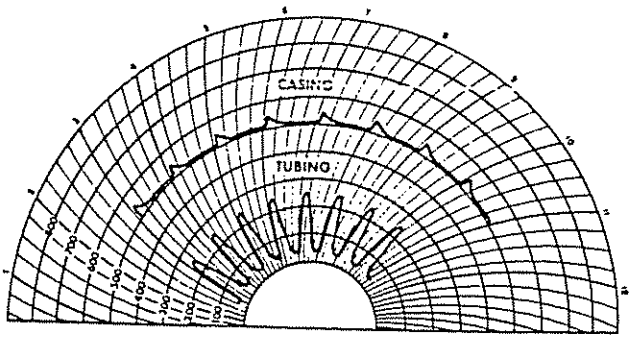


Fig. 3.620 (3) Slow Gas-Passage Rate Gas is passing through the valve at a low rate as indicated by a wide peak.

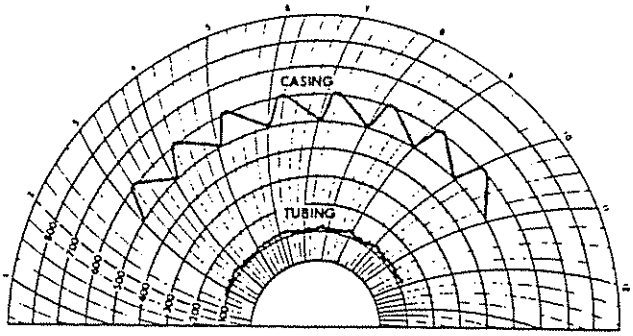


Fig. 3.620 (6) Intermittent Flow - Leak. The saw tooth shape of the casing pressure recording as well as the relatively flat tubing pressure indicates a complete lack of valve operation, and a serious leak in the tubing string. The leak may be in the valve due to trash or cut out seat, or it may be in the tubing or mandrel due to corrosion or erosion.

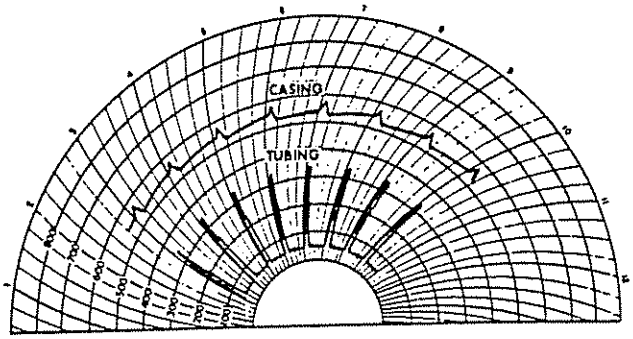


Fig. 3.620 (7) *Intermittent Flow - Leak. A leak in the surface controller is evident from the continual increase in casing pressure between injection cycles. The leak is not yet too serious as evidenced by the good valve action and good slug recovery. The leak can be corrected by replacing the controller seat.*

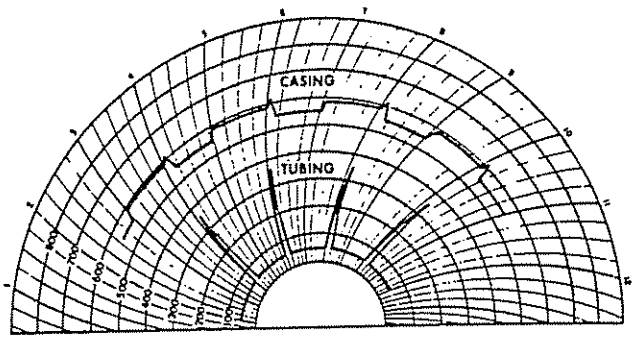


Fig. 3.620 (8) *Intermittent Flow—Improper Controller Setting. An improper controller setting is evident from the casing and tubing recordings. The injection gas shuts off before the casing pressure can build up to the opening pressure of the operating valve. Good valve action and slug recovery are noted when the well does kick. This can be corrected by properly adjusting the time interval between cycles and the duration of each gas injection period.*

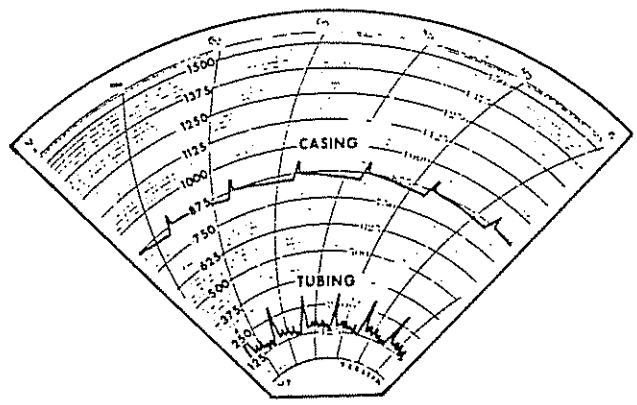


Fig. 3.620 (9) *Surface Controller Leaking. Indication: (1) Casing pressure drops after valve is closed; (2) Tubing pressure still shows decided peak but does not return to separator pressure; (3) Leak is below fluid feed-in level since tubing pressure indicates attempted heading between cycles. Remedy: Retrieve and check operating valve.*

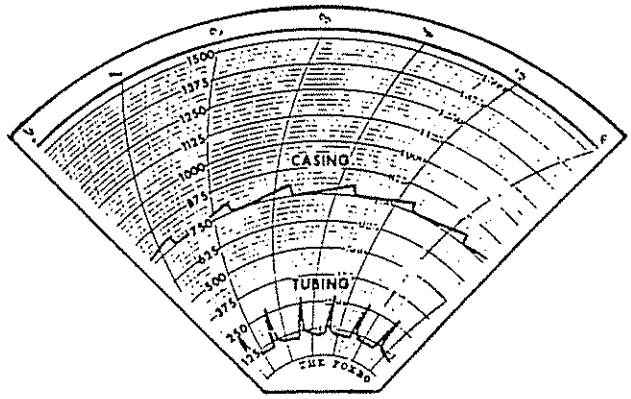


Fig. 3. 620 (10) *Choke Control of Injection Gas (Good Operation). Indication: (1) Casing pressure build-up through small choke at a constant rate; (2) Rapid casing pressure drop after valve opens; (3) Fluid operated or pressure operated valves used.*

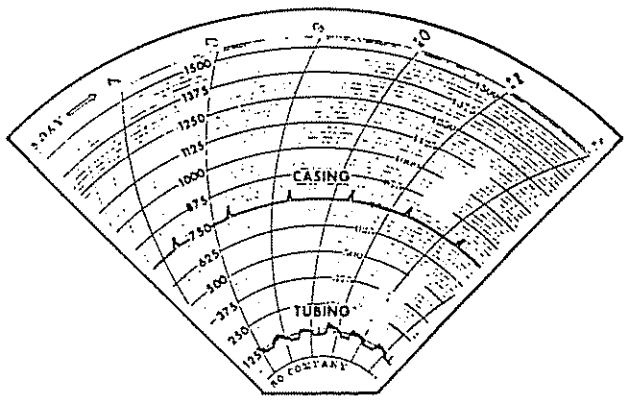


Fig. 3.620 (11) *Good Fluid Head Not Received at Surface. Indication: (1) Tubing pressure does not indicate a peak; (2) Only gas spray received at surface. Remedy: Increase length of cycles, check pressure and spread of valve.*

3.63732 Field cases of two-pen recorder charts for intermittent flow.

Numerous self-explanatory examples are presented in Figures 3.621(1) through 3.621(30). These show various operations.

Figures 3.621(31) A, B, C:

Figures 3.621(31) A, B and C show a valve with a variable closing pressure depending upon the tubing pressure opposite the valve. In practice the closing pressure is not always constant and becomes tubing pressure dependent. This is noted on Fig. 3.621(31)-A where every other kick is a lighter tubing load and the closing pressure on the valve is higher each time this occurs. Fig. 3.621(31)-B shows some of the same effect, while 3.621(31)-C shows this effect has been eliminated. Figure 3.621(32) shows this occurrence again with the lighter tubing loads showing higher closing pressures. This is not too critical in design; the common practice of assuming a constant closing pressure regardless of the load is permissible and, in fact, will be true for some valves.

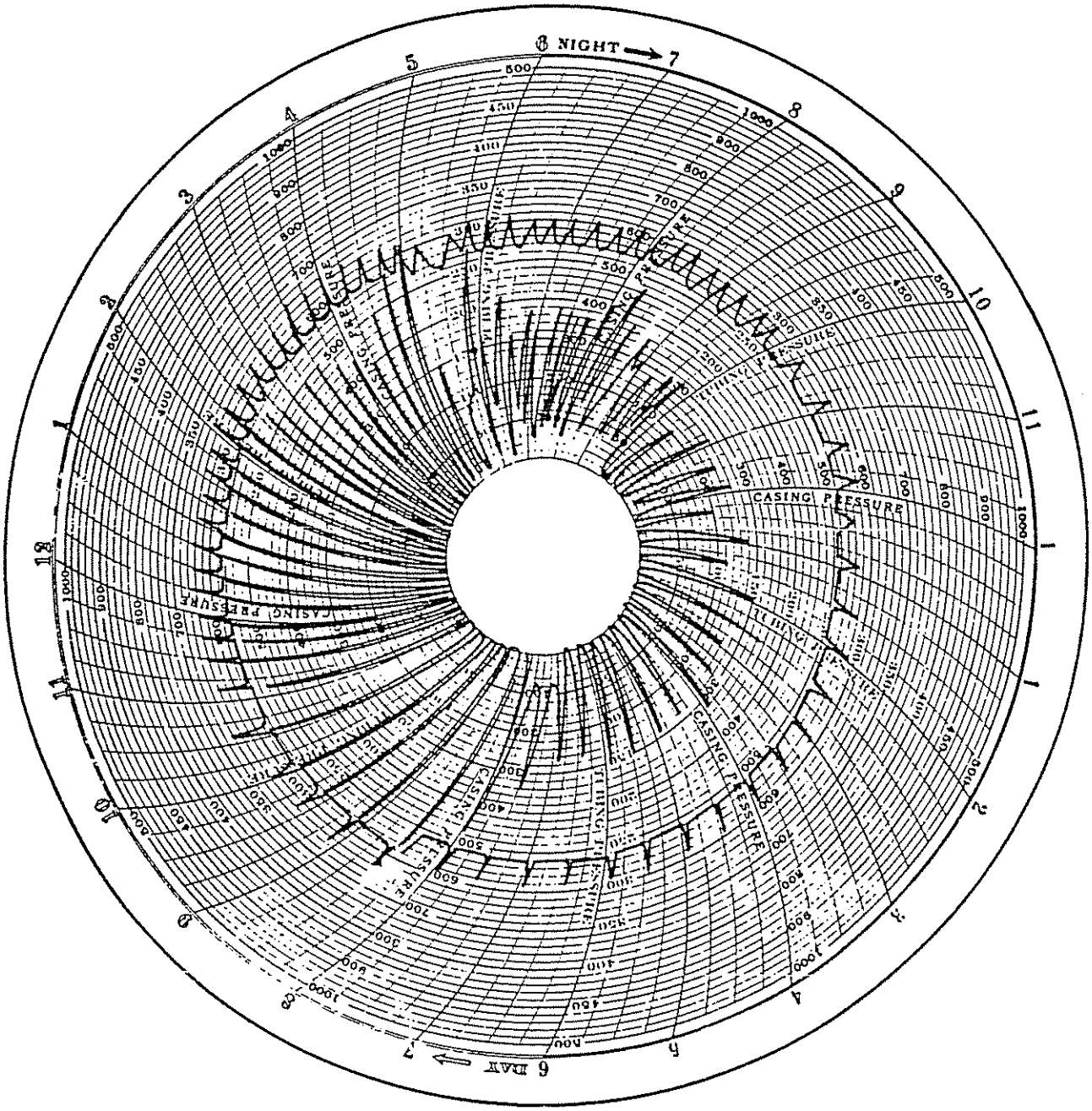


Fig. 3.621 (1) Recorder Chart—Operation: intermittent unloading process. Trouble: none. Remarks: A drop in casing pressure shows a lowering from one valve to the next. Note the well being unloaded to the pit at times.

Figures 3.621(33) A and B:

Figures 3.621(33) A and B represent casing and tubing recordings [Fig. 3.621(33)-A] and gas measurement [Figure 3.621(33)-B] in the same well at the same time. This is an excellent example of an operator desiring a continuous flow operation but must inject excess gas to maintain the fluid operated valve open for continuous flow. As noted from the gas measurement chart, it is possible to maintain continuous flow by keeping the differential high. As the differential is lowered (less gas), the well begins to intermitt. If the intermittent slugs are not too detrimental to the system, the well should be left on intermittent lift since production was higher on intermittent lift in this case.

Figure 3.621(34):

Figure 3.621(34) is a typical two-pen pressure recording for an intermittent well on choke control. Gas is injected continually into the well and liquid slugs are produced at approximately 1-hour intervals. The choke size is determined by trial and error settings.

Figures 3.621(35)A and 3.621(35)B:

Figures 3.621(35)A and 3.621(35)B show the combination control of a choke and regulator shut-off. This eliminates the tedious trial and error procedure of find-

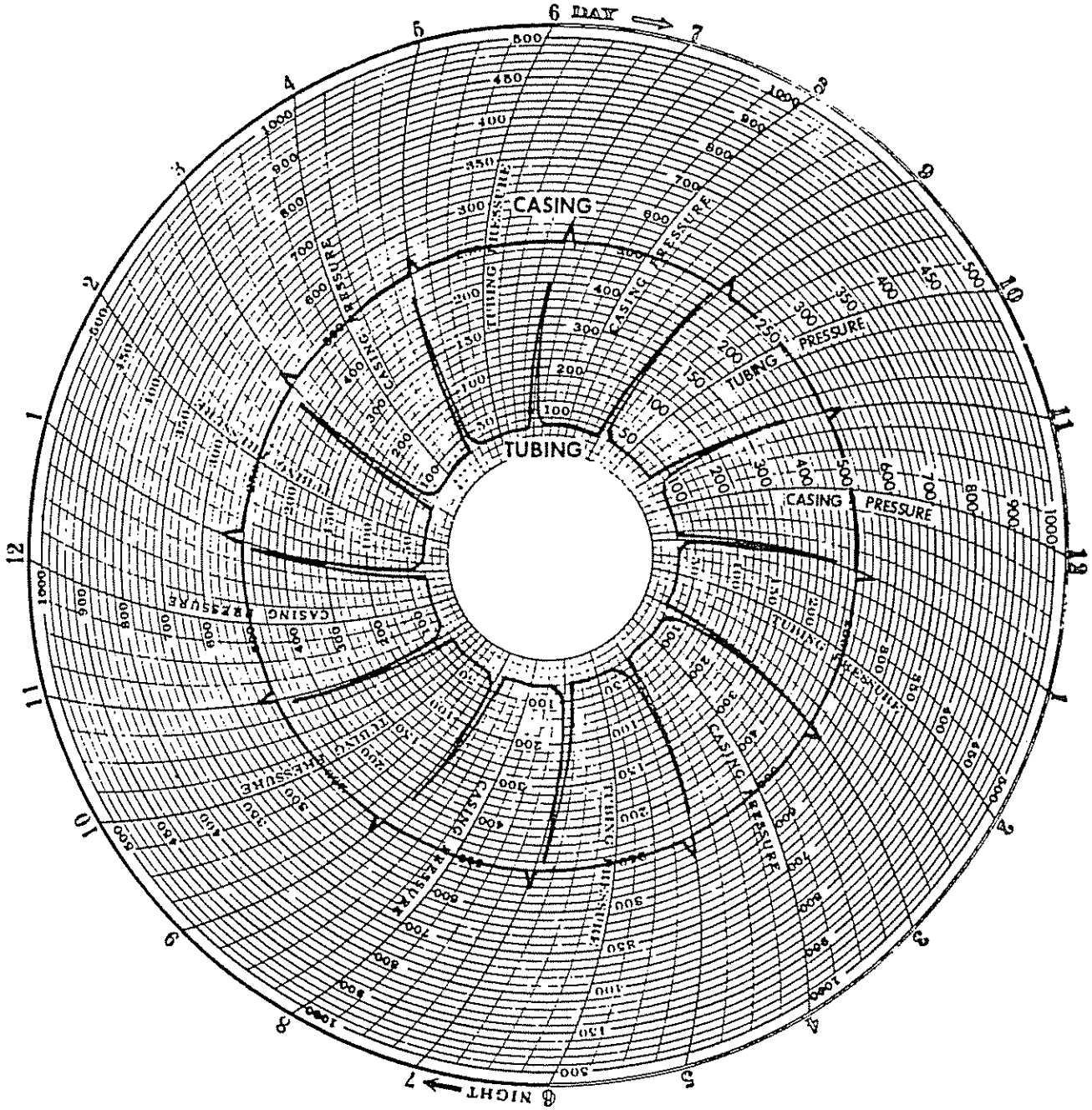


Fig. 3.621 (2) Good Operation—Time Cycle Control

ing the correct choke size. When the surface casing pressure is high enough to furnish sufficient gas, the regulator shuts off the gas injection. The valve remains closed until sufficient tubing load at the valve builds up to open it. At this time the casing pressure drops and furnishes the necessary gas to lift out the slug of liquid.

Figures 3.621(36) and 3.621(37):

These two figures show an inefficient choke flow operation with no clear liquid slug recoveries.

Figure 3.621(38):

This shows a good choke control operation.

Figures 3.621(39) and 3.621(40):

These two figures show time cycle control in small diameter casing. The high casing pressure build-up is required to furnish enough gas.

Figures 3.621(41) through 3.621(57):

Figures 3.621(41) through 3.621(57) show additional field cases of recorder charts for intermittent lift; explanations are on the charts.

Figures 3.621(58) and 3.621(59):

Figure 3.621(58) shows a dual installation with a three-pen pressure recording. Both zones are on intermittent lift, using fluid valves with gas entering into the annulus through a choke.

Figures 3.621(59)A through 3.621(59)B illustrate the

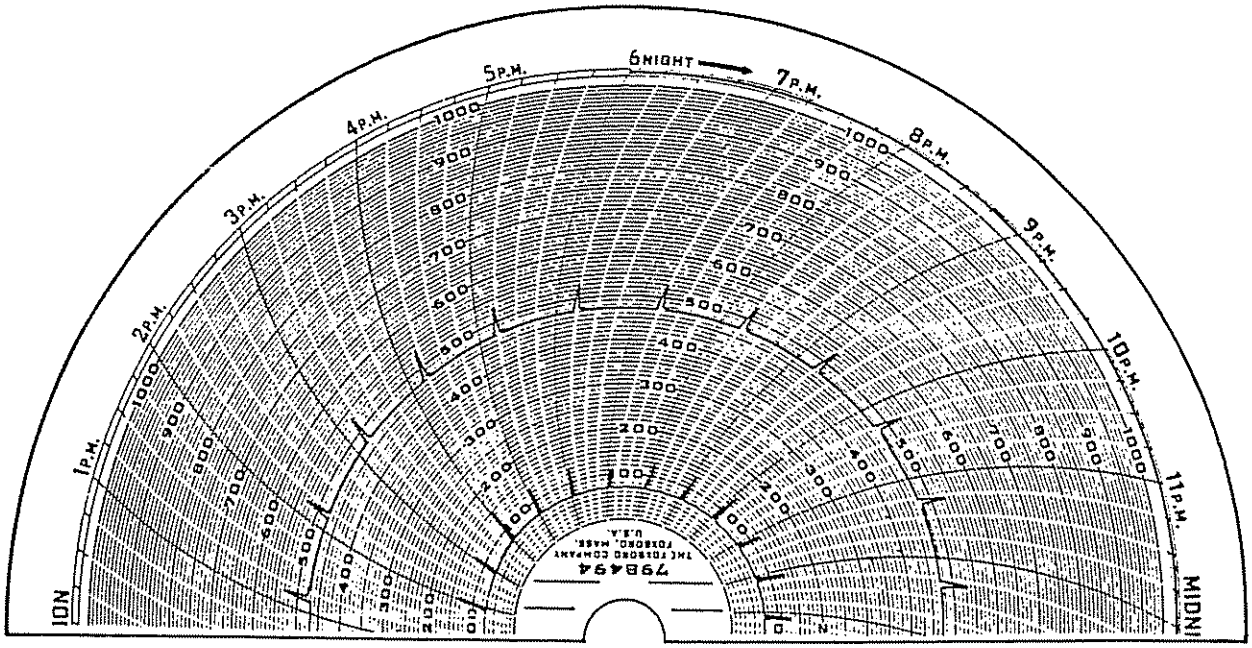


Fig. 3.621 (3) Recorder Chart—Operation: Intermittent lift. Trouble: None. Remarks: Good operation—low productivity well. Time Cycle Control.

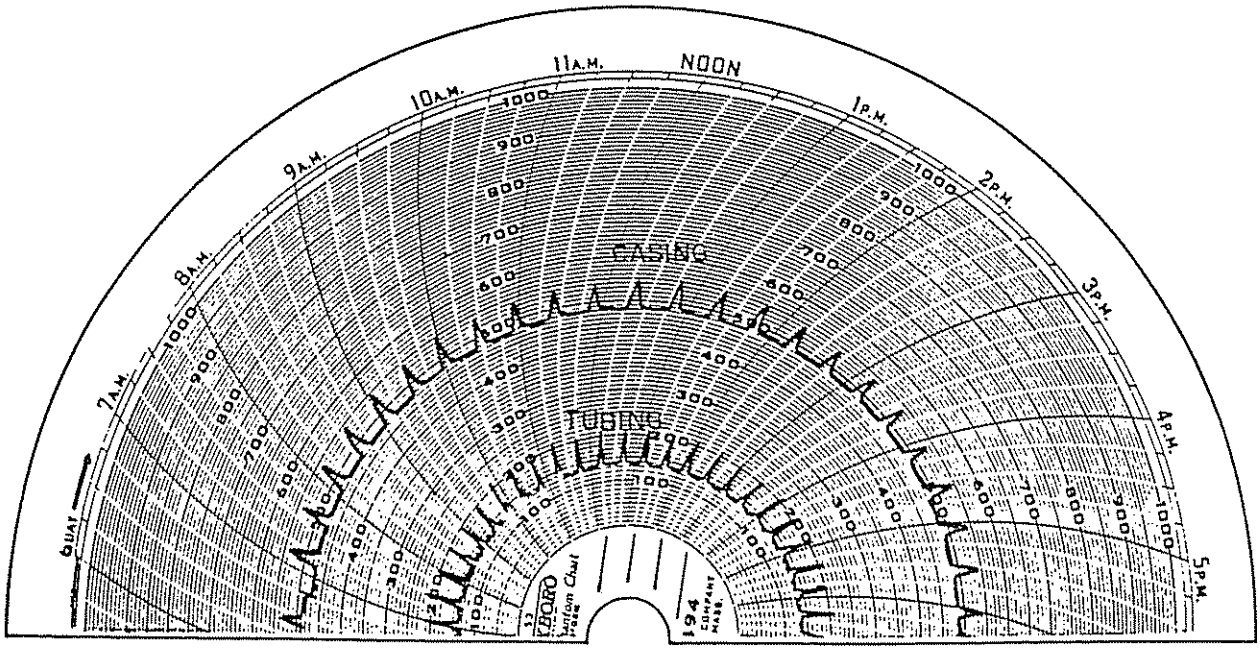


Fig. 3.621 (4) Recorder Chart—Operation: Intermittent lift. Trouble: None. Remarks: Good operation—Time Cycle Control.

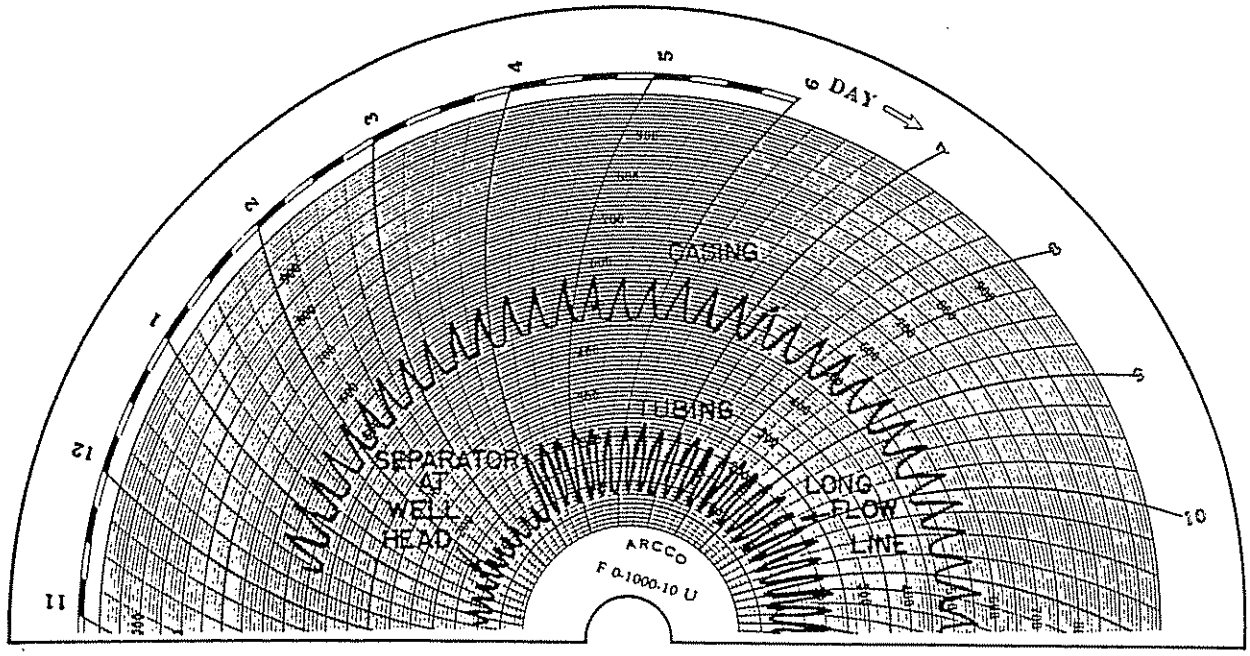


Fig. 3.621 (5) Recorder Chart—Operation: Intermittent lift (6000 ft.). Trouble: None. Remarks: Experimental tests, showing same slugs as produced into a separator at the well head as compared to one over a long flowline.

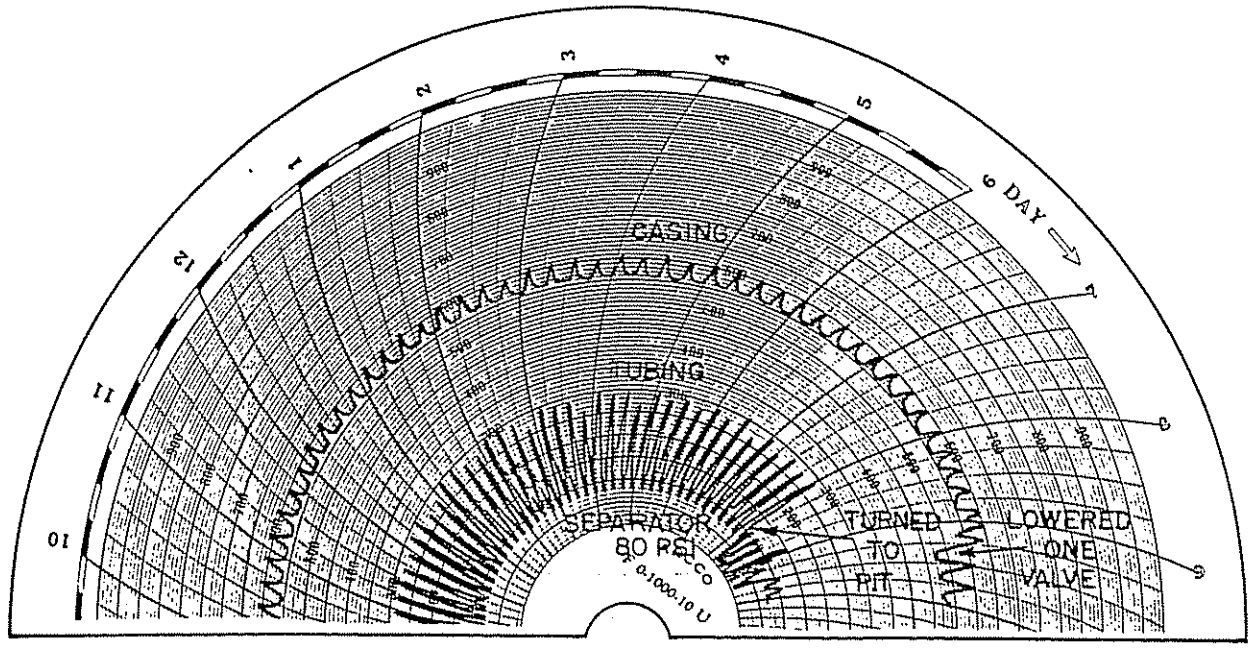


Fig. 3.621 (6) Recorder Chart—Operation: Intermittent lift (5000 ft.). Trouble: Well would not unload to bottom valve. Remarks: Well was turned to the low pressure and immediately unloaded to bottom valve at 6000 ft.—many wells can reach a deeper operating depth by unloading to a lower pressure for a while.

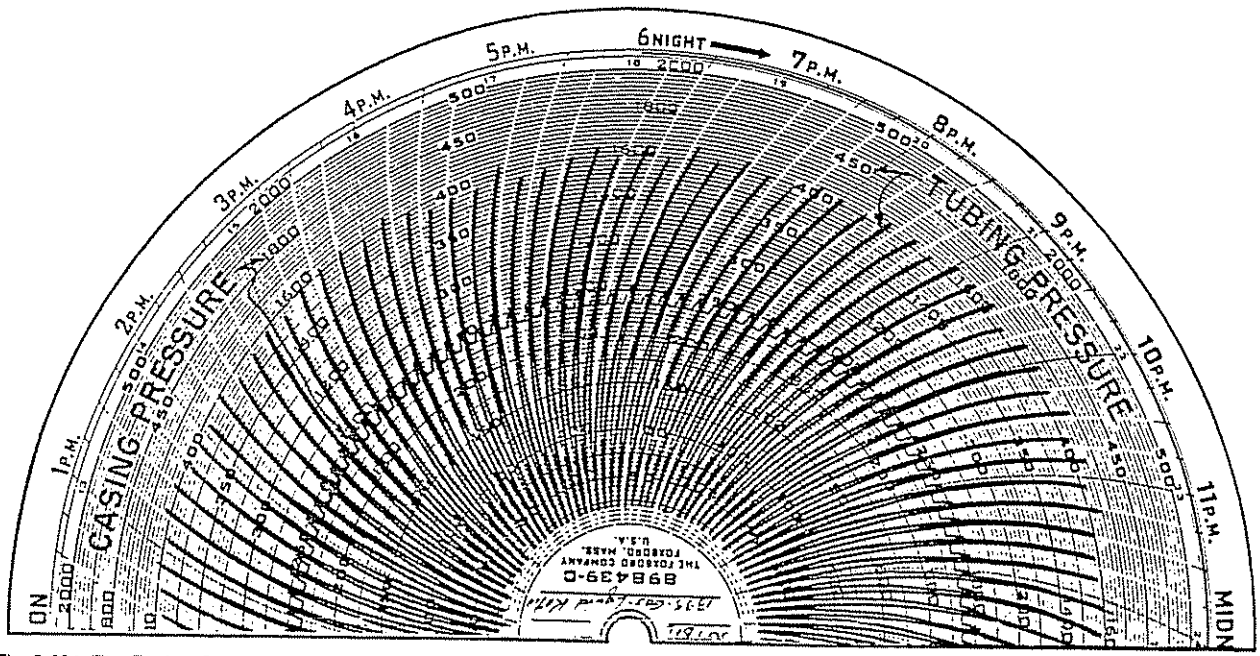


Fig. 3.621 (7) Typical Recording of Casing and Tubing Pressure for High Productivity Chamber Performance (7-in. casing with 5-in. liner; 2 ⅞-in. tubing; production, 565 BOPD and no water; GLR, 1395/1; 12 minute cycles; 4.70 bbls/cycle; 8 bbl. chamber; depth to operating valve, 6274 ft.)

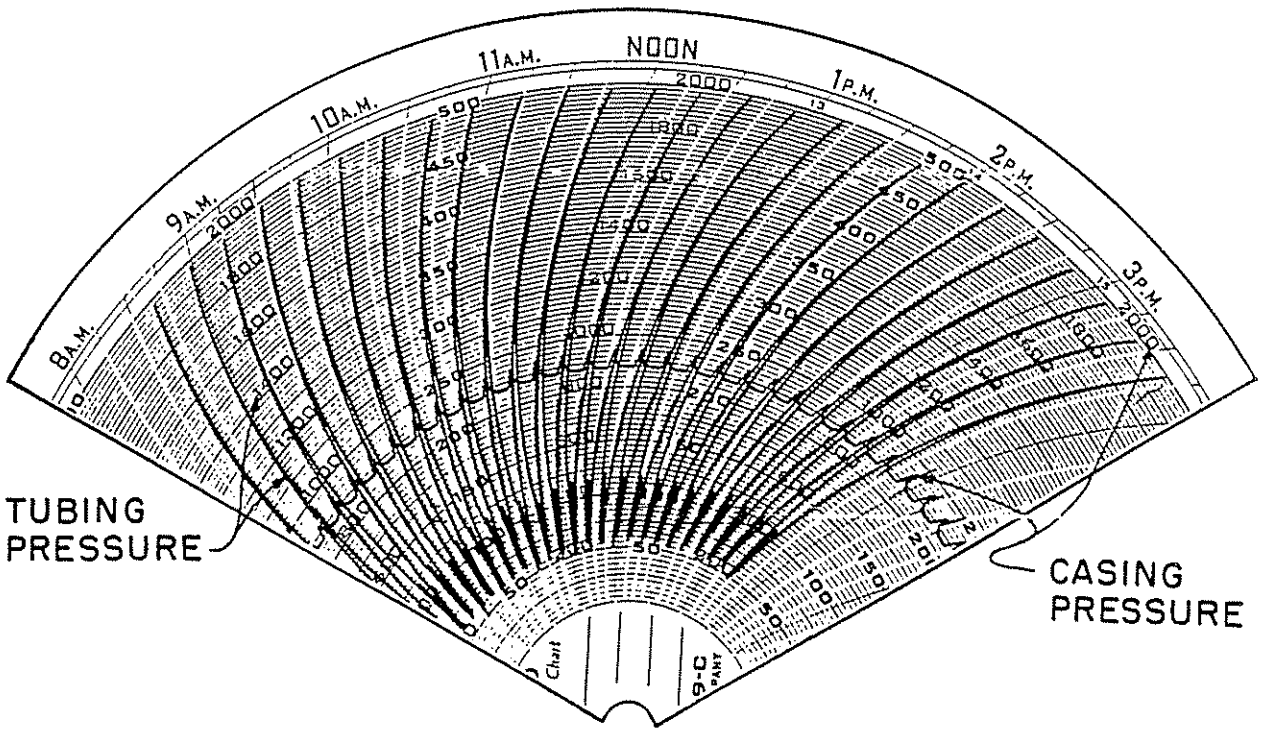


Fig. 3.621 (8) Typical Recording of Casing and Tubing Pressure for High Productivity Chamber Performance (7-in. casing; 2 ⅞-in. tubing; production, 639 BOPD with no water; GLR, 1059/1; 15-minute cycles; 6.65 bbls/cycle; 10 bbl chamber; depth to operating valve, 6315 ft.)

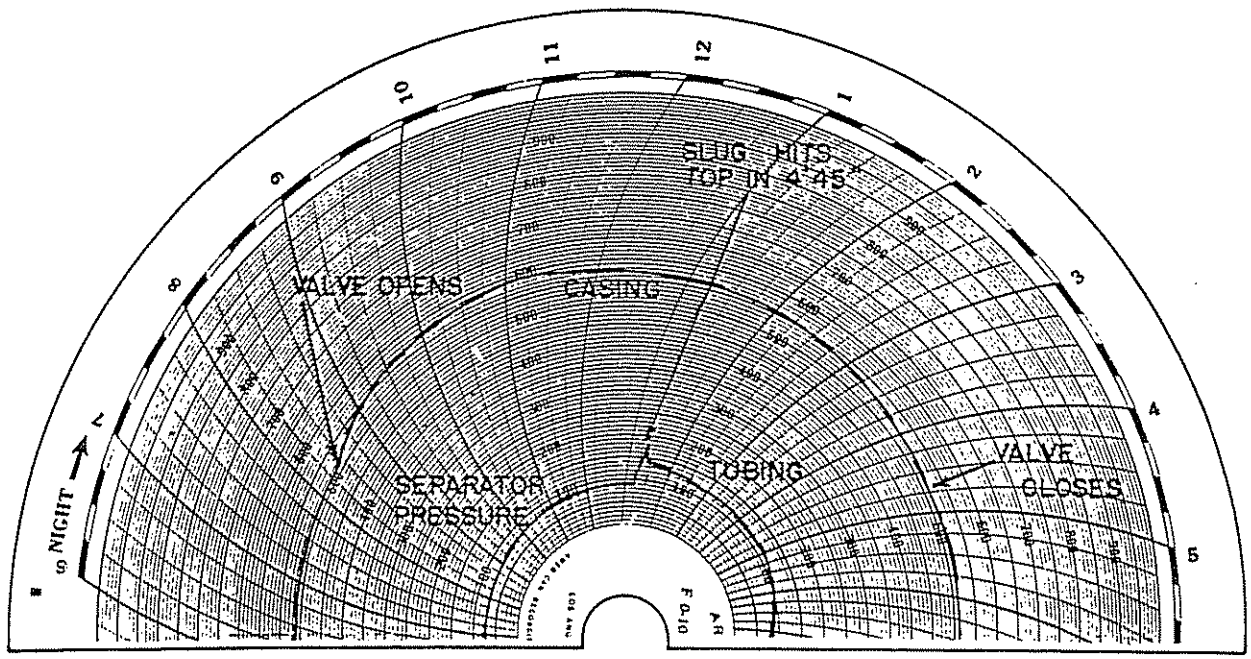


Fig. 3.621 (9) Recorder Chart—Operation: Intermittent lift as recorded on a 24-minute revolution clock (6000 ft. depth; 1-in. ported valve; GLR 2000/1; recovery, 2 bbls.). Trouble: None. Remarks: Shows typical recording of an intermittent kick on a 24-minute clock—this recording gives details on time valve opens, time valve closes, slug arrival time, good or bad slug; broken slug behind main slug, as well as other information.

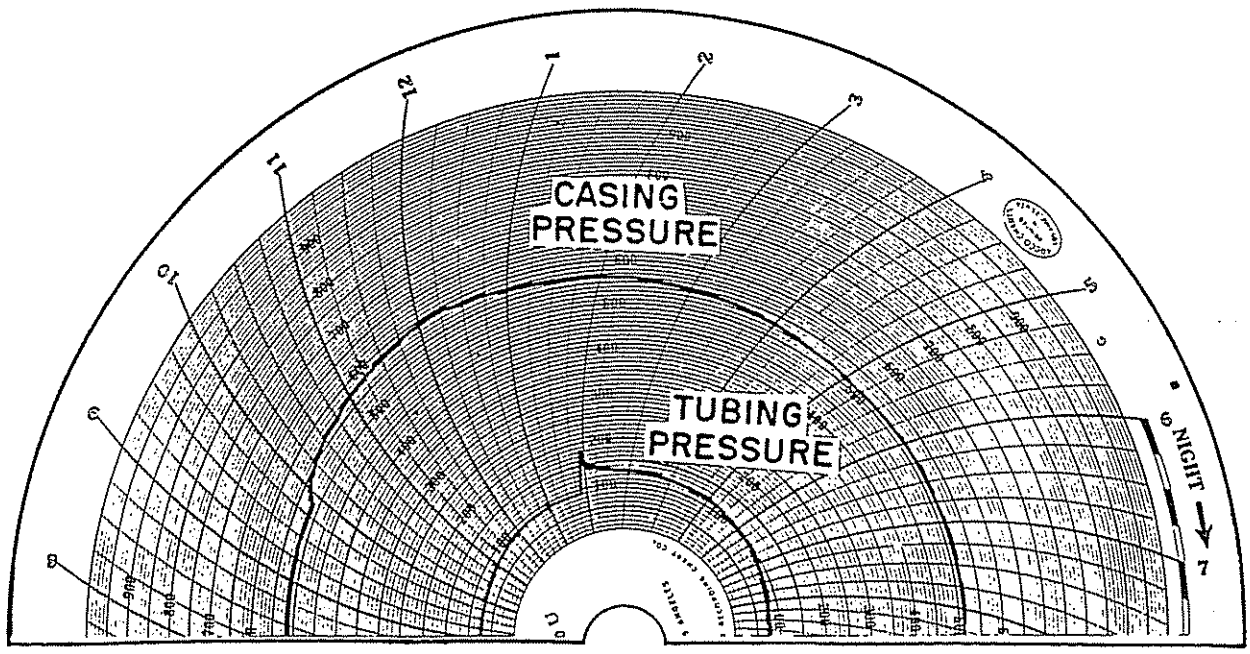


Fig. 3.621 (10) 24-Minute Revolution Recording Chart

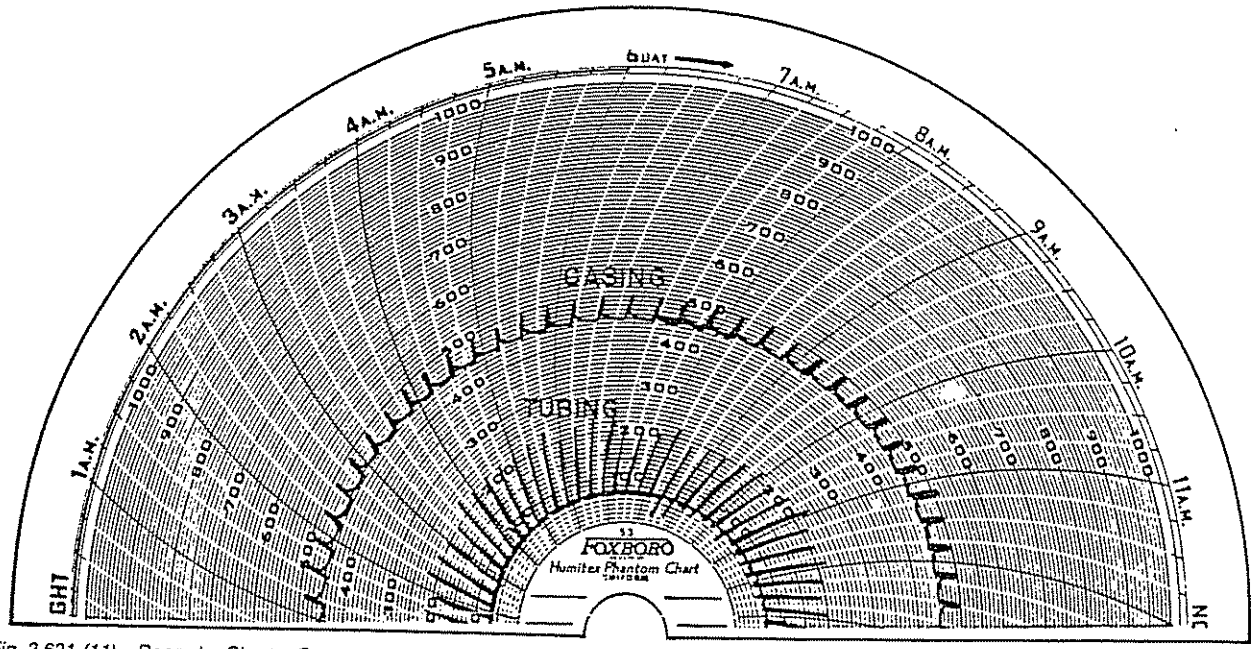


Fig. 3.621 (11) *Recorder Chart—Operation: Intermittent lift. Trouble: High injection GLR. Remarks: Well is cycling too fast—lifting very light loads as indicated by fast drop in casing pressure and very thin tubing heads. Valve operation is good.*

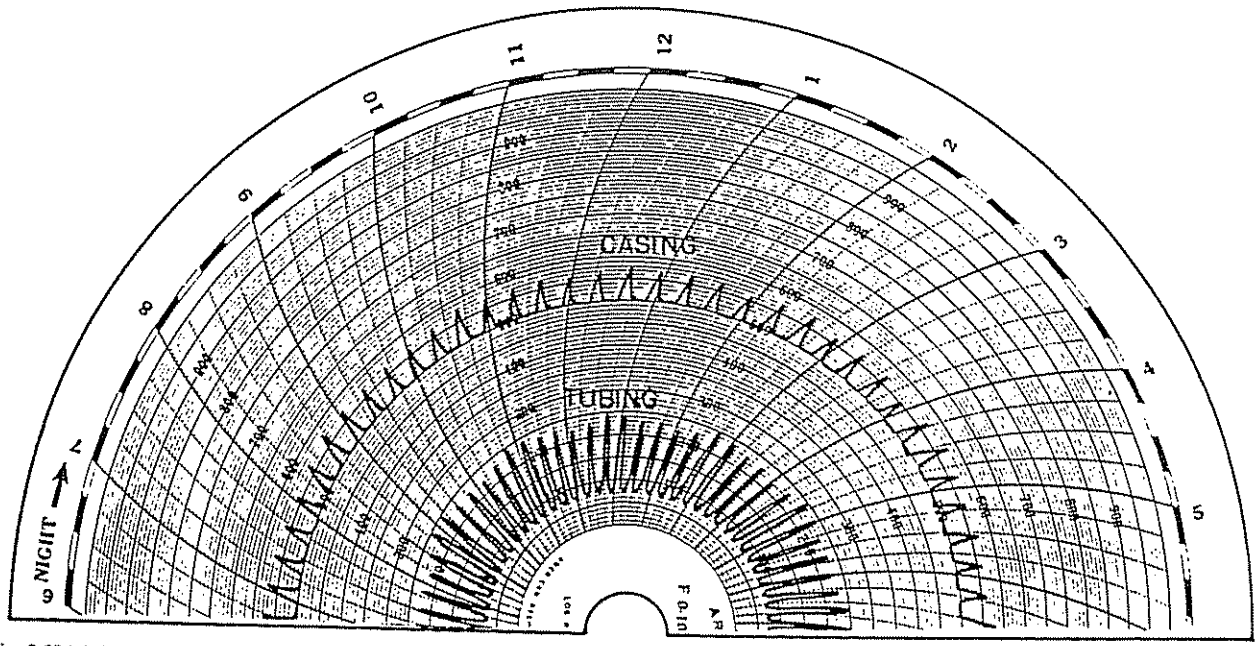


Fig. 3.621 (12) *Recorder Chart—Operation: Intermittent lift (82 BID; 20-minute cycle; 6000 ft. depth; ½-in. surface choke; 2-in. tubing). Trouble: High back pressure—cycle time not long enough. Remarks: Surface choke should be removed—cycle frequency could be increased.*

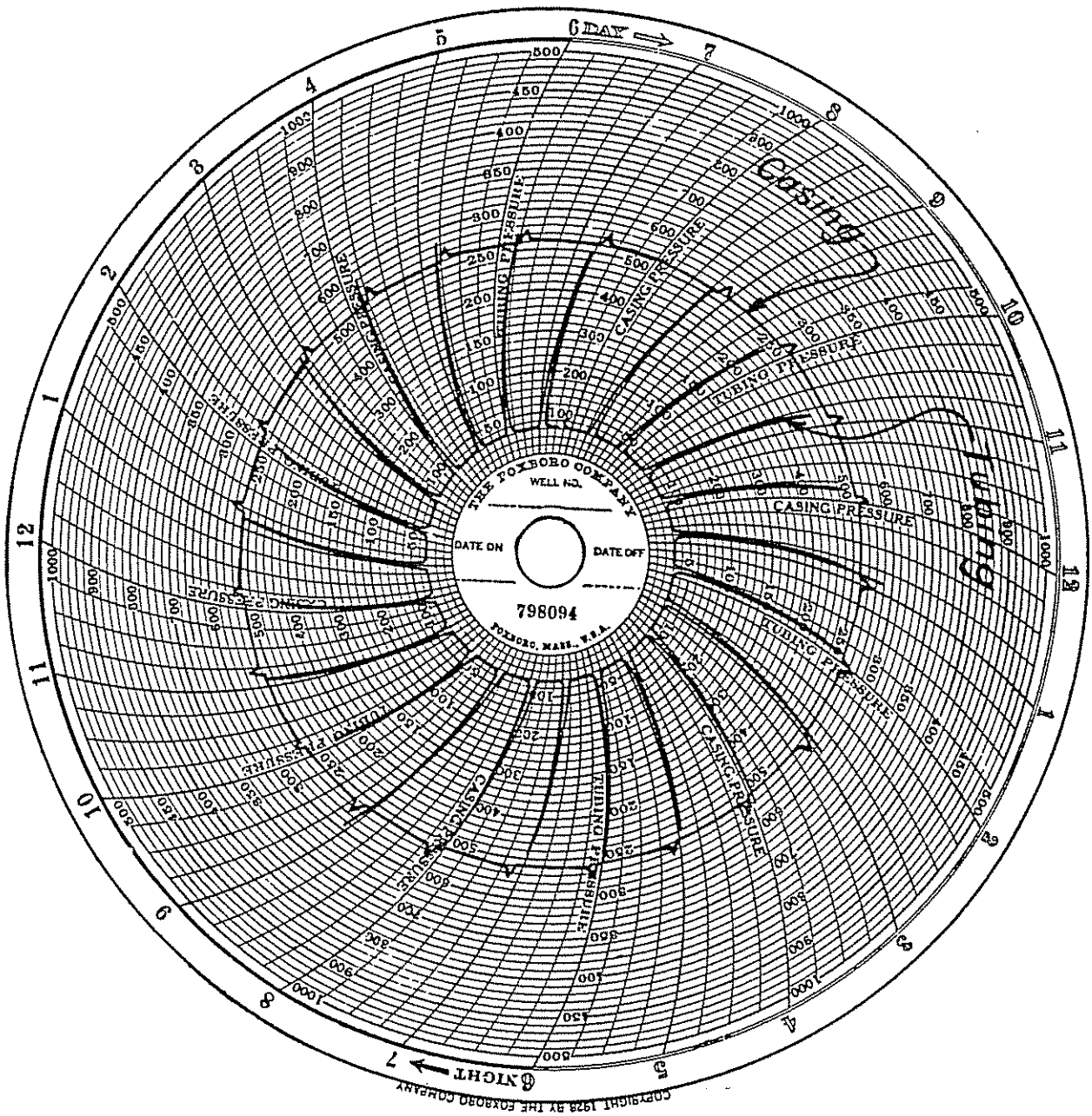


Fig. 3.621 (13) Intermittent Lift with Pack-off Valve (depth to operating valve, 8900 ft; production, maximum 60 BlD; 2 ⅜-in. tubing; 7-in. casing; valve size, 1-in. ID × 1 $^{25}/_{32}$-in. OD). Remarks: Regular valves were installed originally to 8450 ft. After BHP lowered, one pack-off was placed on bottom at 8900 ft. Production jumped from 29 BlD to 60 BlD and stabilized at 40 BlD.

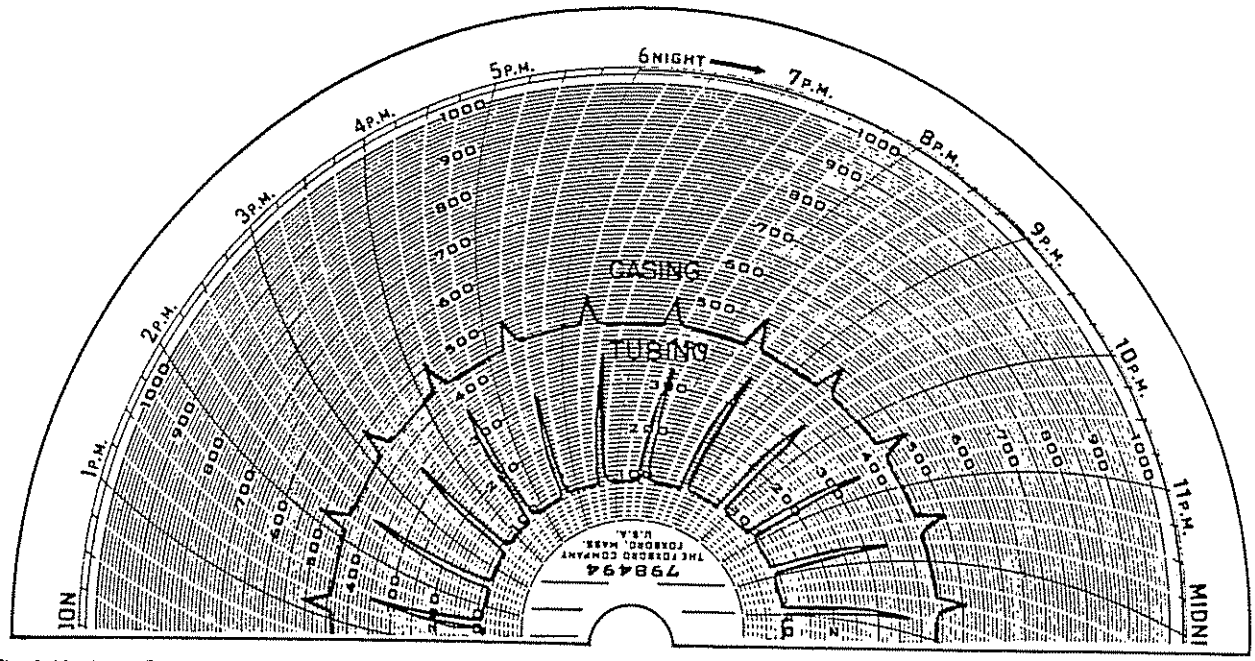


Fig. 3.621 (14) Recorder Chart—Operation: Intermittent lift. Trouble: None. Remarks: Deep lift (8000 ft.)—Tree should be examined for possibility of eliminating high maximum tubing slug pressure.

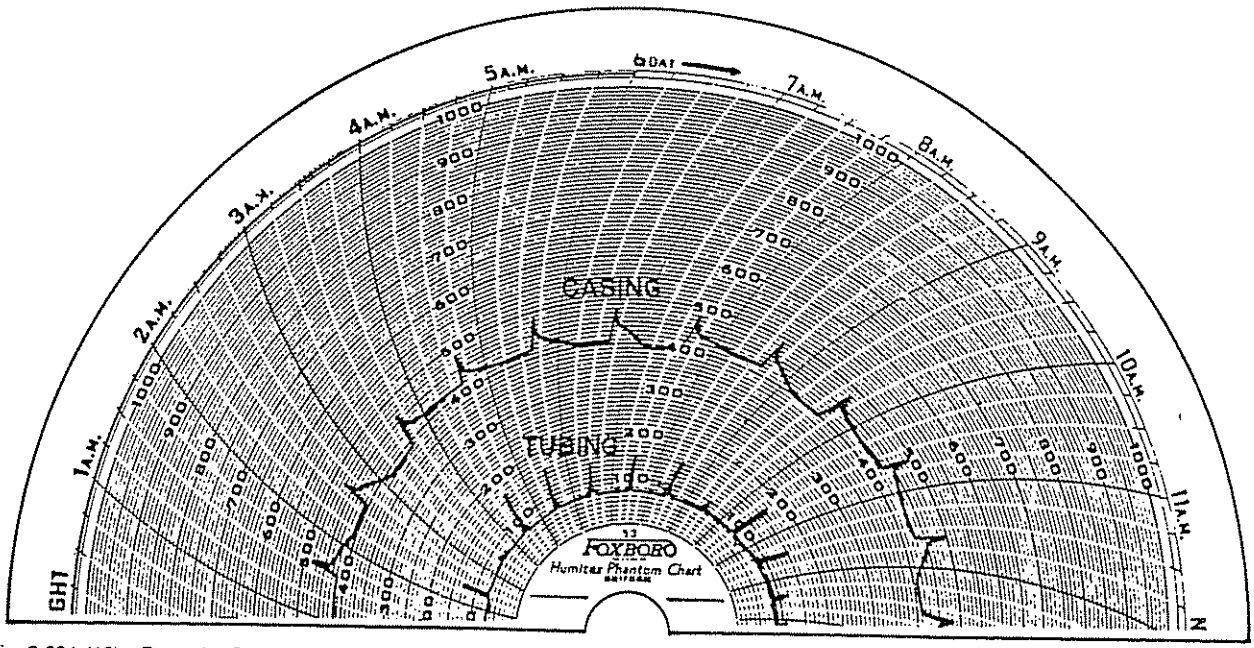


Fig. 3.621 (15) Recorder Chart—Operation: Intermittent lift. Trouble: Valve leaking and alternating valve operation. Remarks: A shorter cycle will maintain lower valve operation—if valve leakage is too bad, valve will have to be replaced.

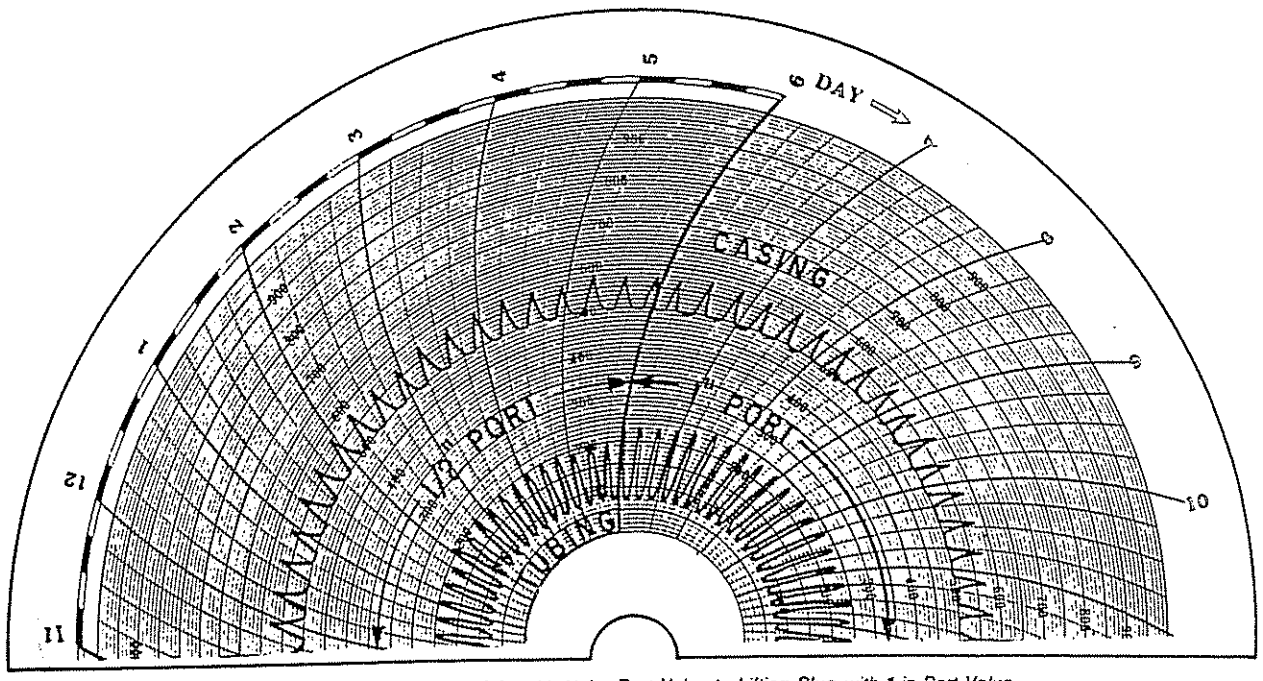


Fig. 3.621 (16) Comparison of Lifting Slug from 6000 ft. with ½-in. Port Valve to Lifting Slug with 1-in Port Valve

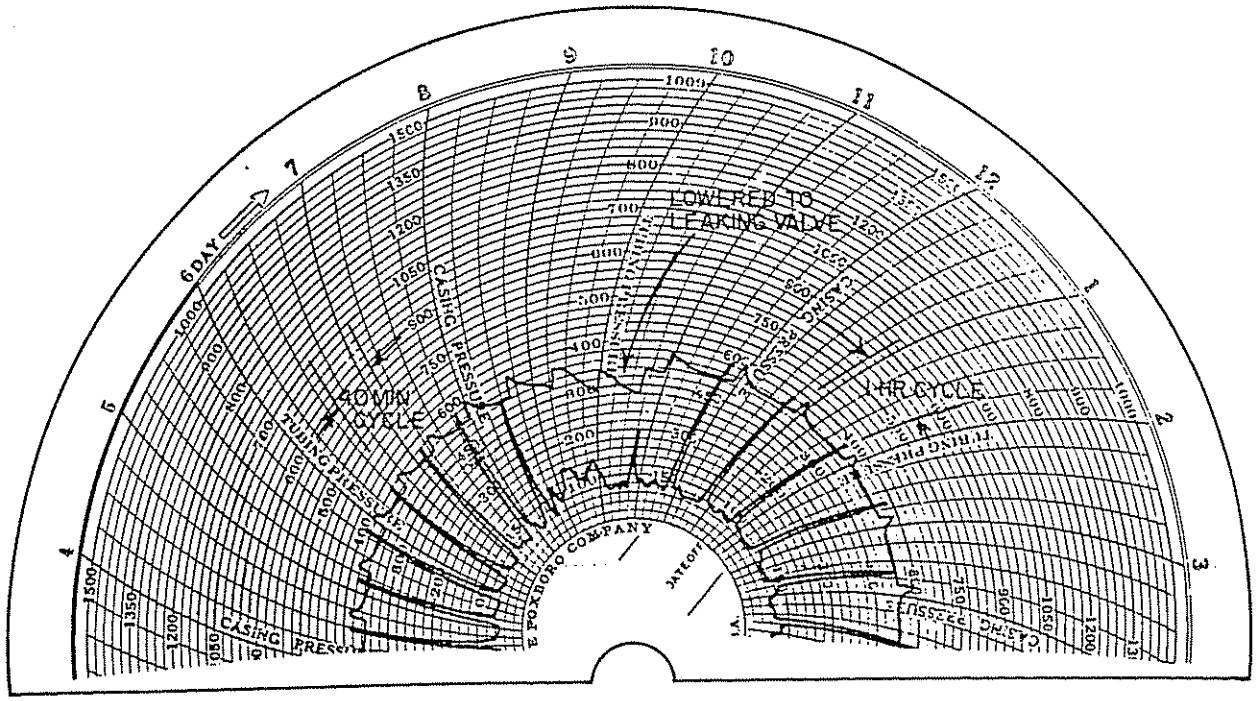


Fig. 3.621 (17) Recorder Chart—Operation: Intermittent lift. Trouble: Leaking lower valve. Remarks: Well placed on 40-minute cycle and lowered to a leaking gas lift valve; well placed back on one-hour cycle and worked from good upper valve.

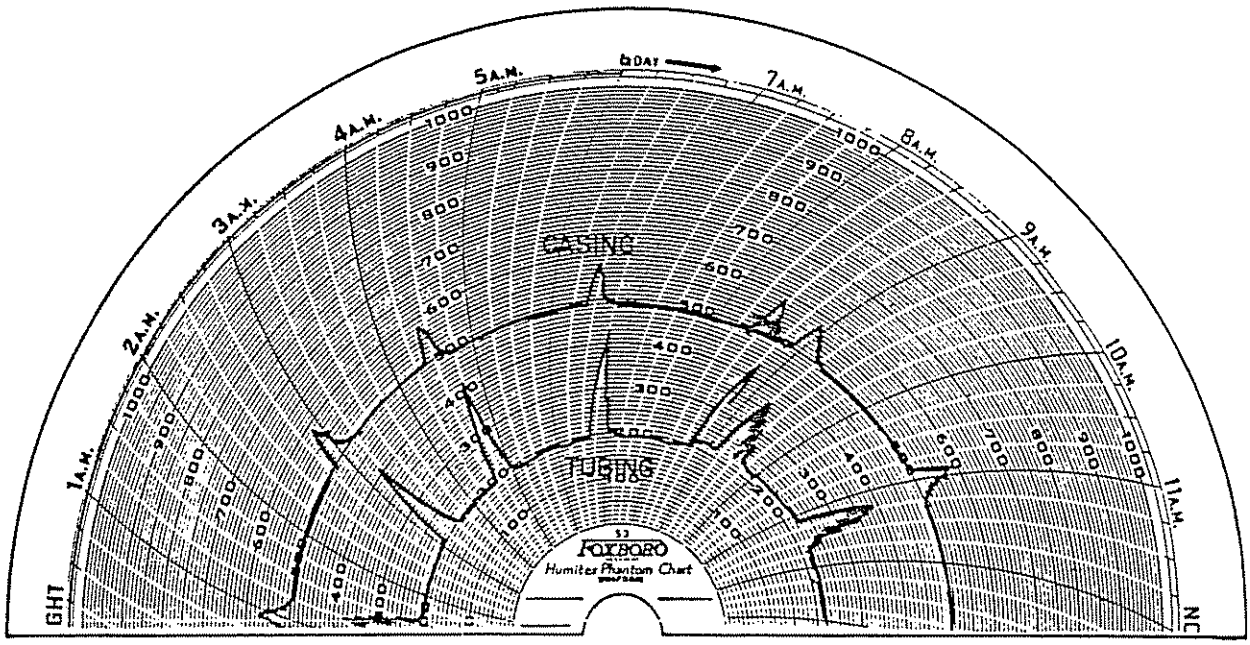


Fig. 3.621 (18) *Recorder Chart—Operation: Intermittent lift. Trouble: Cycles too long. Remarks: Well should be capable of producing more fluid on shorter cycles. Well heading at end of slug indicating excessive fallback of fluid. Reason for high trap pressure should be determined.*

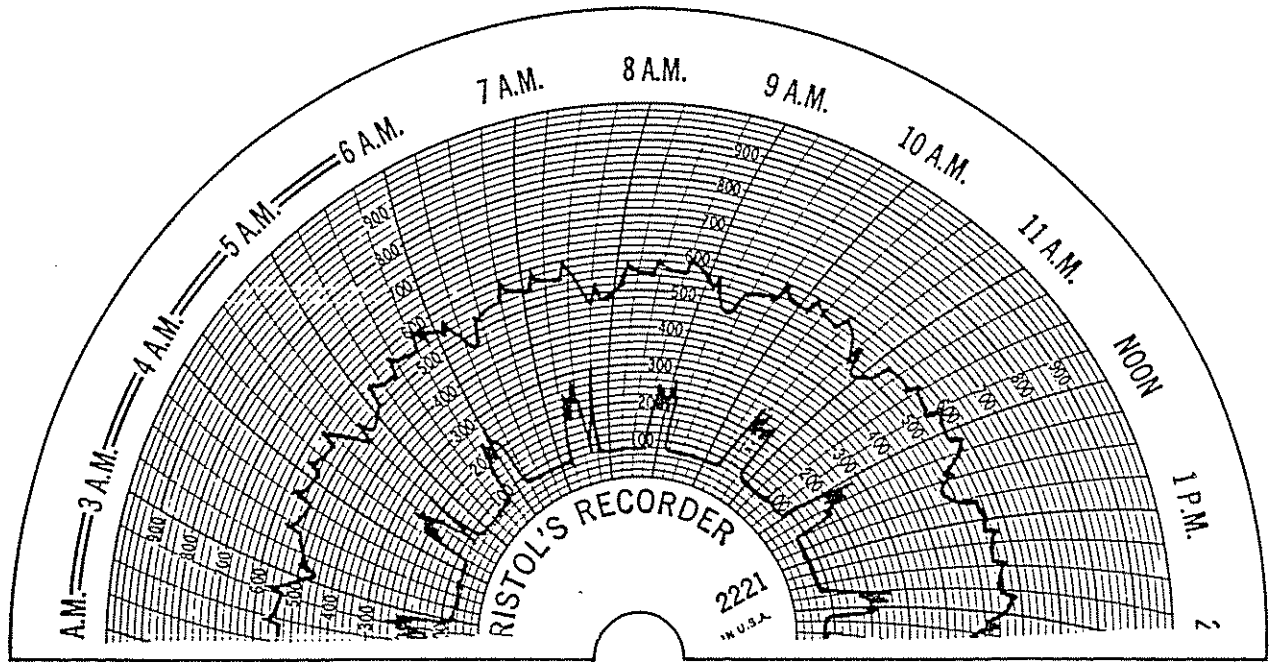


Fig. 3.621 (19) *Recorder Chart—Operation: Intermittent lift. Trouble: Not enough gas per cycle; cycle frequency too long; alternating valve action. Remarks: Try more gas per cycle and shorten cycle frequency.*

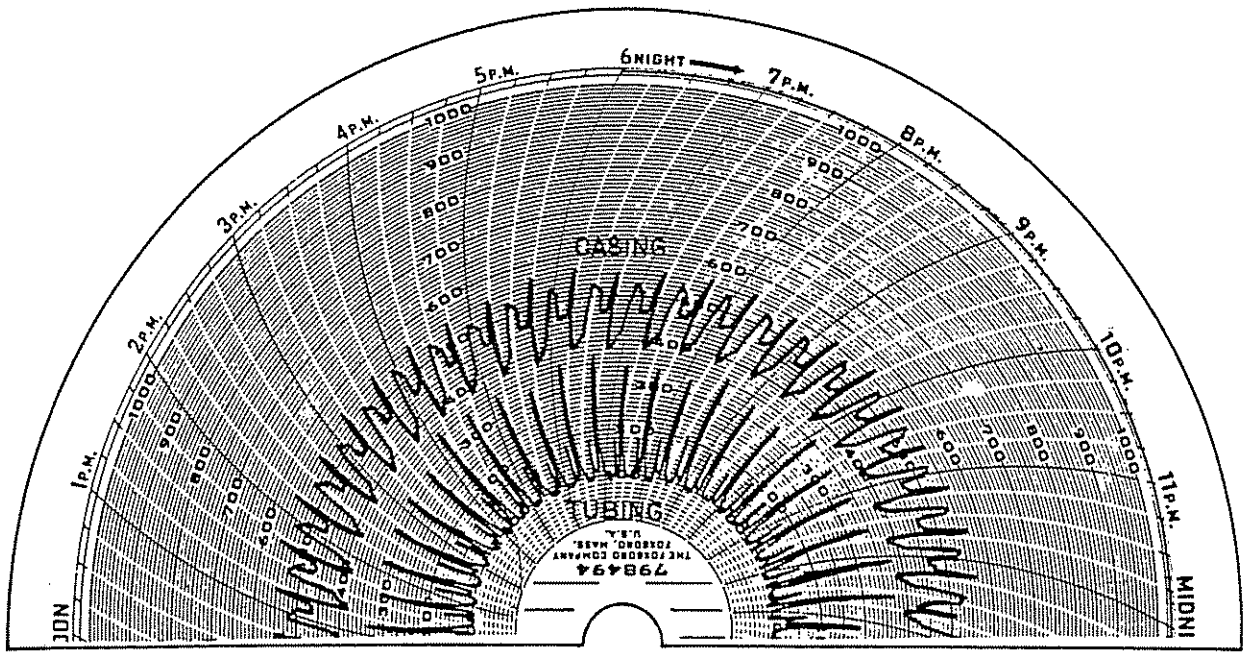


Fig. 3.621 (20) Recorder Chart—Operation: Intermittent lift. Trouble: Alternating valve action. Remarks: Operation is alternating between valves with too much drop in pressure; lower valve possibly leaking.

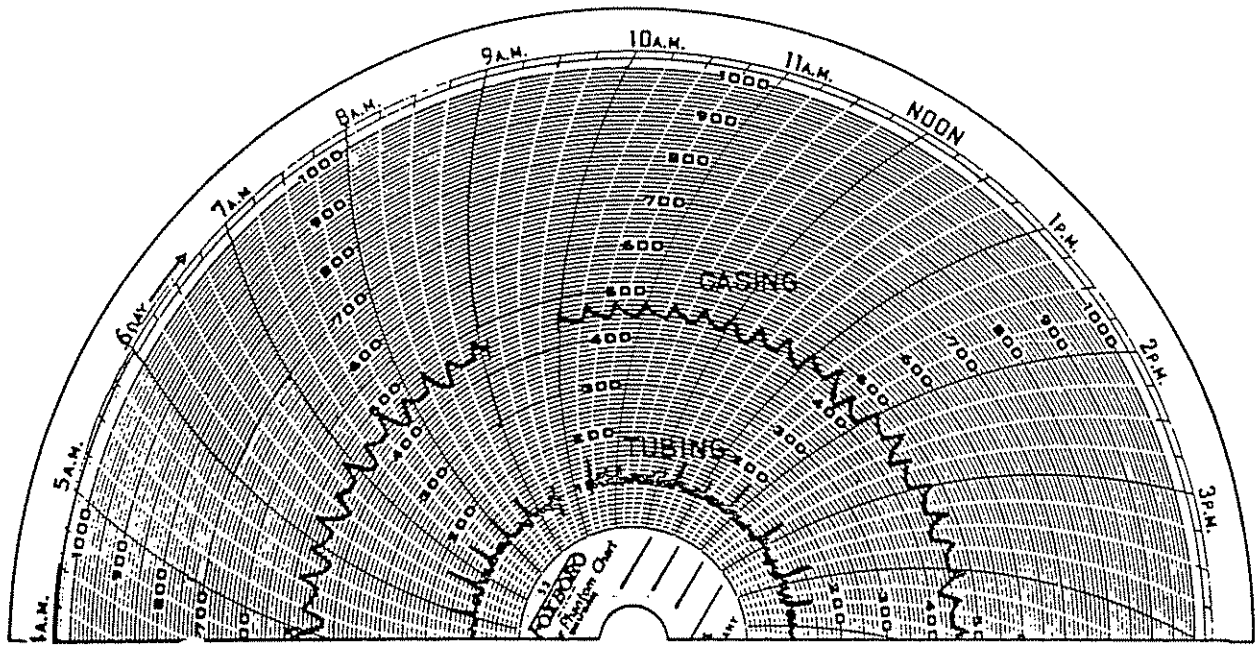


Fig. 3.621 (21) Recorder Chart—Operation: Intermittent lift. Trouble: Questionable. Remarks: Operation is alternating between two valves. For greater fluid recovery, shorten injection cycle to maintain lower valve operation. If well is making allowable, operation is satisfactory.

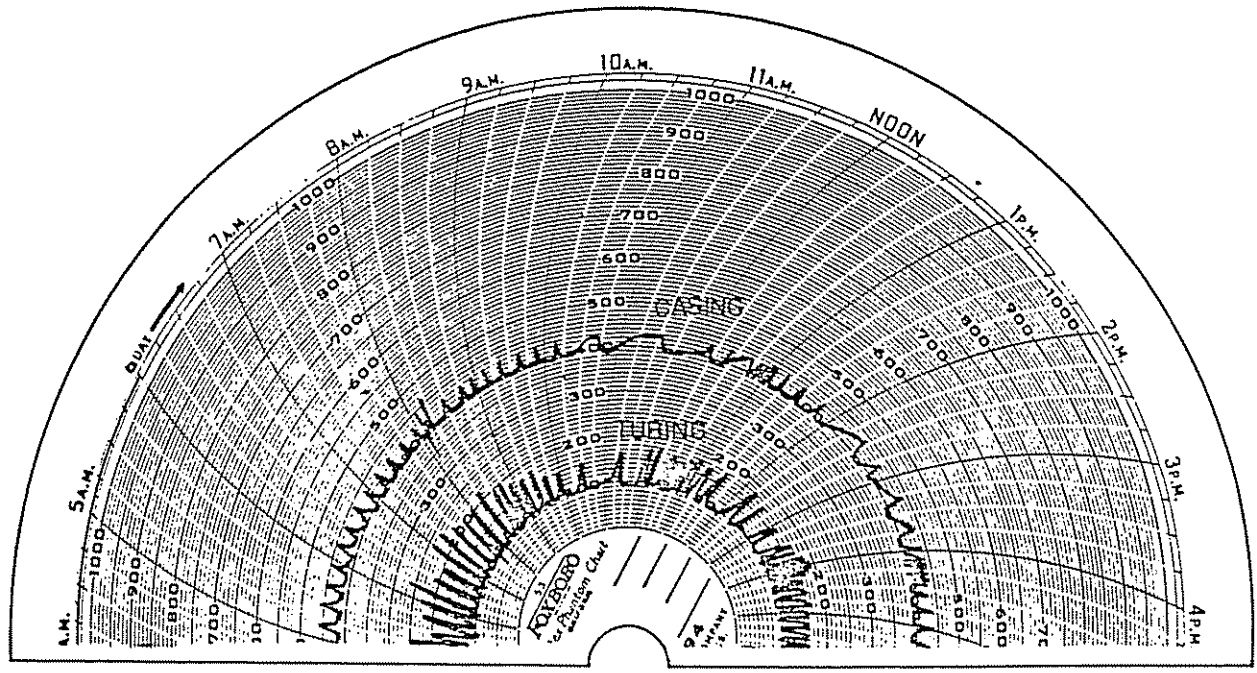


Fig. 3.621 (22) Recorder Chart—Operation: Intermittent lift. Trouble: Clock trouble. Remarks: The recorder clock should be checked. Nonuniformity of chart shows clock trouble (loose hub, etc.)

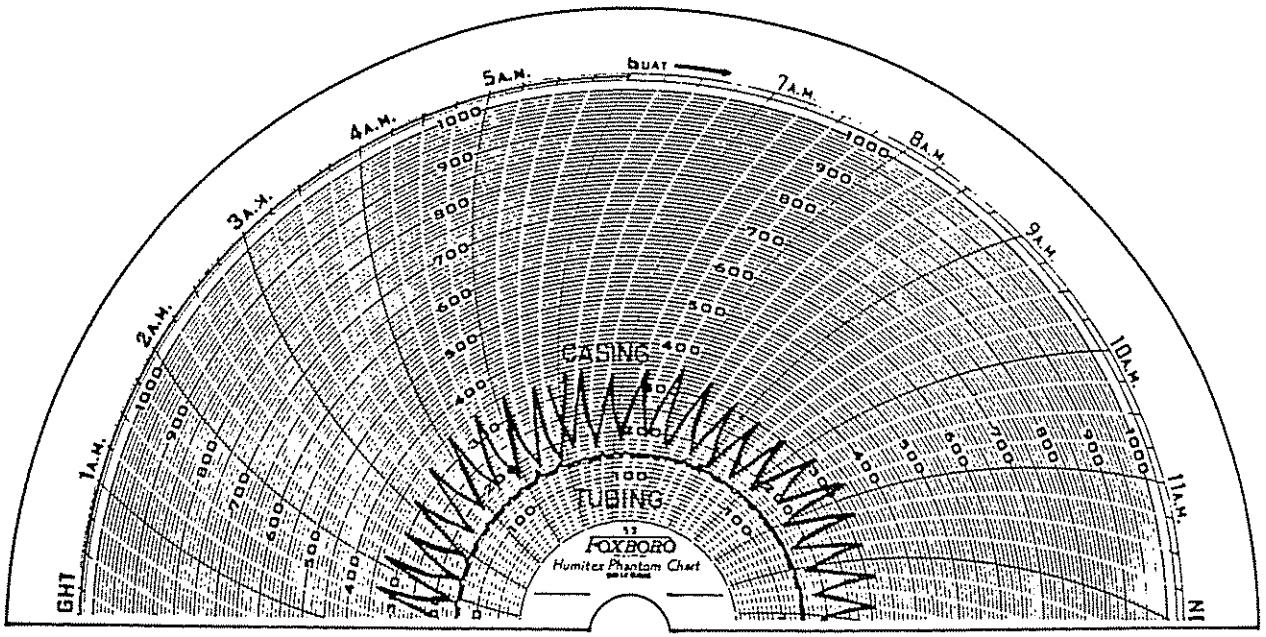


Fig. 3.621 (23) Recorder Chart—Operation: Intermittent lift. Trouble: Gas lift valve cut out or tubing leak. Remarks: Not making production, blowing dry gas.

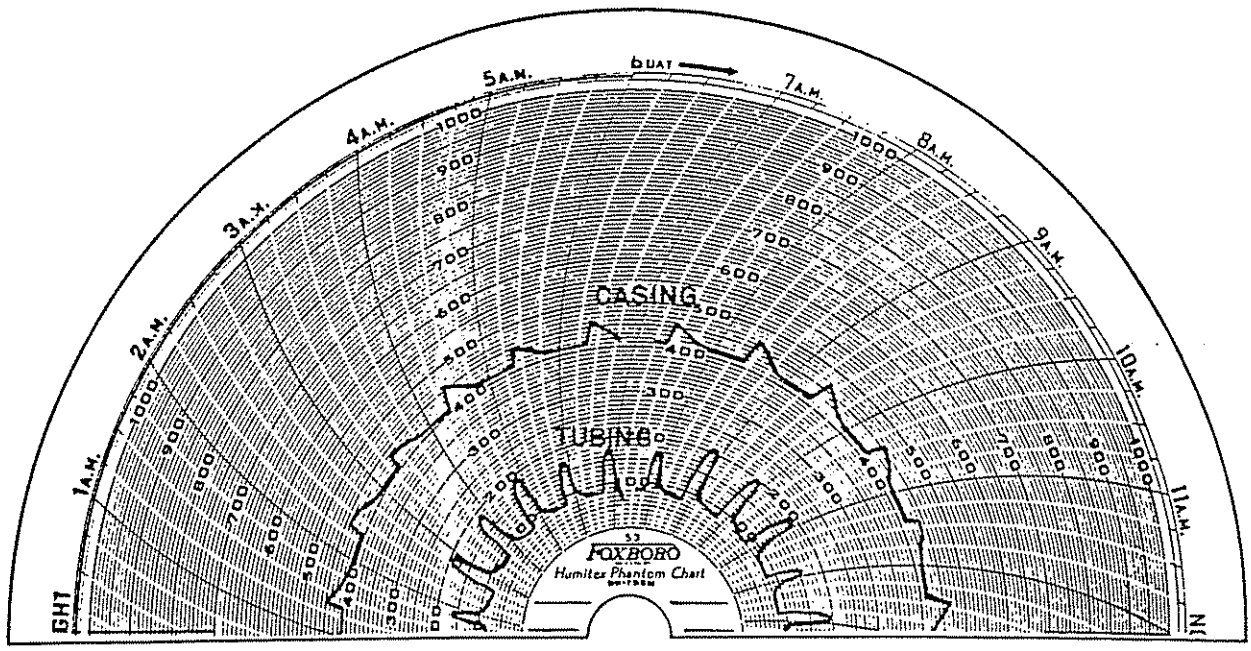


Fig. 3.621 (24) Recorder Chart—Operation: Intermittent lift. Trouble: Tubing restriction, or injection cycle too long. Remarks: Remove restriction from tubing or well head, or try a shorter injection cycle.

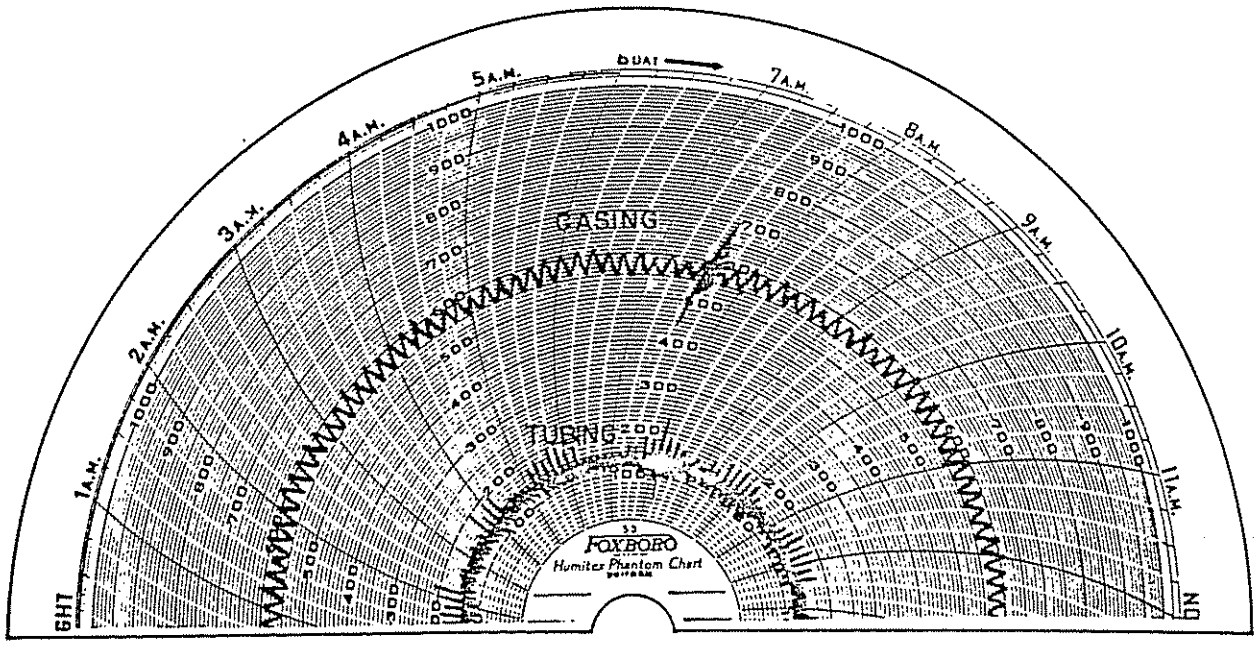


Fig. 3.621 (25) Recorder Chart—Operation: Intermittent lift. Trouble: Gas lift valve cut out or tubing leak. Remarks: This is indicated by casing pressure failing to stabilize. Casing gas should be shut off to determine the extent of the leak.

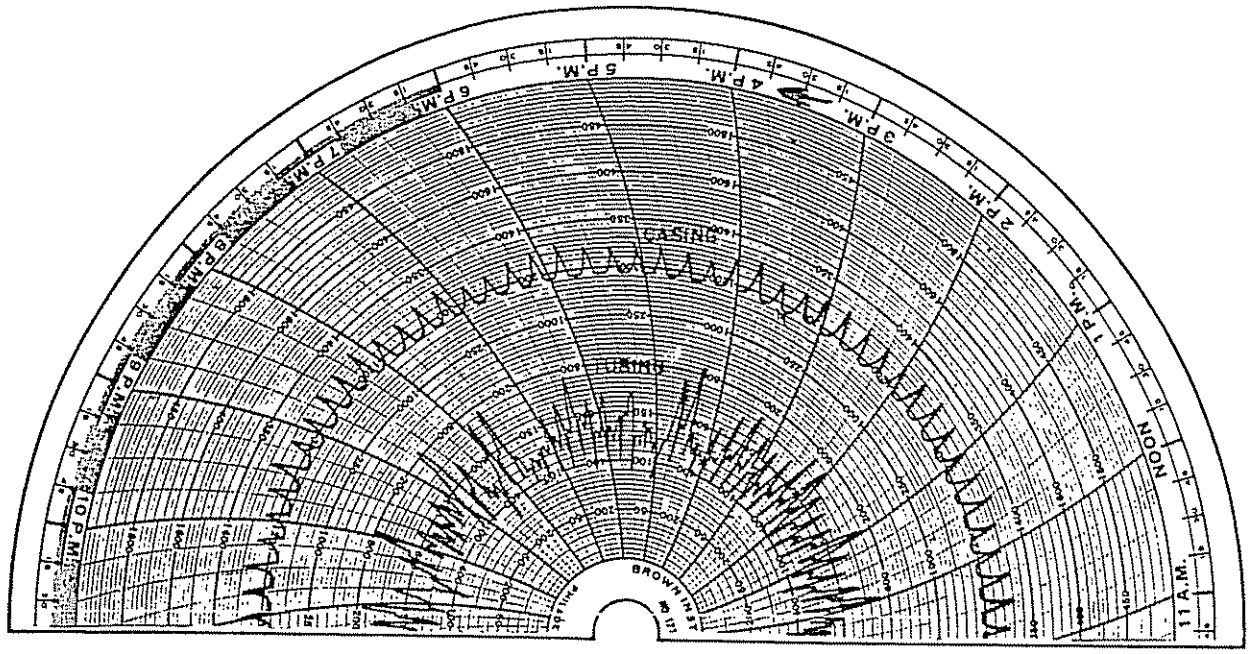


Fig. 3.621 (26) Recorder Chart—Operation: Intermittent flow (7000 ft.; 5/16-in. port). Trouble: High tubing pressure; slugs not good, clean production of 35 B/D. Remarks: 5/16-in. port too small for intermittent lift from 7000 ft.; long flowline and high separator pressure should be changed.

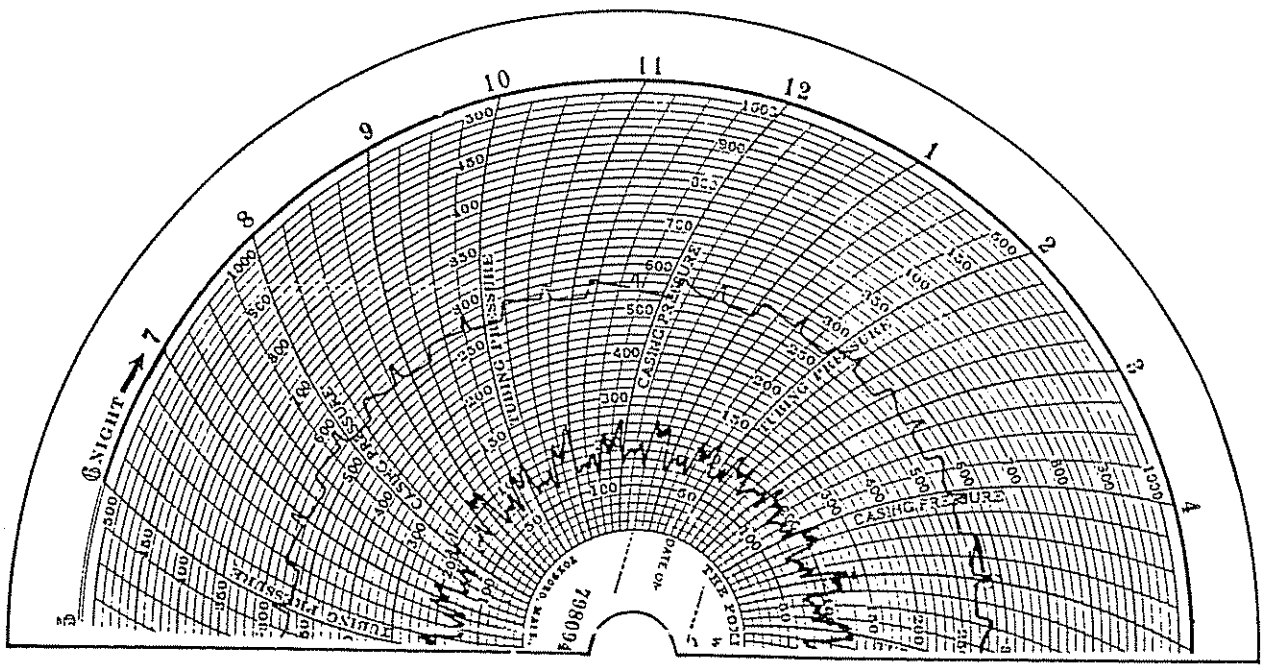


Fig. 3.621 (27) Recorder Chart—Operation: Intermittent lift. Trouble: Leaking valve and only enough injection gas to open valve every other cycle. Remarks: Increase cycle frequency—may need to replace valve.

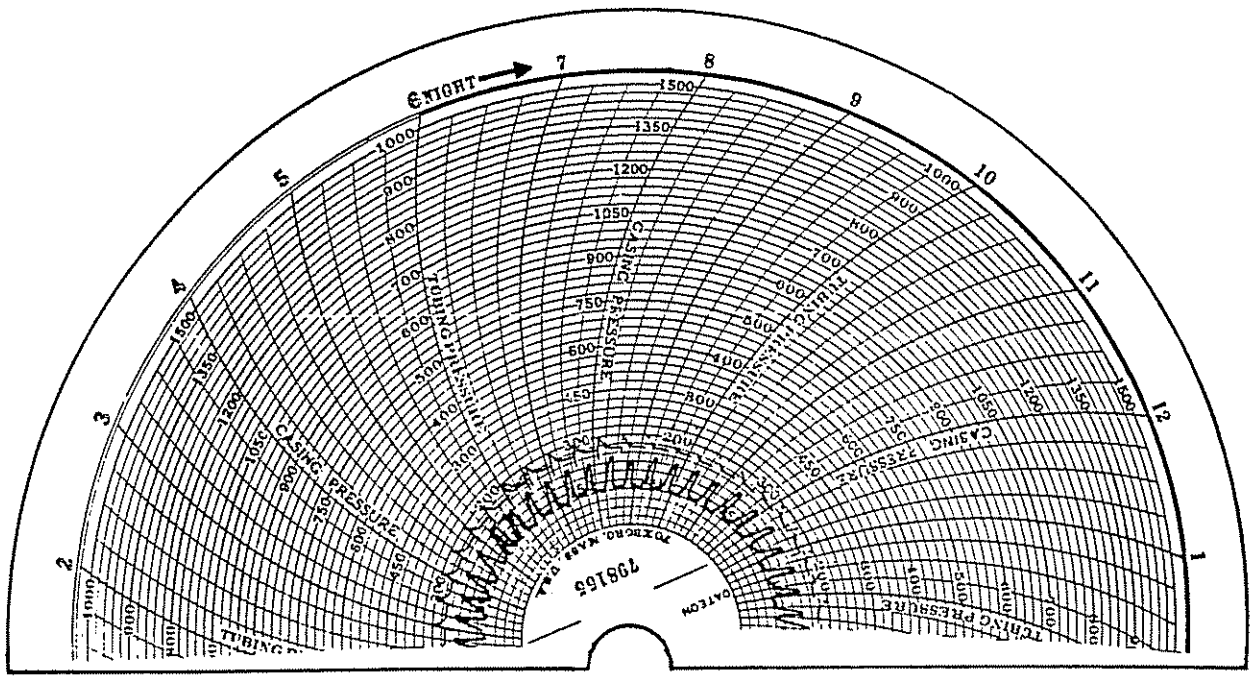


Fig. 3.621 (28) Recorder Chart—Operation: Intermittent lift. Trouble: Gas lift valve lost pressure. Remarks. Should replace valve.

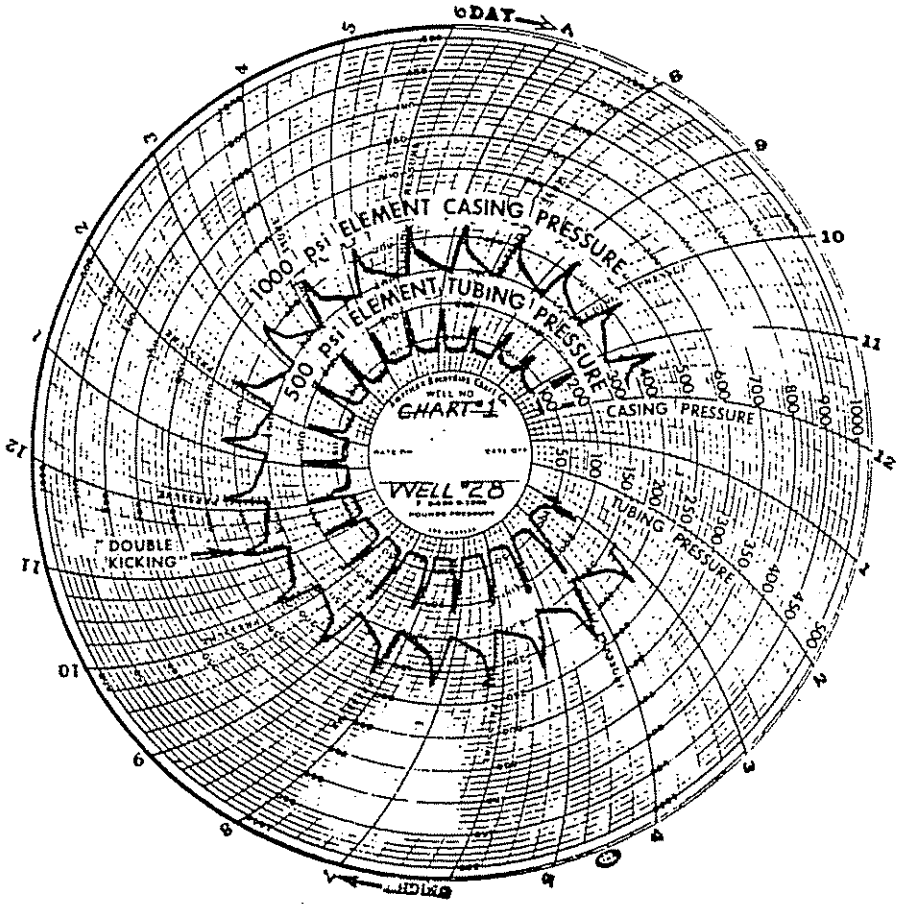


Fig. 3.621 (29) Recorder Chart—Trouble: Leak in flow string. Cure: Pull well and repair faulty equipment. Type of Well: Low pressure, low productivity. Remarks: Leak in flow string evident from extreme downward slope of casing pressure between injections. This leak continuously aerates the fluid in the tubing. Well is being "double-kicked" (2 injections of gas at very short intervals) in an effort to overcome the leak and produce some semblance of intermittent flow. The aerated fluid and extra volume of gas lift gas is responsible for the fairly low, fat tubing pressure kicks.

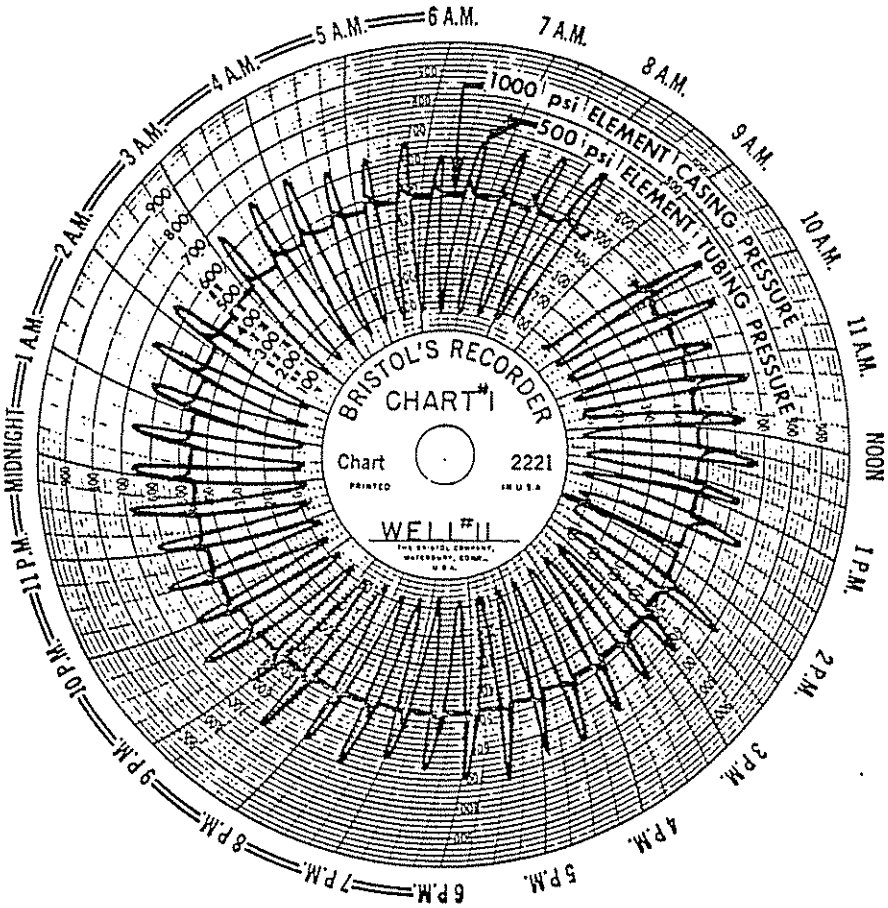


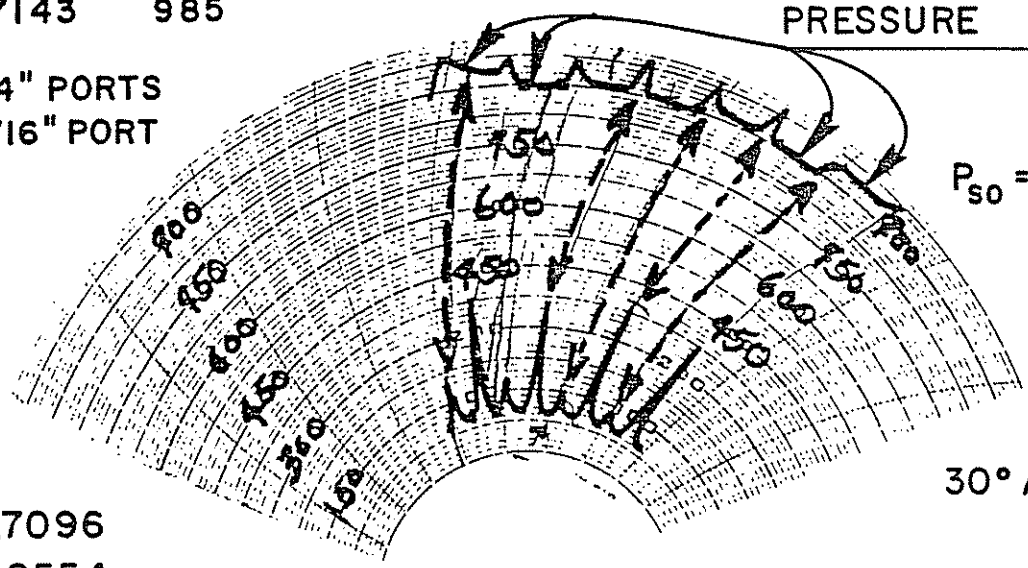
Fig. 3.621 (30) *Recorder Chart—Trouble: Excessive back pressure. Cure: Remove source of back pressure. Type of Well: High pressure, low productivity. Remarks: Excess back pressure may be due to: (1) Tubing choke too small; (2) Faulty flowline check valve (clapper broken, etc.); (3) Plugged choke; (4) Mashed or bent flowline; (5) Excessive number of right angle bends in flowline near Christmas Tree. The tubing choke in this well was too small (¼") for efficient slug type flow and as a result fluid production was low and the input gas fluid ratio was high. The problem was corrected by installing a larger tubing choke and reducing the number of cycles per day.*

NO	DEPTH	P_{TRO}
1	1548	1150
2	2724	1130
3	3551	1110
4	4124	1105
5	4478	1085
6	5109	1070
7	5773	1050
8	6474	1035
9	7143	985

q_L = 30 B/D (OIL)
P_{so} = 1100
2" TUBING
5-1/2" CSG.

APPEARS TO BE ALTERNATING VALVE ACTION - LATER DIAGNOSED TO BE VARIABLE CLOSING PRESSURE DEPENDING ON TUBING PRESSURE

8 - 1/4" PORTS
1 - 5/16" PORT



P_{so} = 1100

30° API OIL

A_b = .7096
A_b = .0554
A_b = .0797 - 5/16

CHART (1) OF (3)

20/64 SURFACE CHOKE AND LONG FLOWLINE

Fig. 3.621 (31)-A Variable Valve Closing Pressure

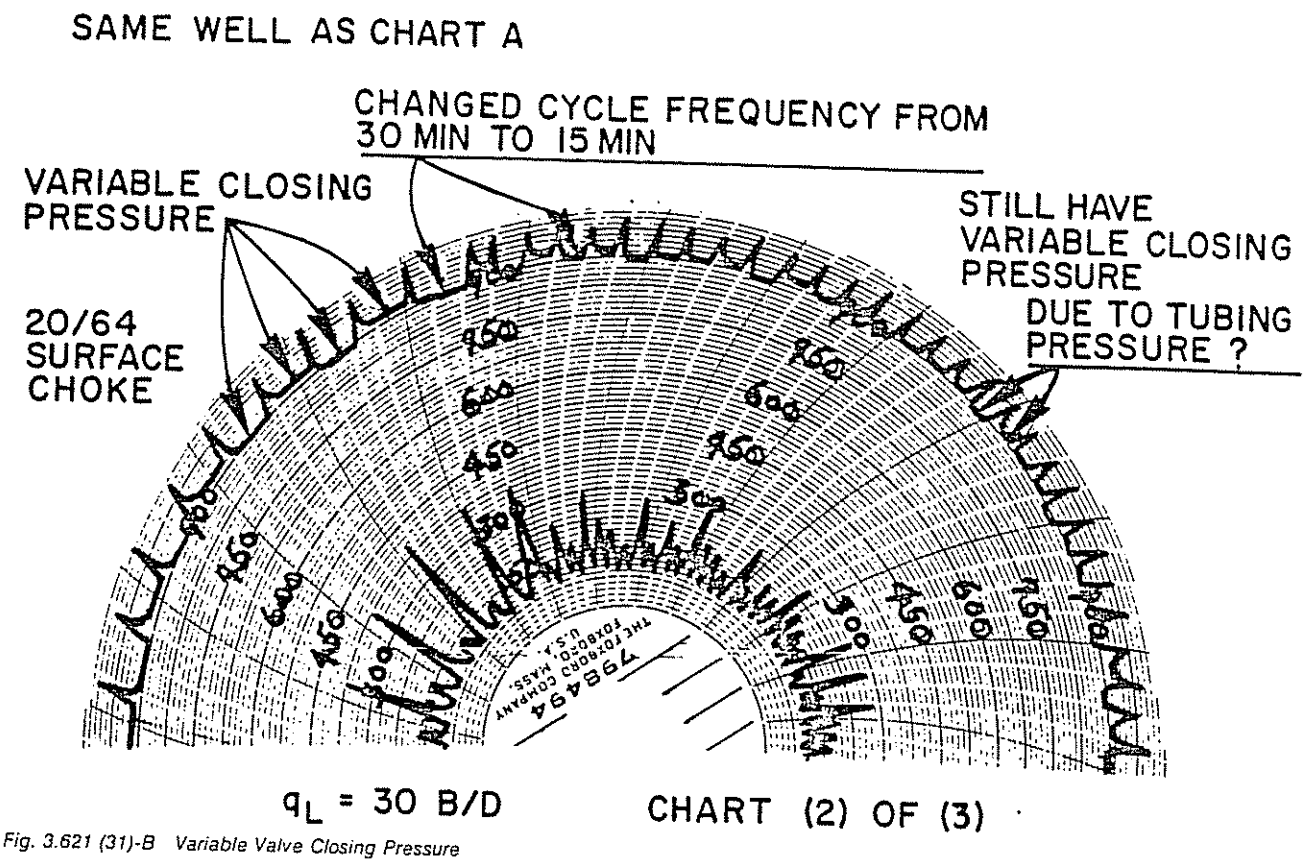


Fig. 3.621 (31)-B Variable Valve Closing Pressure

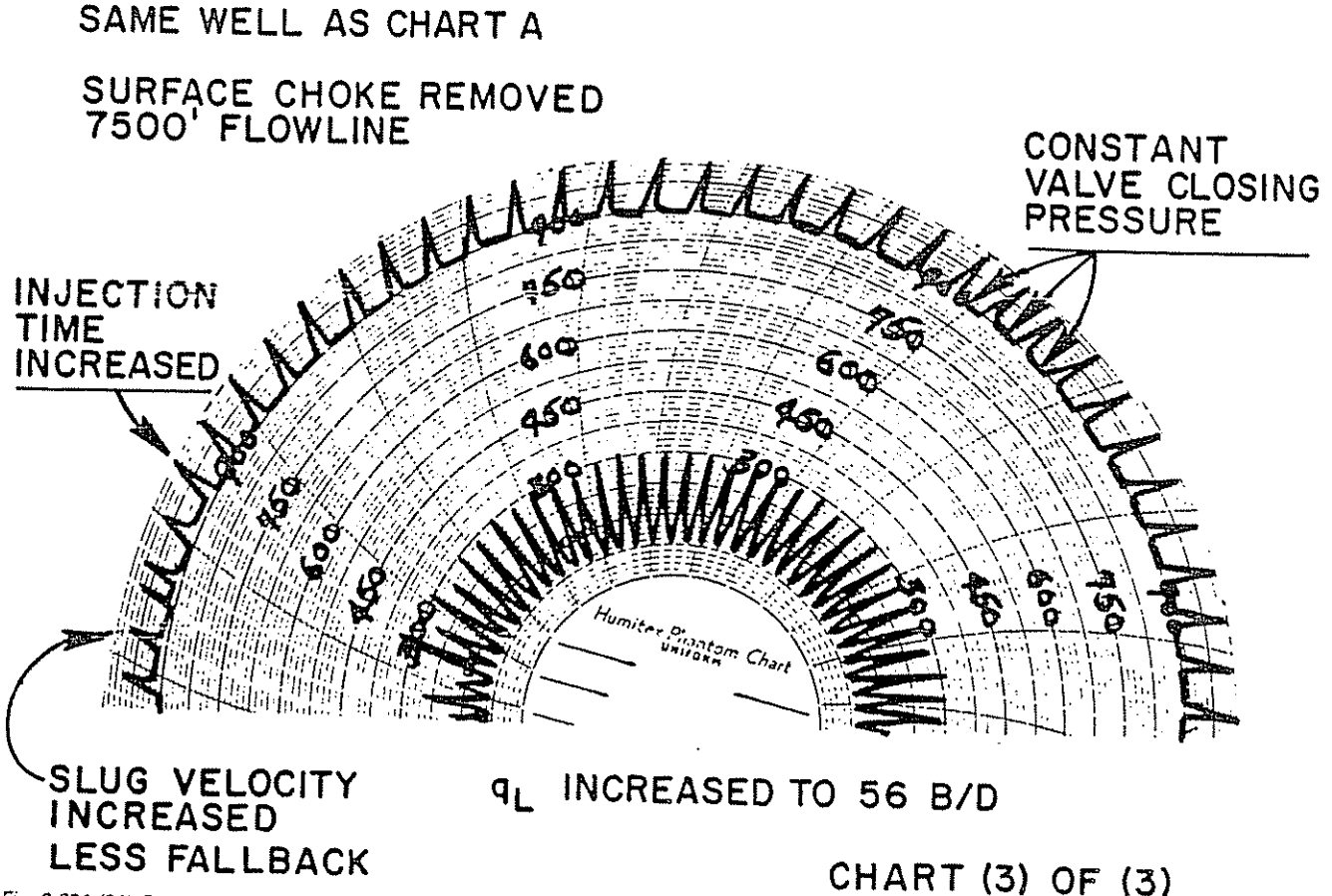
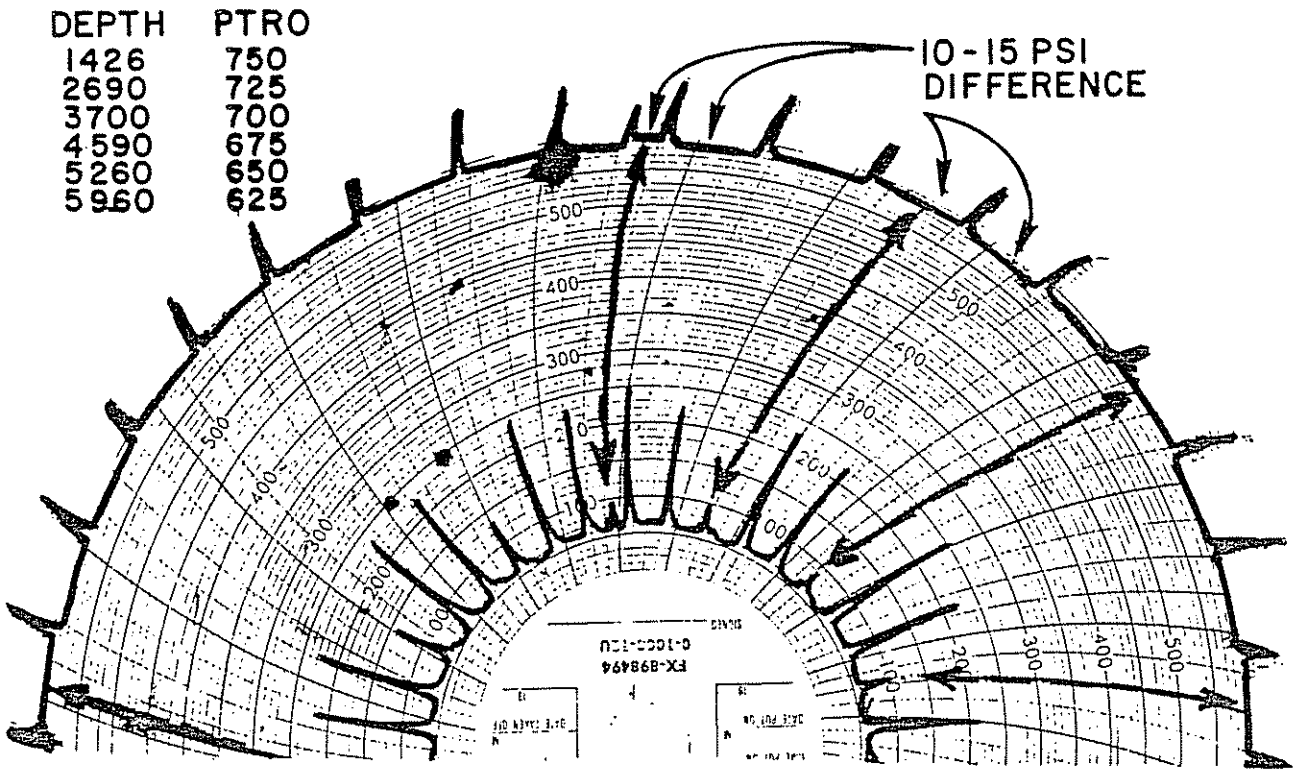


Fig. 3.621 (31)-C Elimination of Variable Closing Pressure



1/4" PORTS
q_L = 40 B/D 60% WATER
30° API CRUDE 2-3/8" O.D TUBING
5-1/2" 17 #/FT CASING γ_g = 0.65
TEMP 7660 - 160°F MMA MANDRALS

Fig. 3.621 (32) Variable Closing Pressure on Same Valve

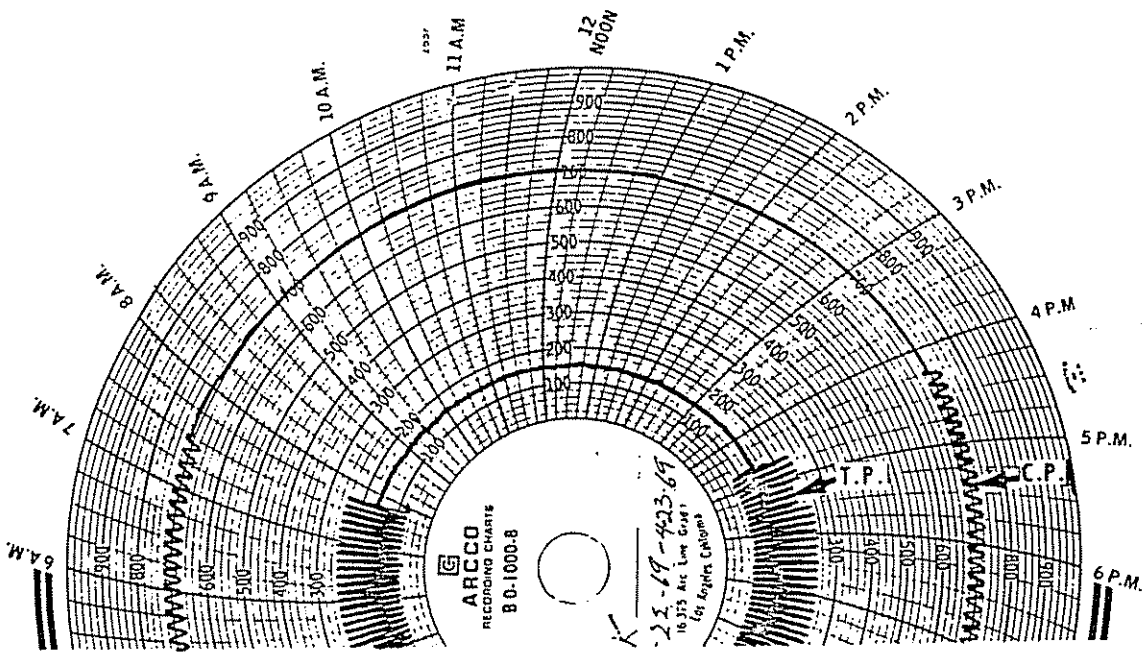


Fig. 3.621 (33)-A Intermittent and Continuous Flow Operation with Same Fluid Operated Valve

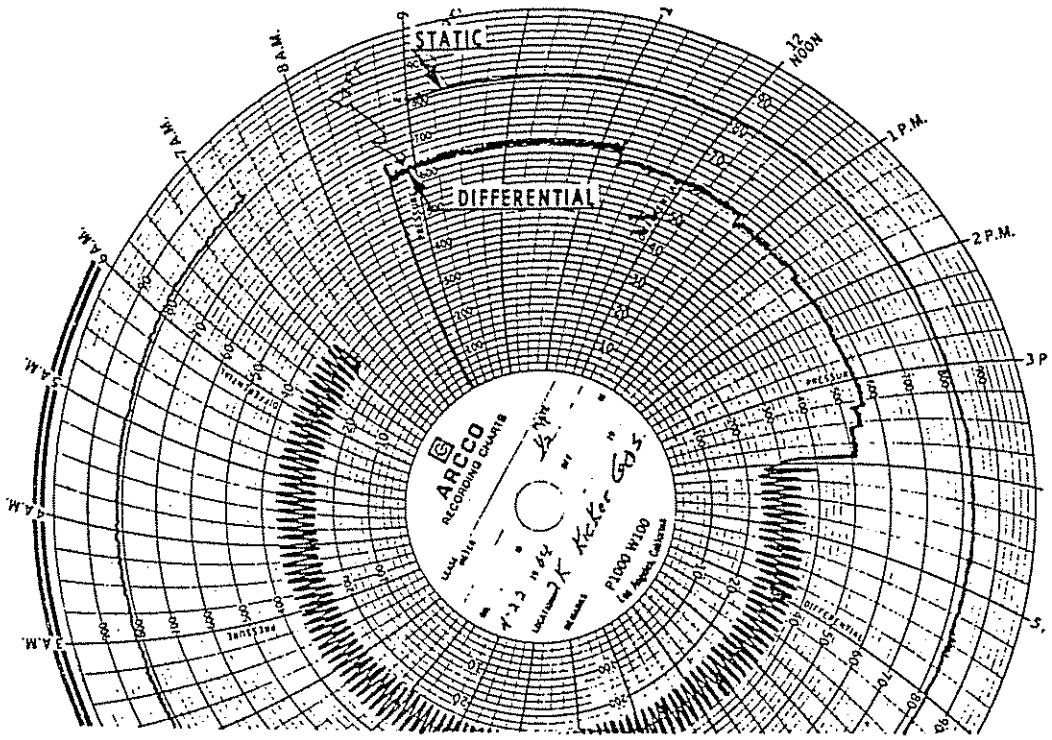


Fig. 3.621 (33)-B Gas Measurement Chart for Fig. 3.621 (33)-A shows both intermittent and continuous flow for same fluid valve. Port size too large for continuous flow. Production highest on intermittent lift. Can be made to continuous flow by injecting more gas.

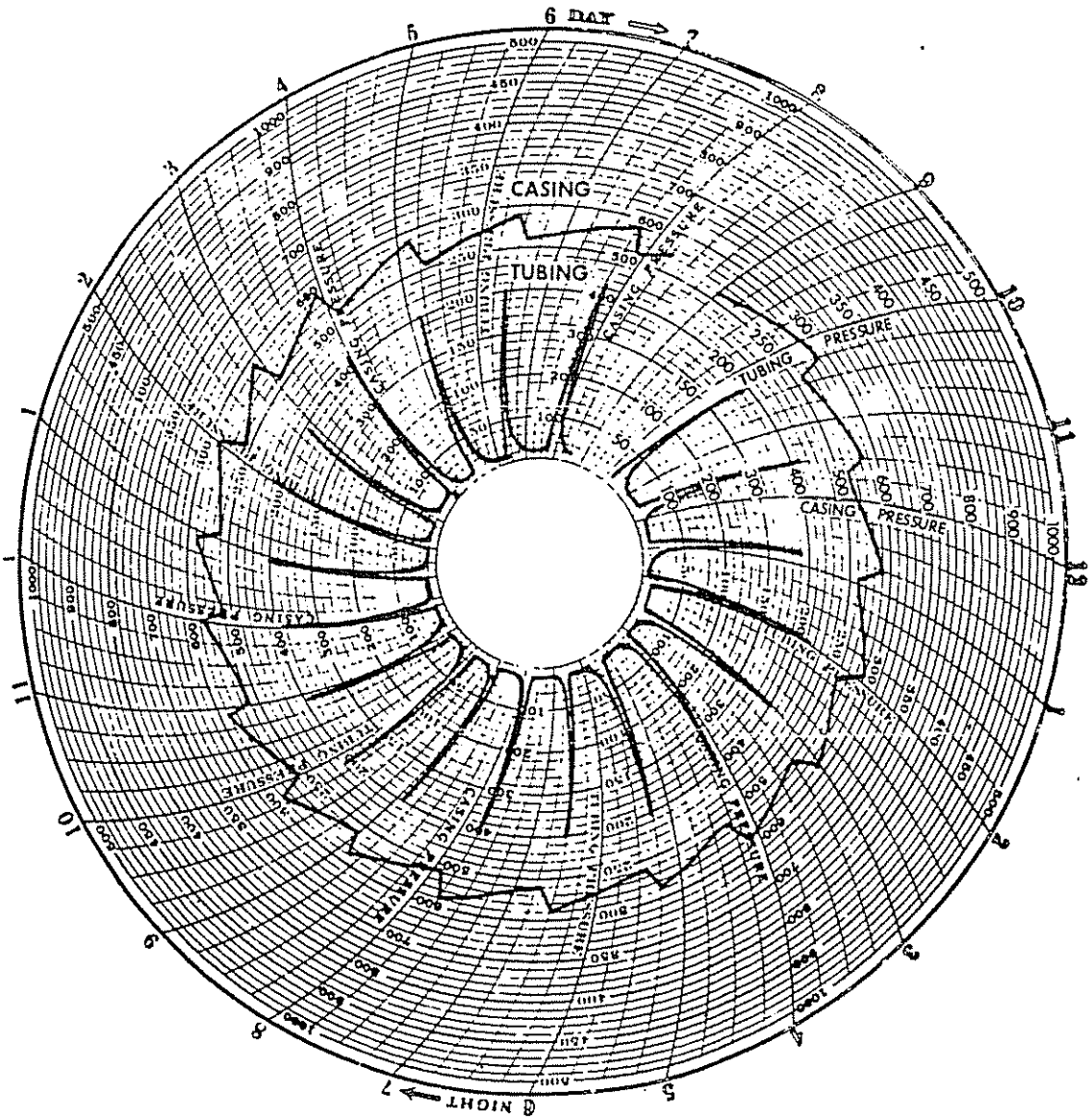


Fig. 3.621 (34) Choke Control—Continuous Injection of Gas for Intermittent Lift.

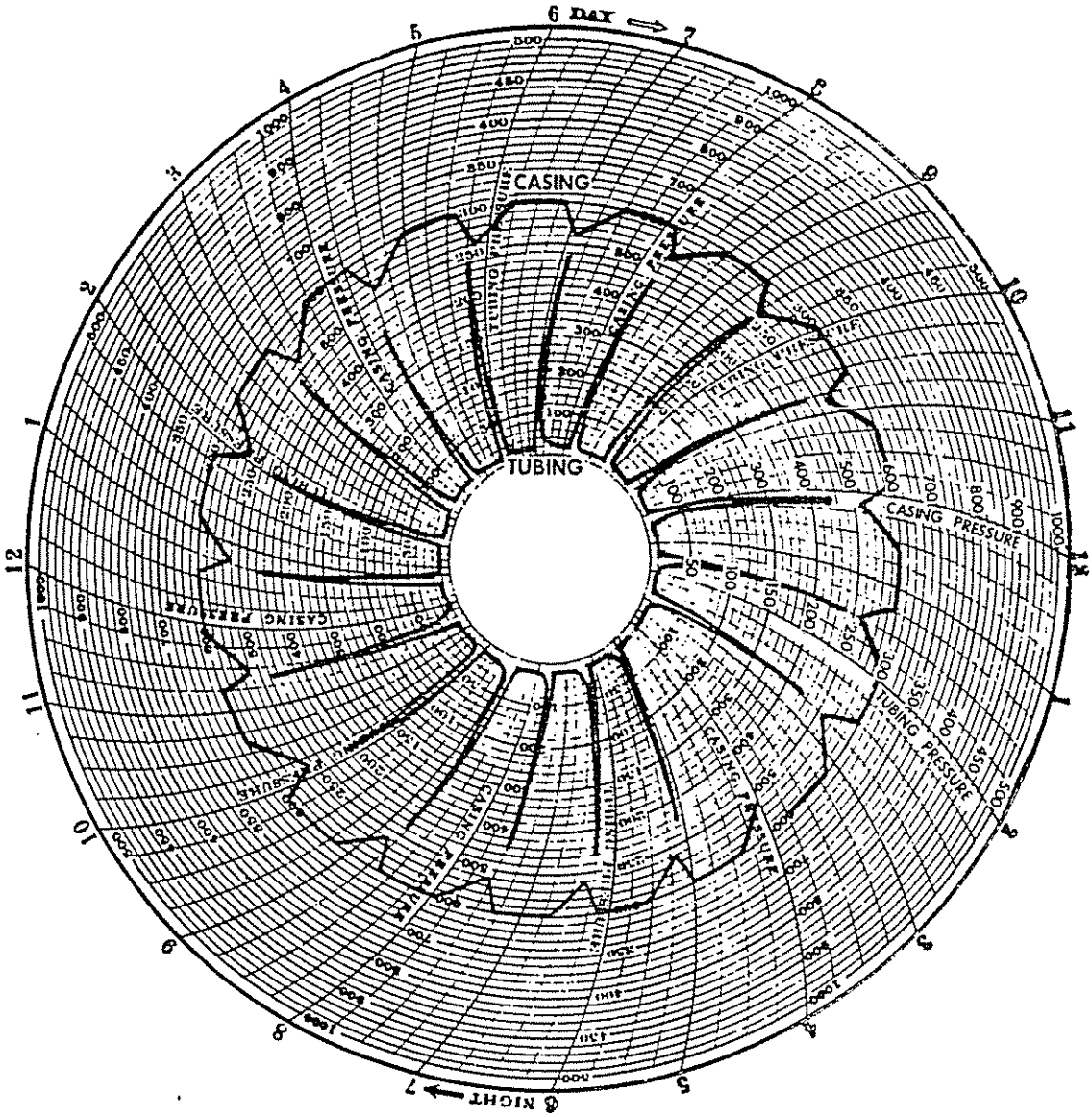


Fig. 3.621 (35)-A Choke and Regulator Control

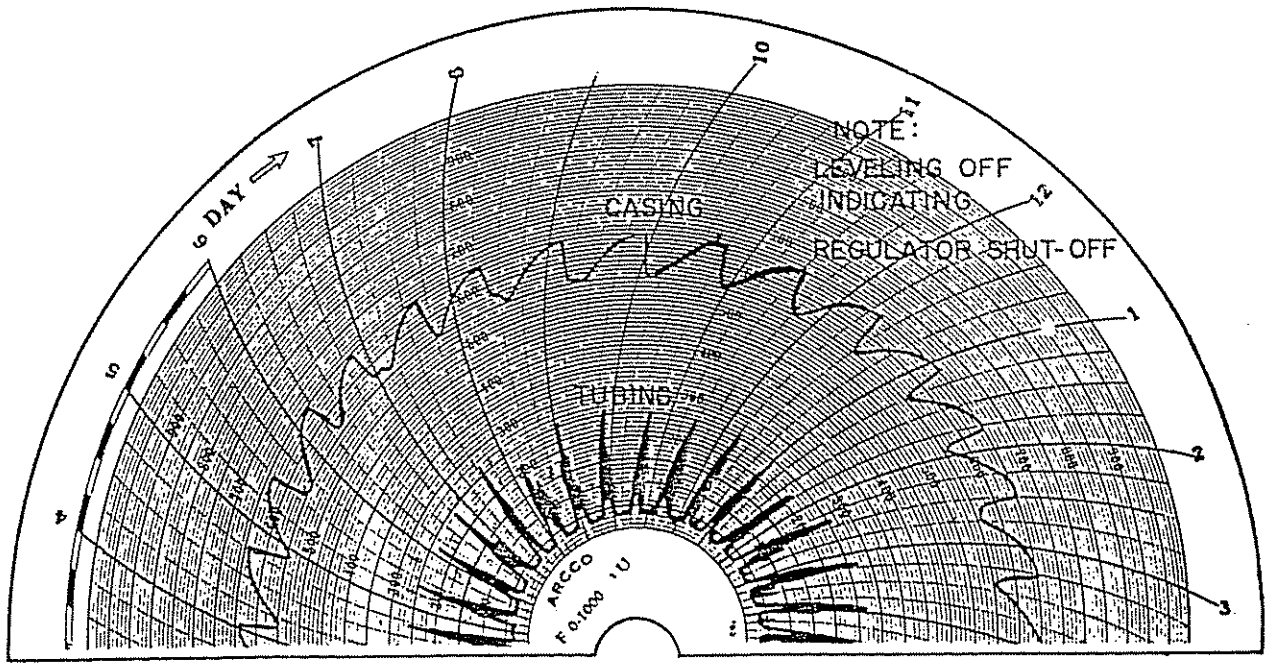


Fig. 3.621 (35)-B Recorder Chart—Operation: Intermittent lift, choke control with regulator shutoff (shuts on 650 psi on casing). Trouble: Producing into surface restriction. Remarks: Good operation for valve with spread on surface choke control—system should be investigated for surface restriction—might increase cycle frequency.

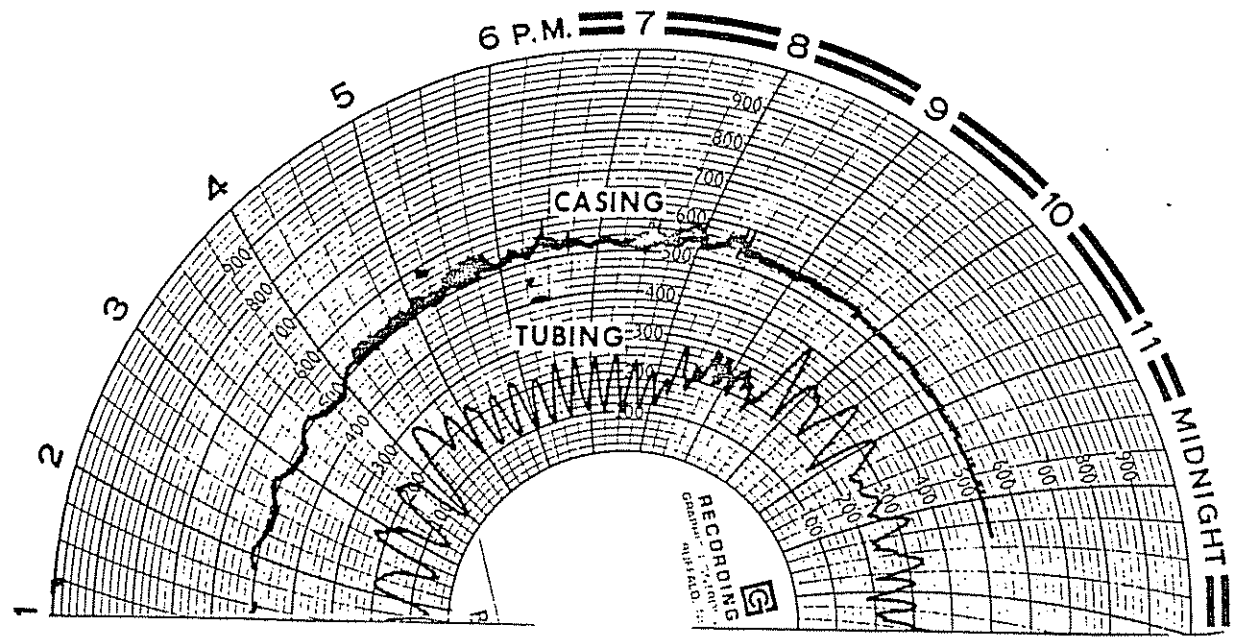


Fig. 3.621 (36) Choke Control for Intermittent Lift. (Not enough spread in gas lift valve. Well trying to continuous flow) .

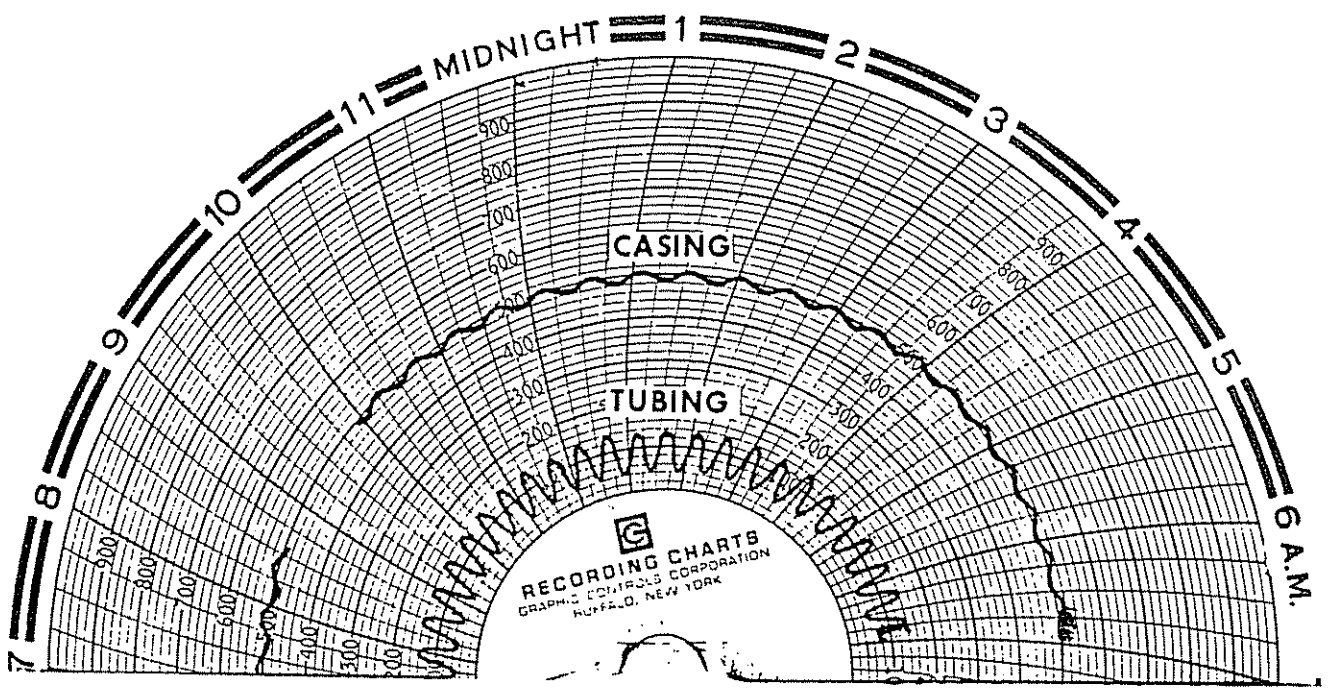
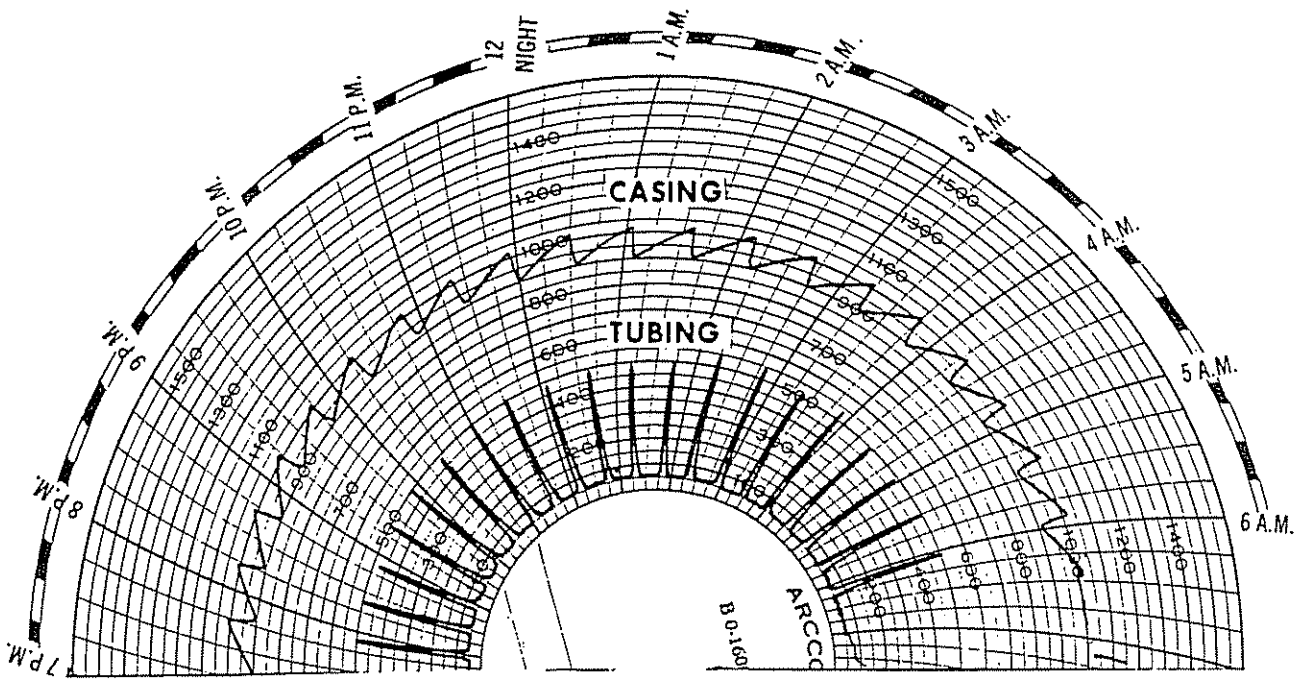
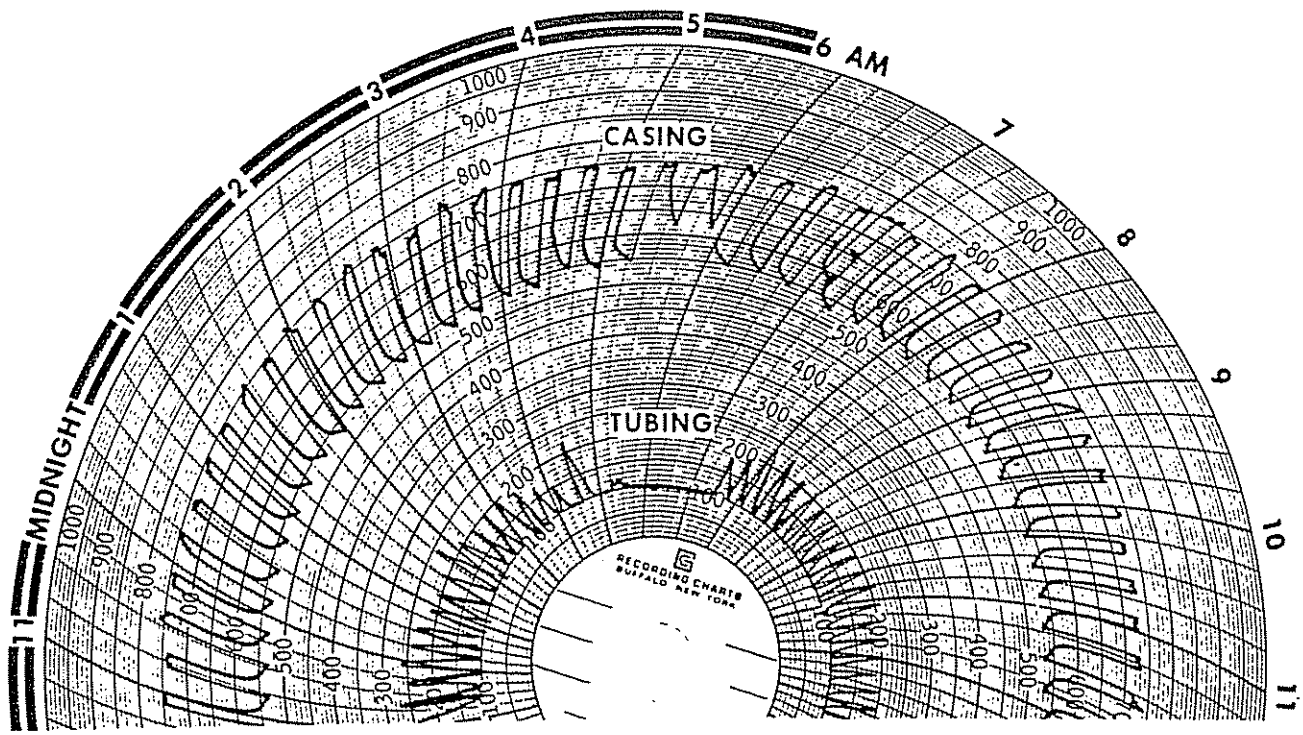


Fig. 3.621 (37) Choke Control for Intermittent Flow. (Not enough spread in valve. Cycles should be faster.)



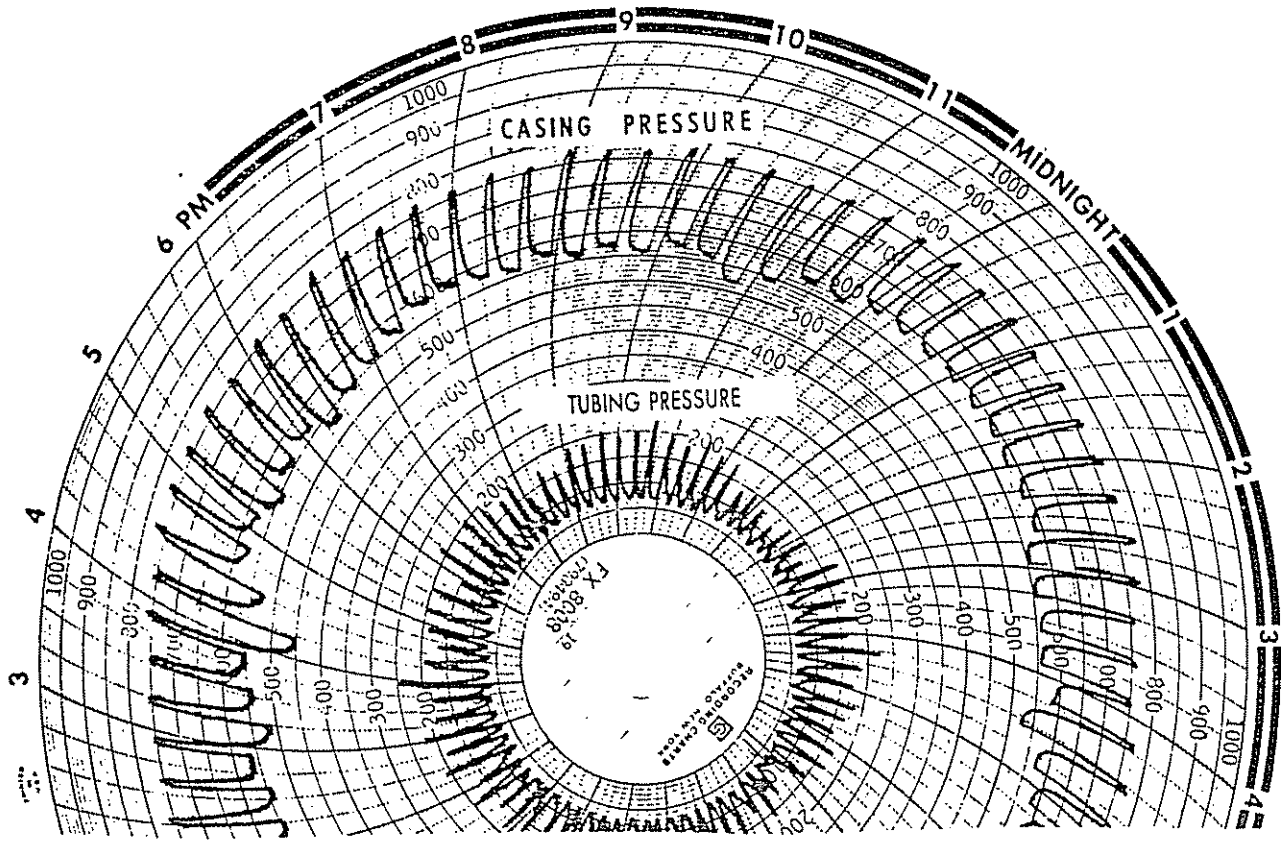
q = 150 B/D (80% WATER)

Fig. 3.621 (38) Choke Control. (Good operation.)



CASING	2½ "	11 BOPD
TUBING	1½ "	95 BWPD

Fig. 3.621 (39) Time Cycle Control. (Ultra slim hole completion. Note high casing pressure buildup.)



CASING	2½"	25 BOPD
TUBING	1½"	100 BWPD

Fig. 3.621 (40) Time Cycle Control. (Ultra slim hole completion.)

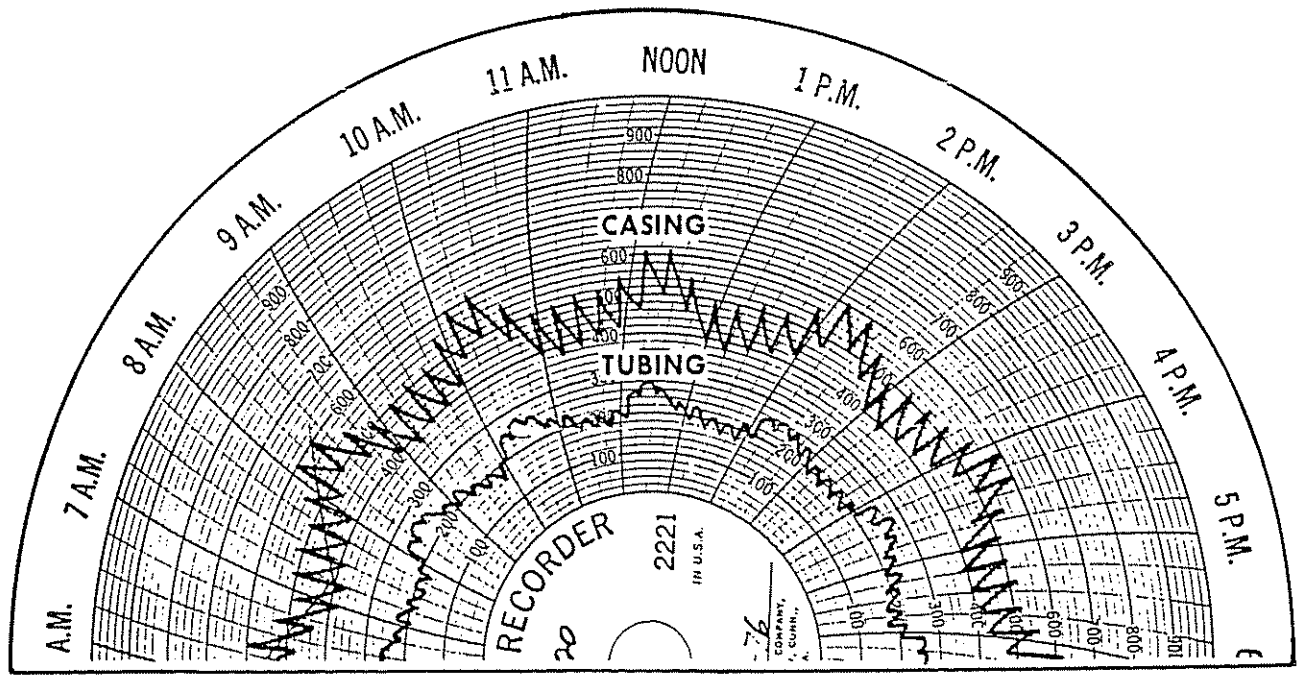


Fig. 3.621 (41) Leak in Valve or Tubing. Leak near liquid level because well producing some oil and casing pressure buildup shows fluid feed-in.

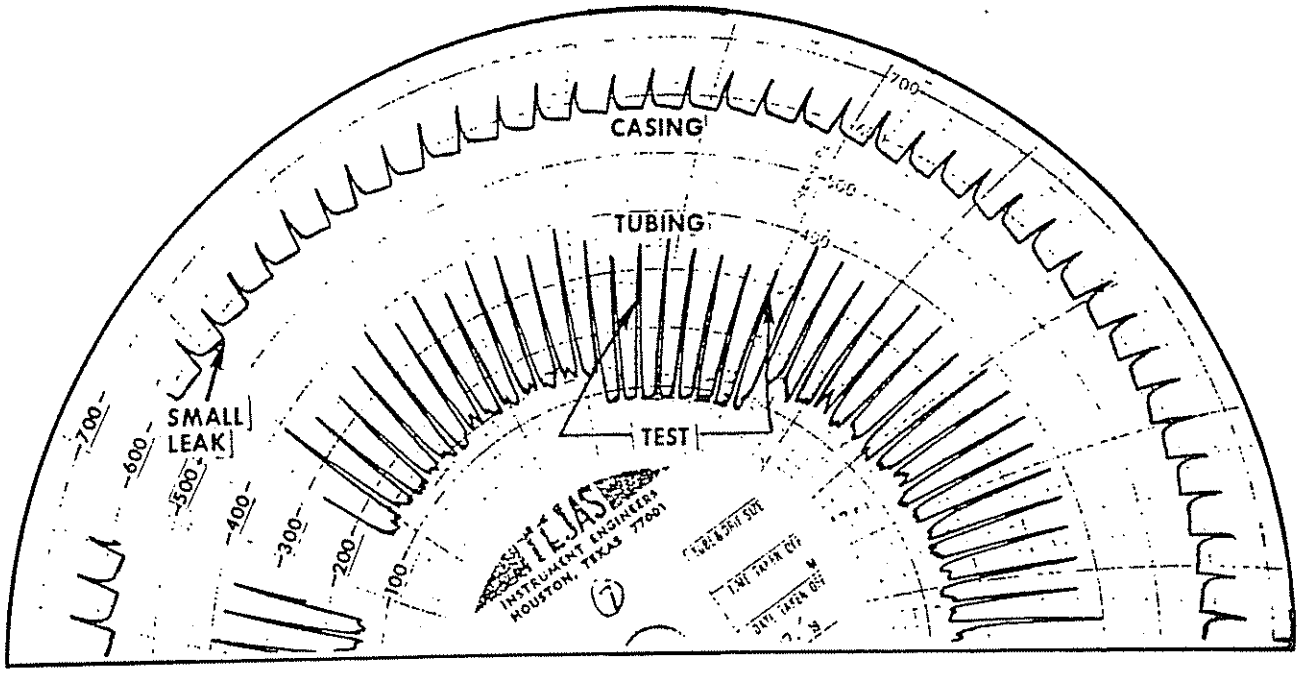


Fig. 3.621 (42) Recorder Chart—Tested in lower pressure separator. Care should be taken to maintain same pressure in test separator.

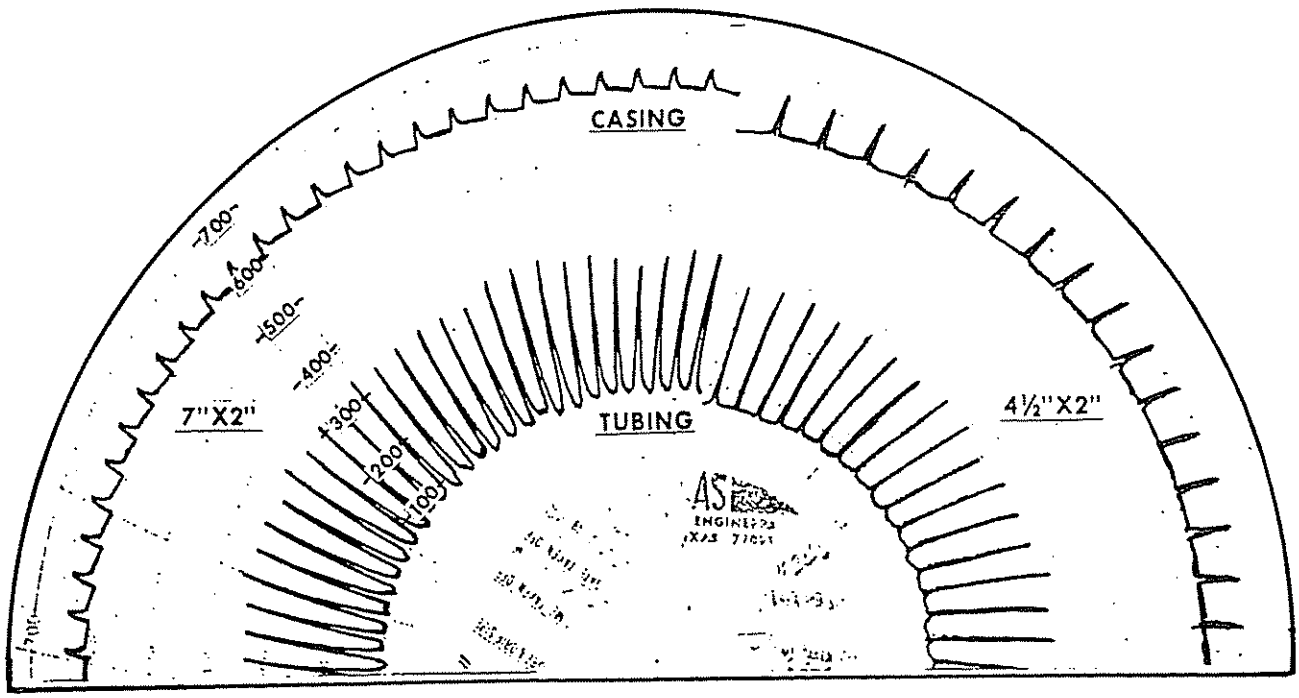


Fig. 3.621 (43) Different Spread for 7" Casing as Compared to 4½" Casing

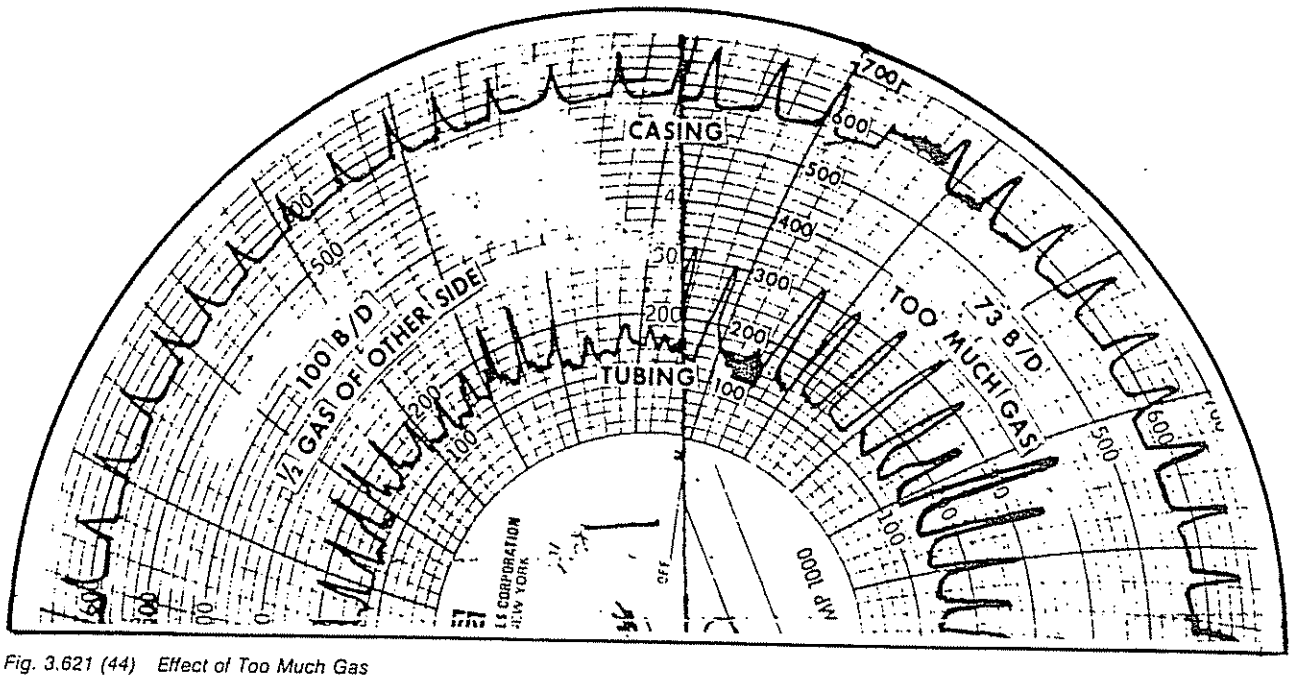


Fig. 3.621 (44) Effect of Too Much Gas

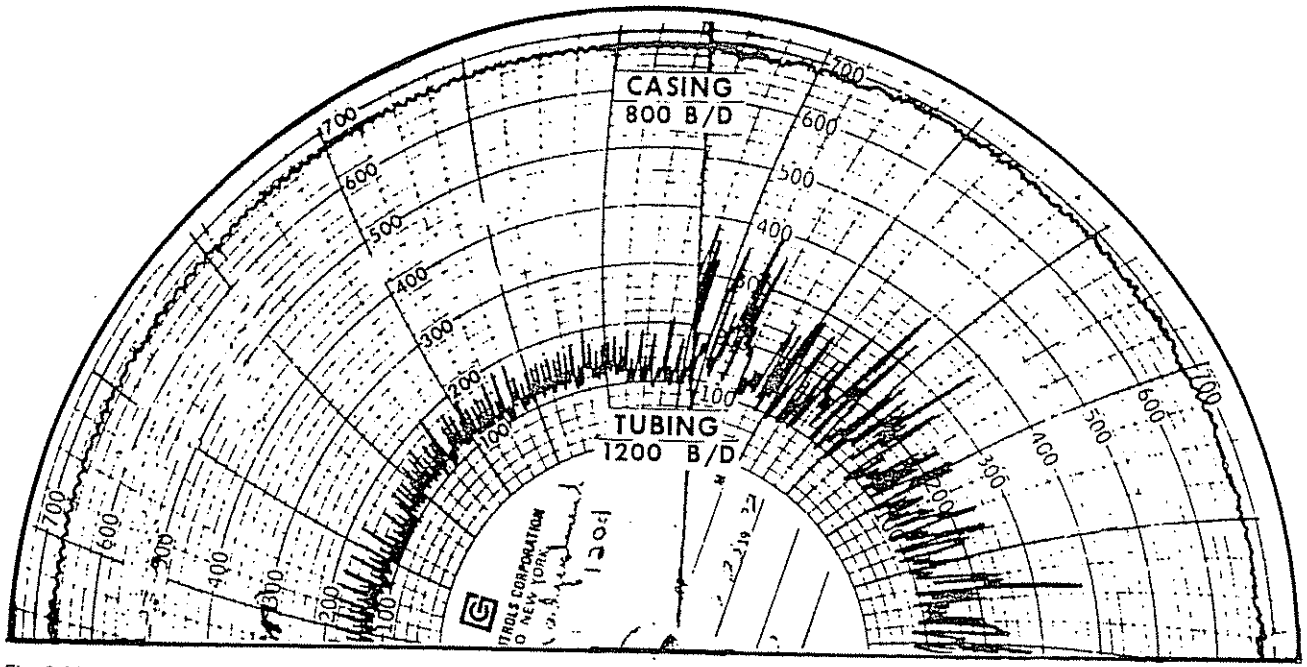


Fig. 3.621 (45) Effect of Injecting Gas through 1/64" Choke as Compared to Cracked 2" Valve

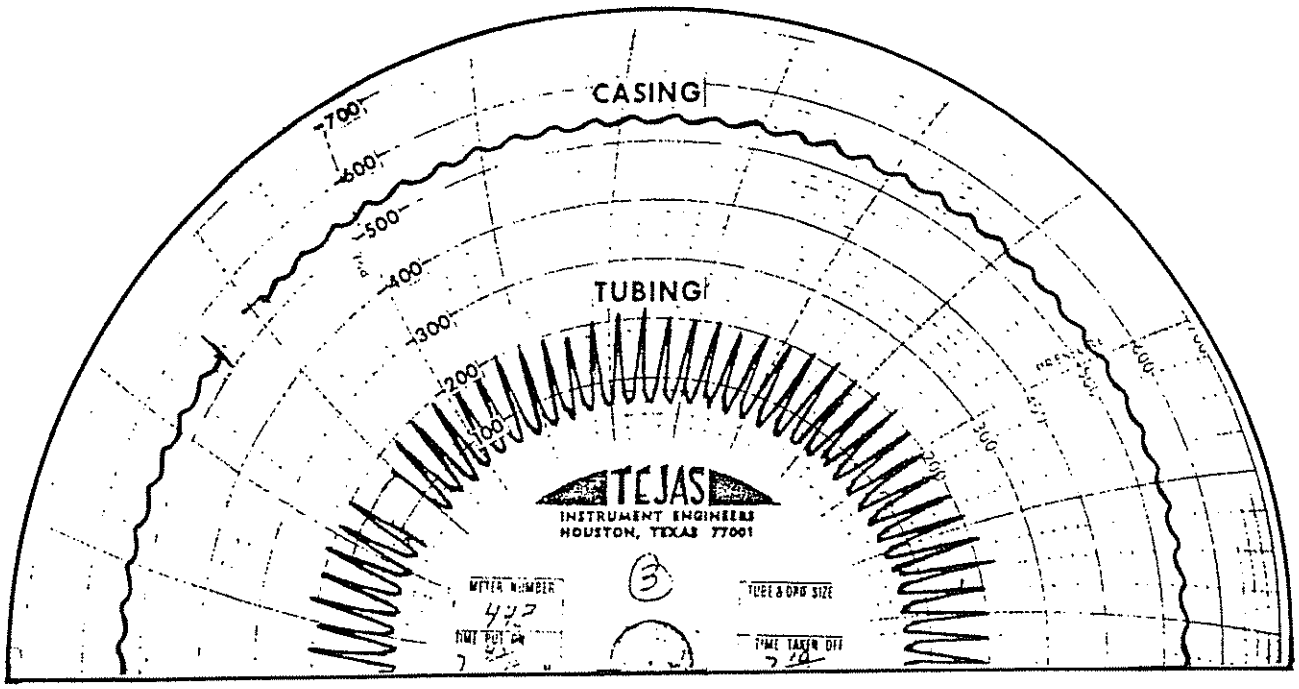


Fig. 3.621 (46) Injection of Gas Continually for Intermittent Lift (Choke Control)

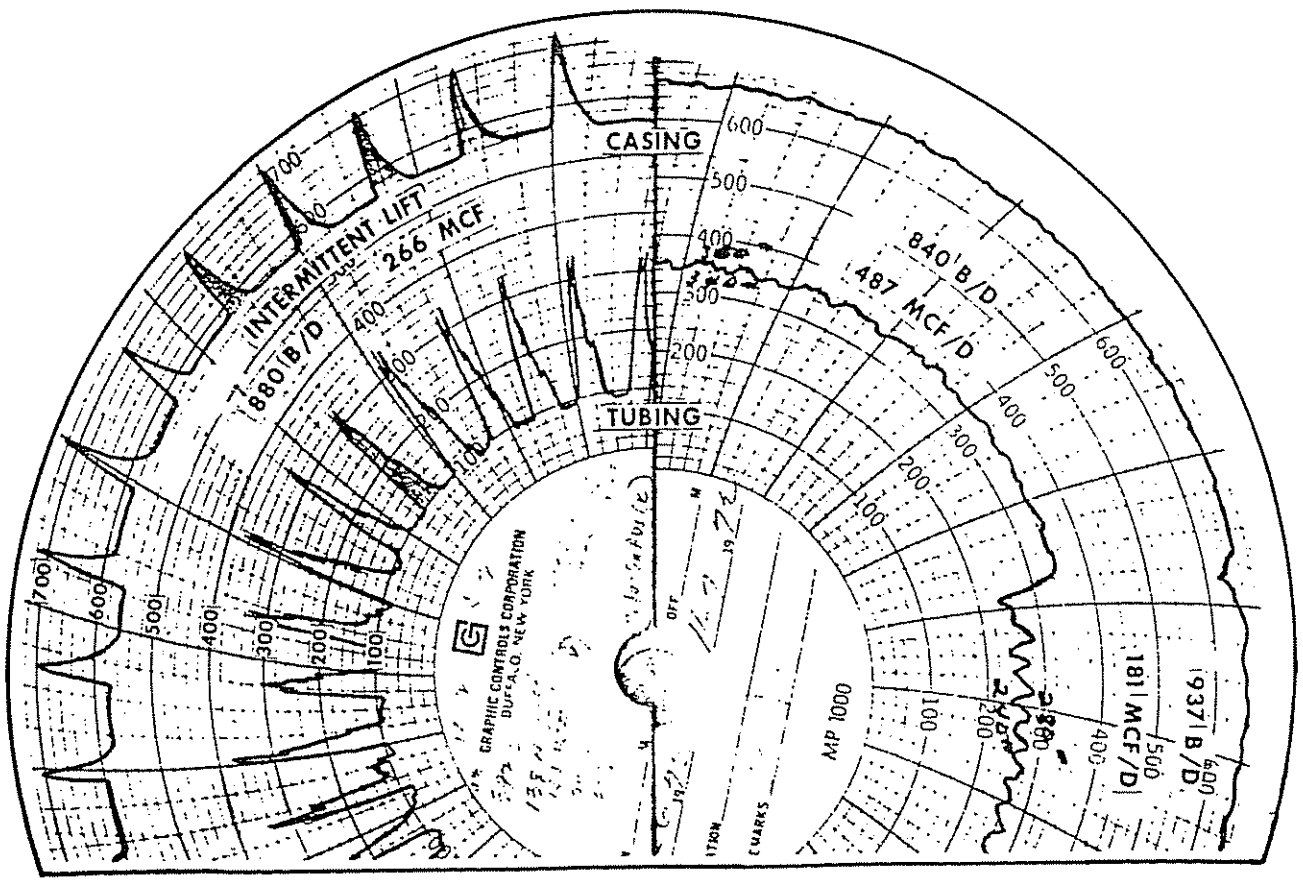


Fig. 3.621 (47) Changed from Intermittent Lift to Continuous Flow

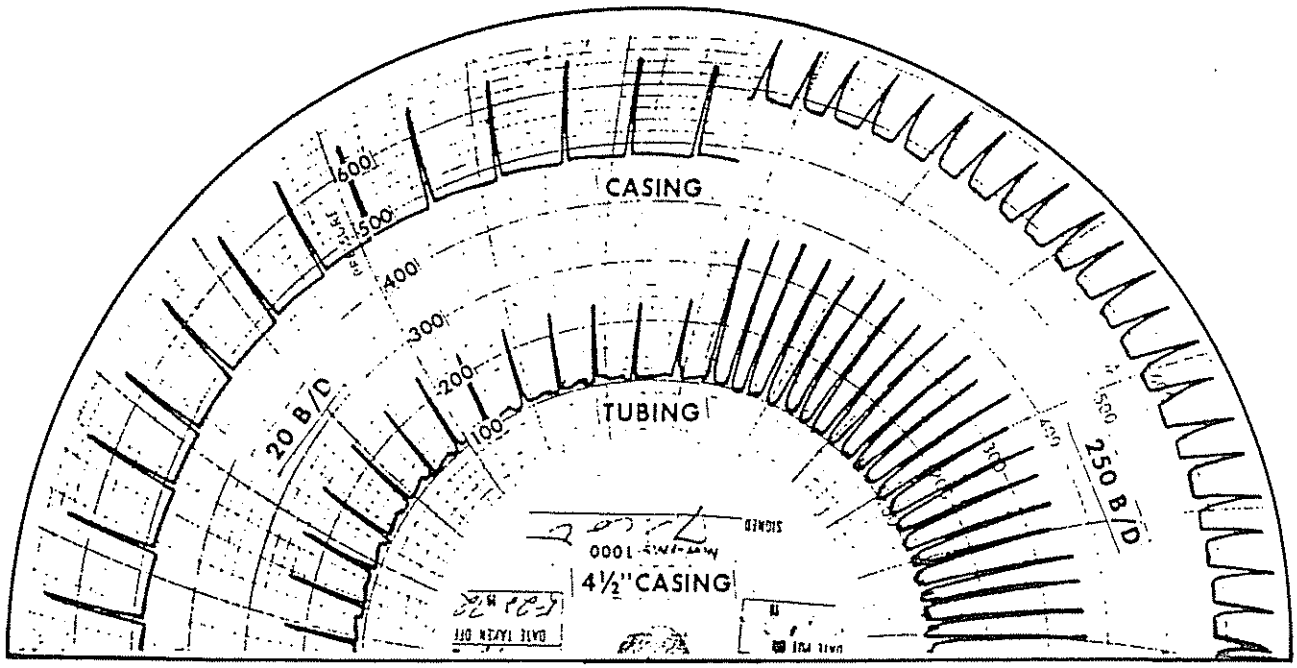


Fig. 3.621 (48) Comparison of Lifting Light Tubing Load for Same Valve Seat Size. Casing spread decrease with higher rate and heavier tubing load.

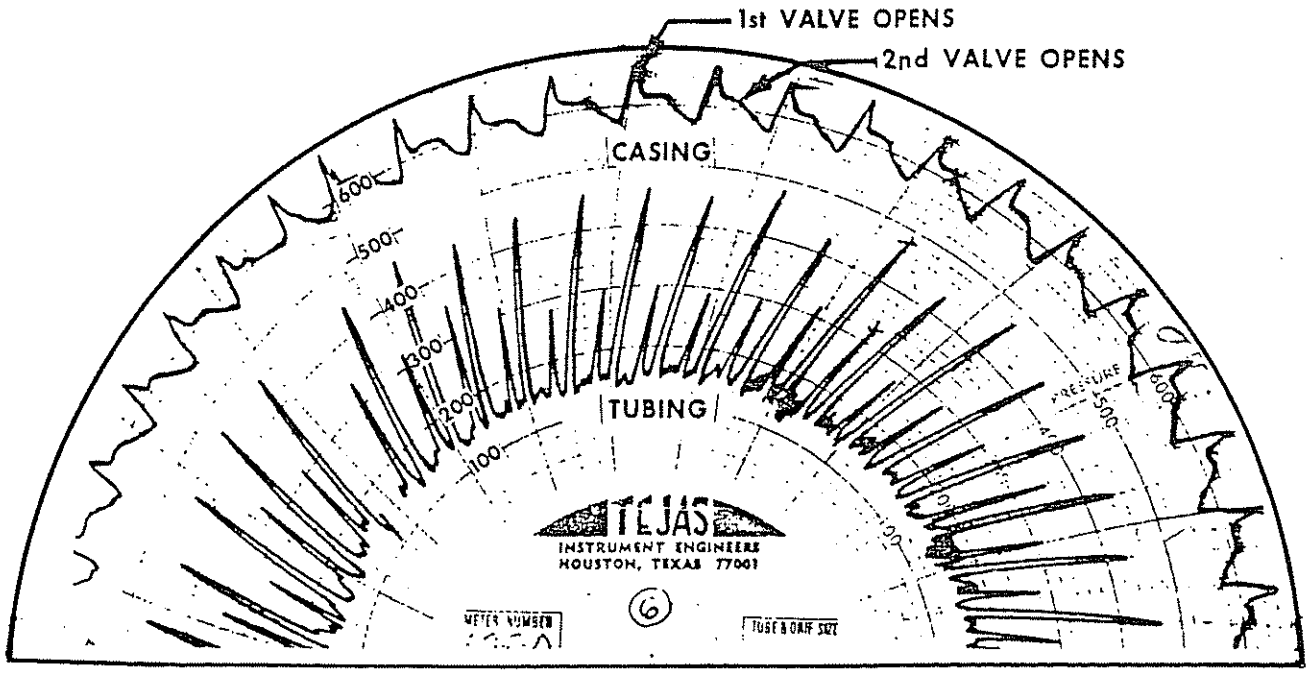


Fig. 3.621 (49) Opening Two Gas Lift Valves

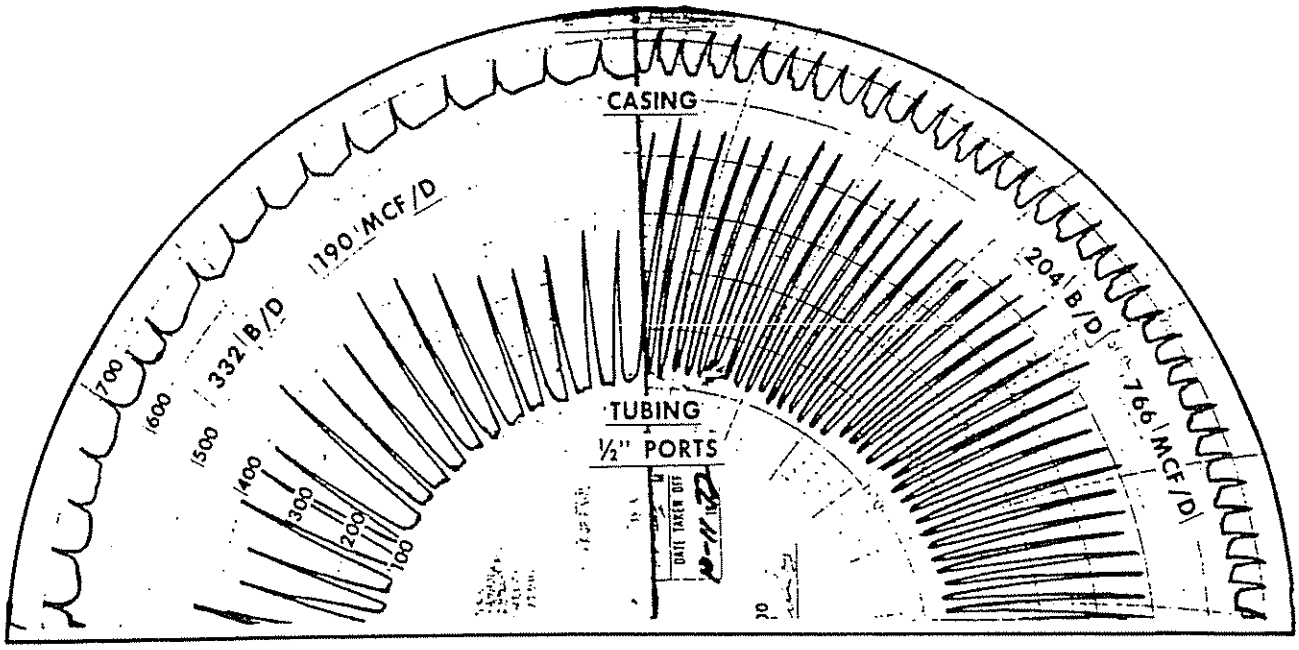


Fig. 3.621 (50) Comparison of Cycle Frequencies—Slower Cycles Make More Production

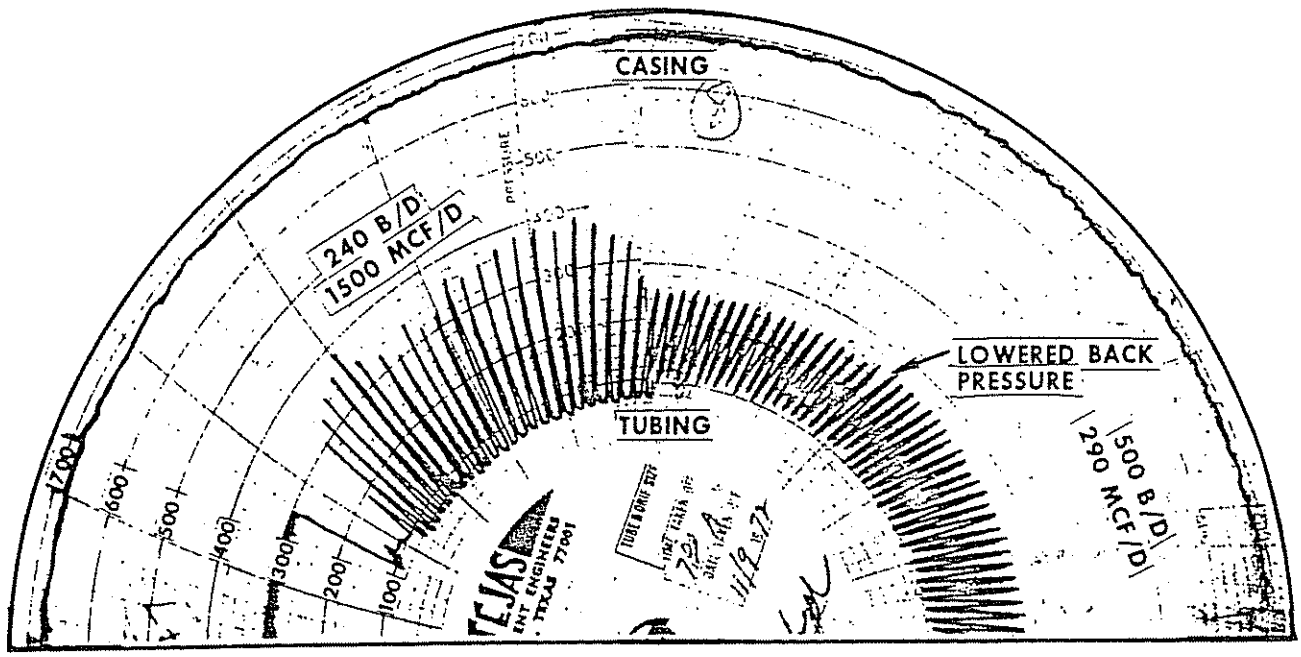


Fig. 3.621 (51) Effect of Decreasing Back Pressure

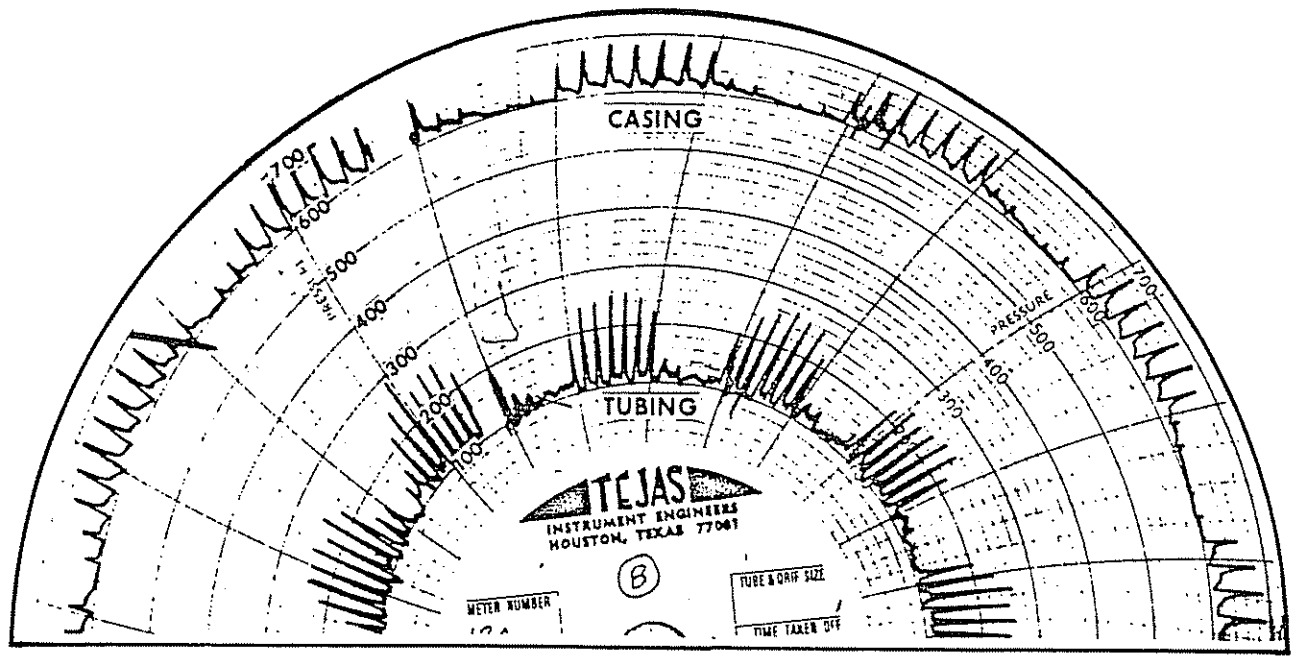


Fig. 3.621 (52) Bad Wheel on Clock or Intermitter

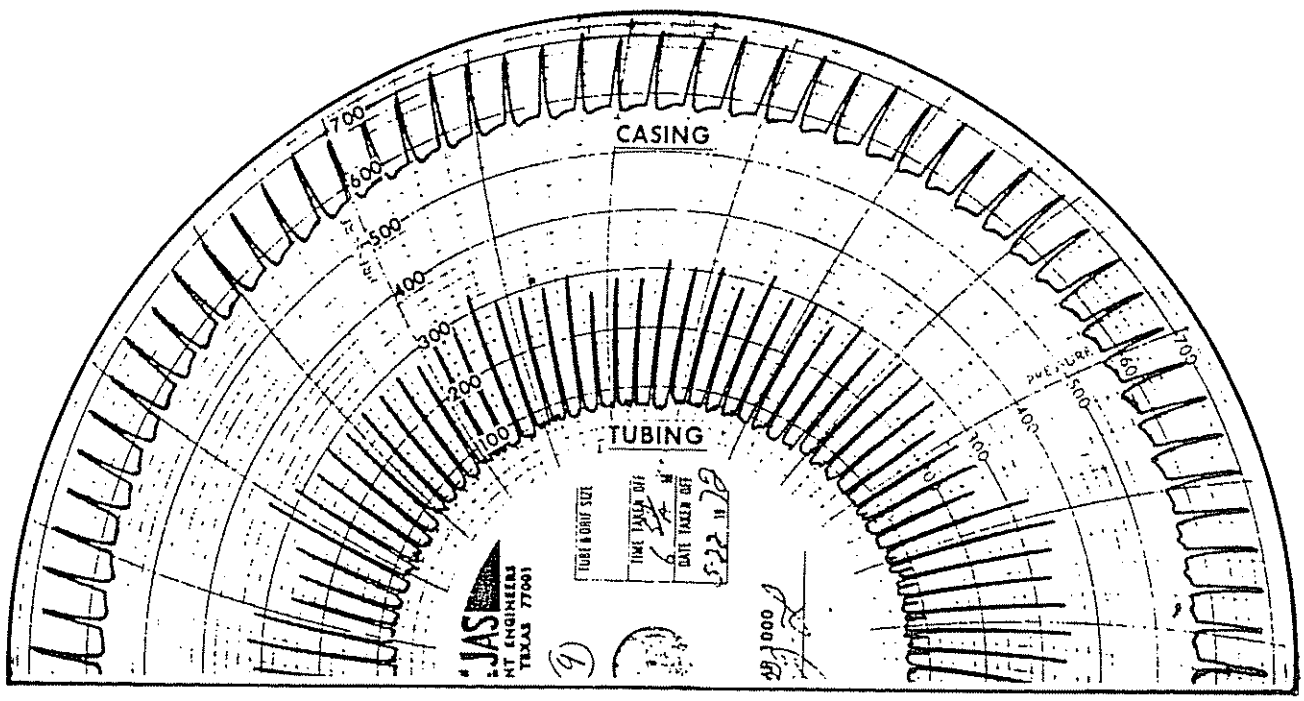


Fig. 3.621 (53) Increase in Casing Pressure between Cycles Shows Controller Seat Leak

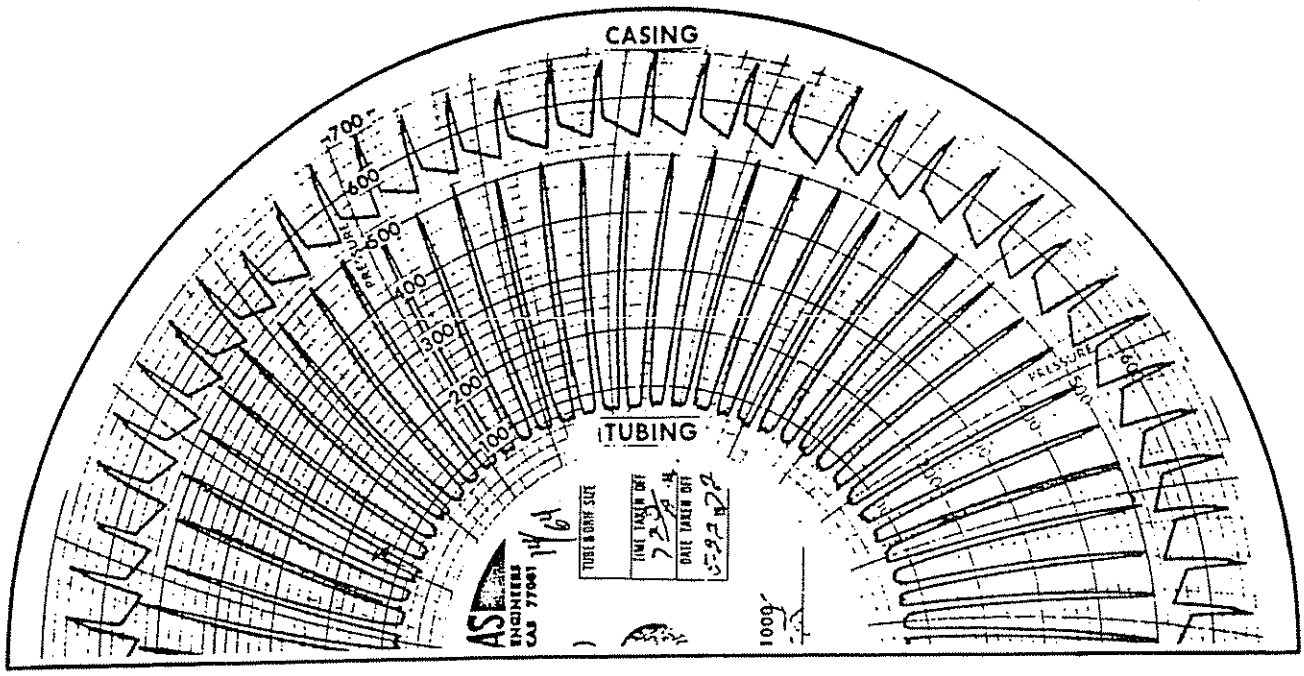


Fig. 3.621 (54) Valve or Tubing Leak and High Back Pressure

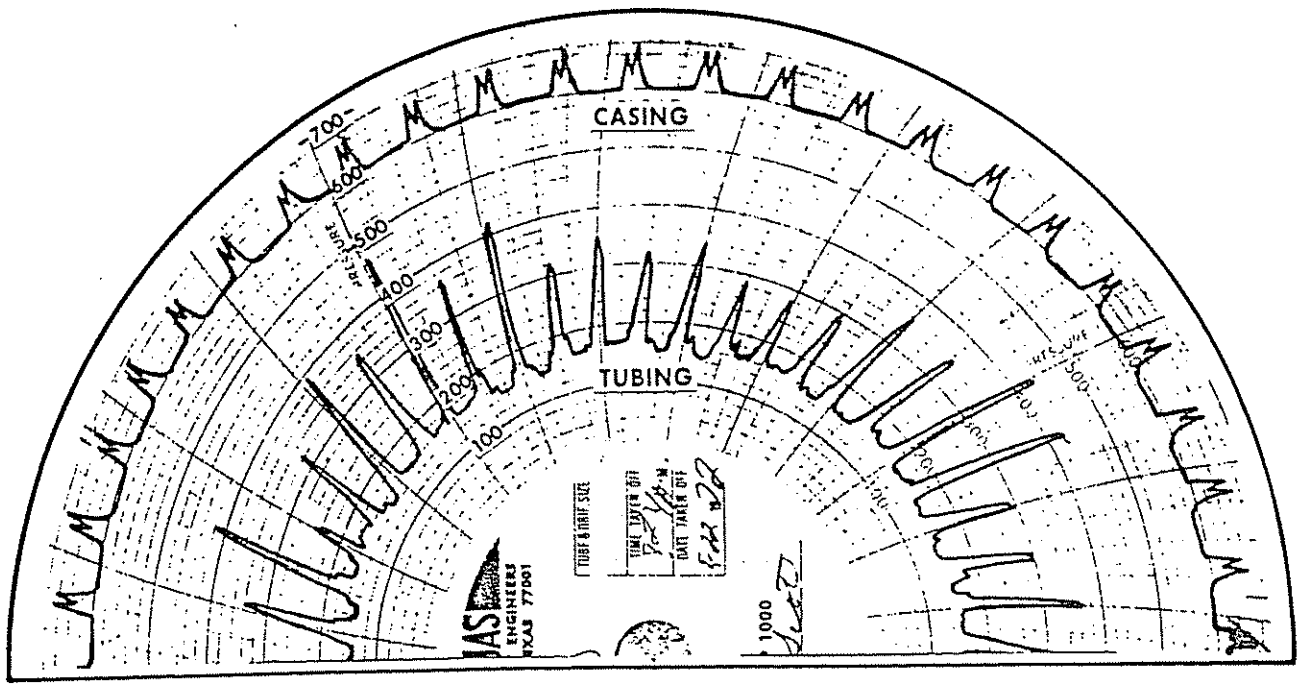


Fig. 3.621 (55) High Back Pressure Two Gas Injections Per Cycle

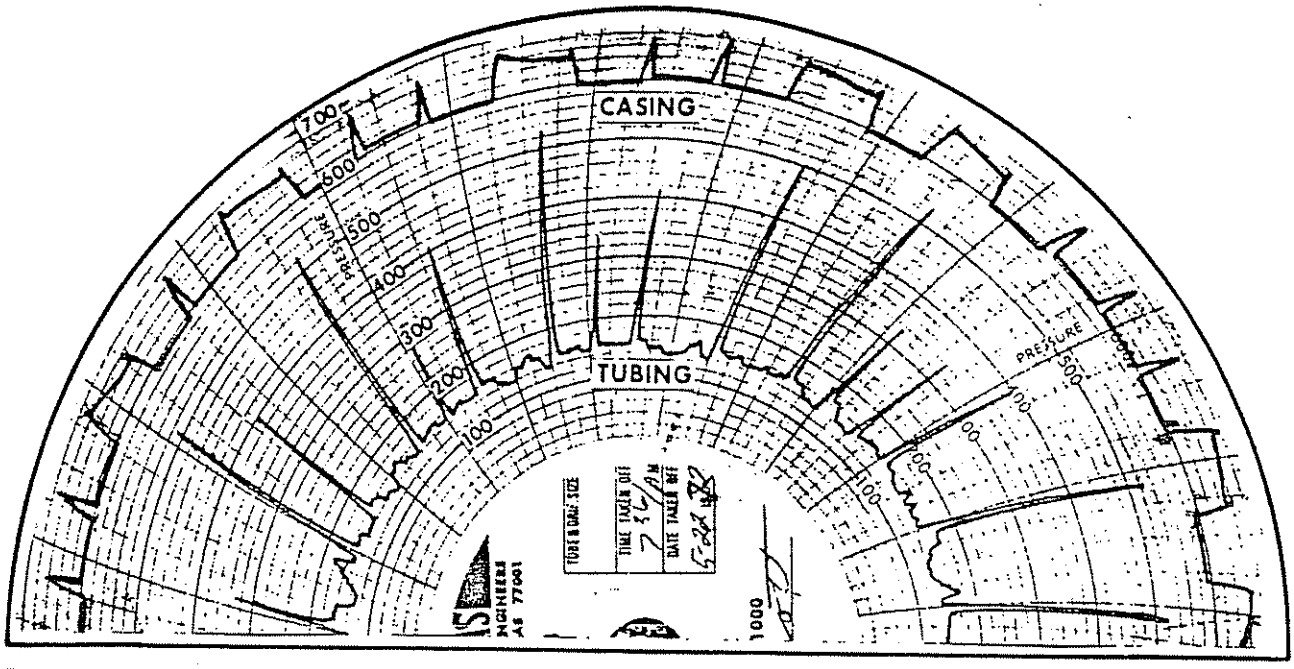


Fig. 3.621 (56) Not Enough Gas or Not Enough Tubing Load to Open Valve Each Injection Cycle

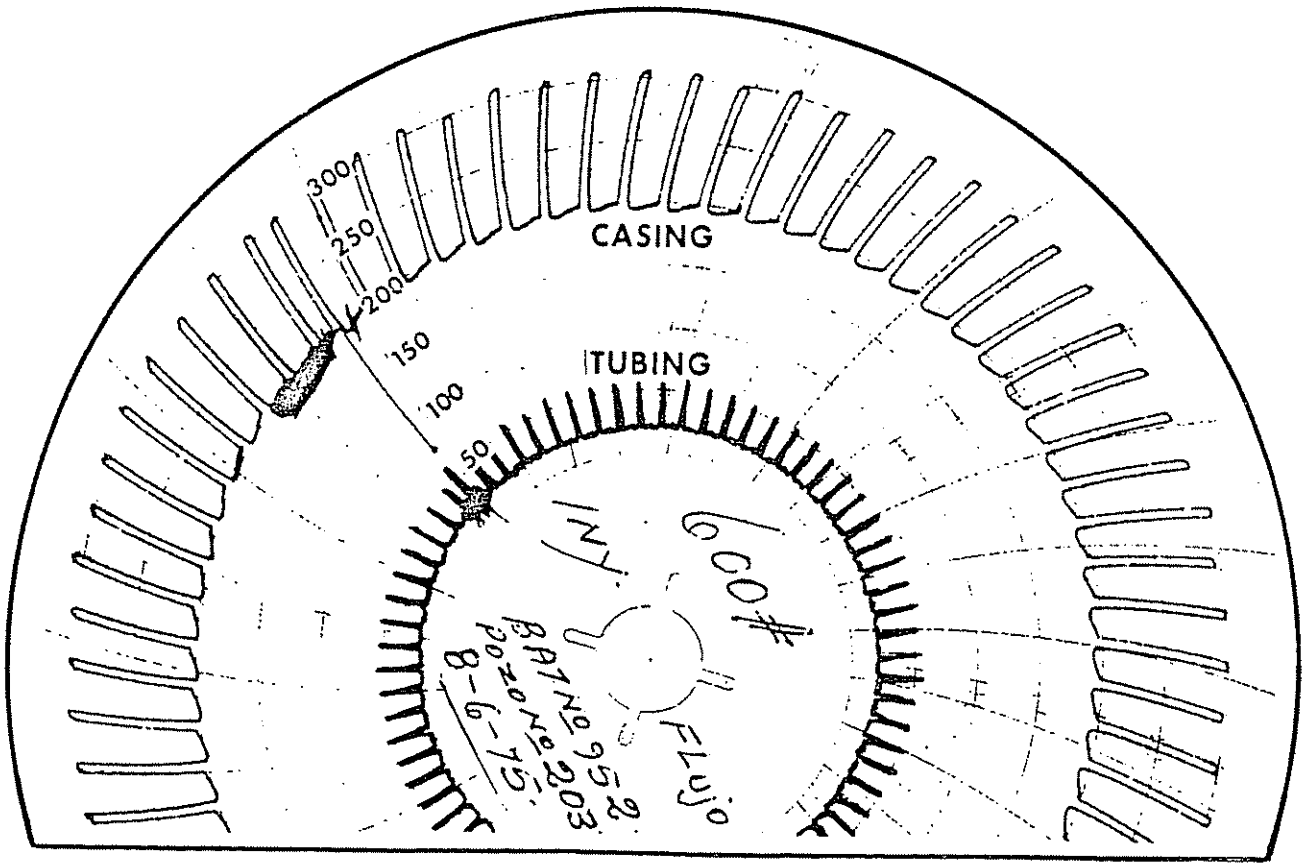


Fig. 3.621 (57) Intermittent Lift—2500 ft., 15 BiD Small Casing

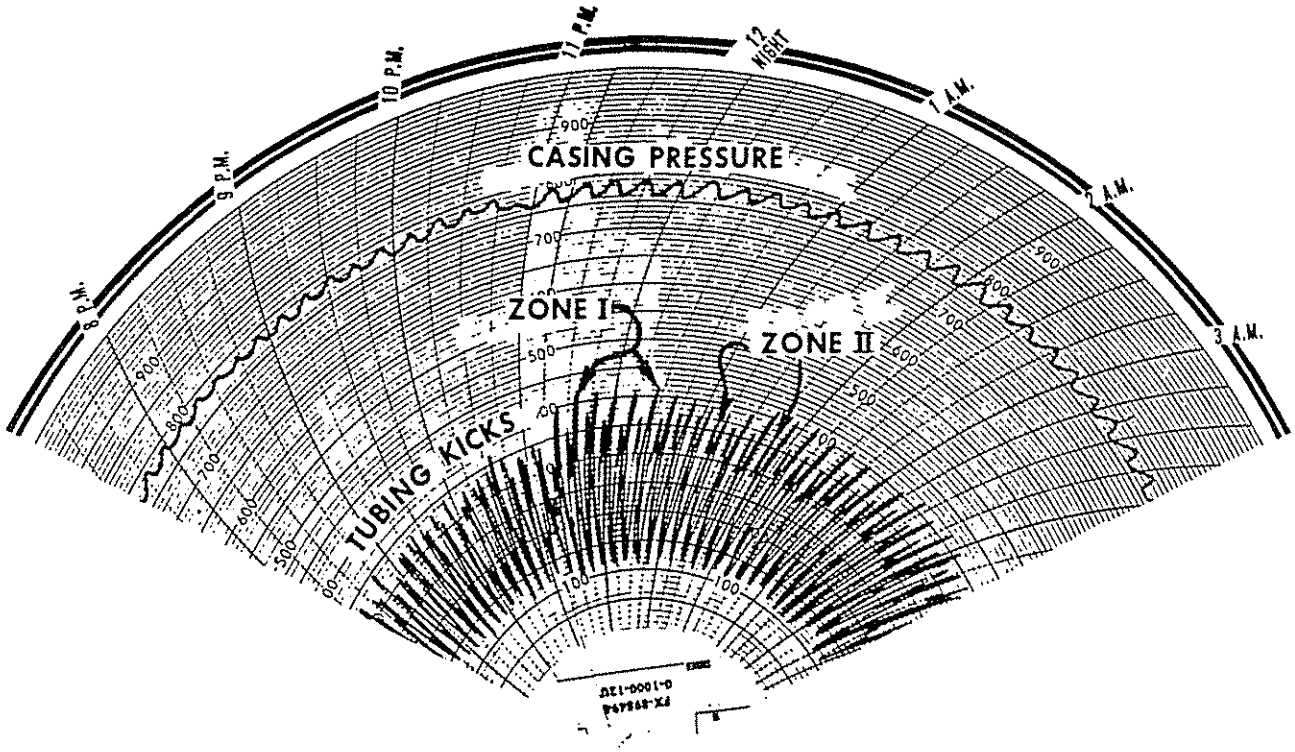


Fig. 3.621 (58) Common Annular Dual. Both sides intermittent lift. Type Valve: Both sides fluid valves. Surface Control: Variable choke.

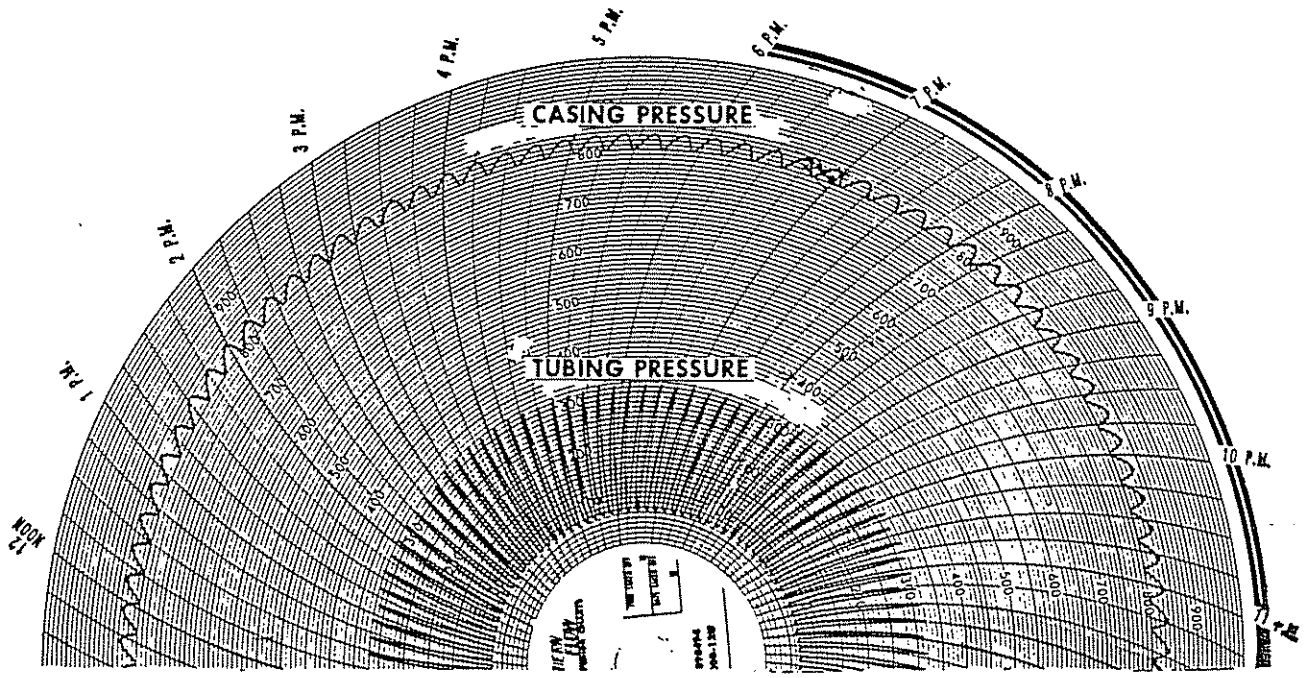


Fig. 3.621 (59)-A Two Pen Pressure Recording. Fluid Valve Intermittent Installation, Single Completion

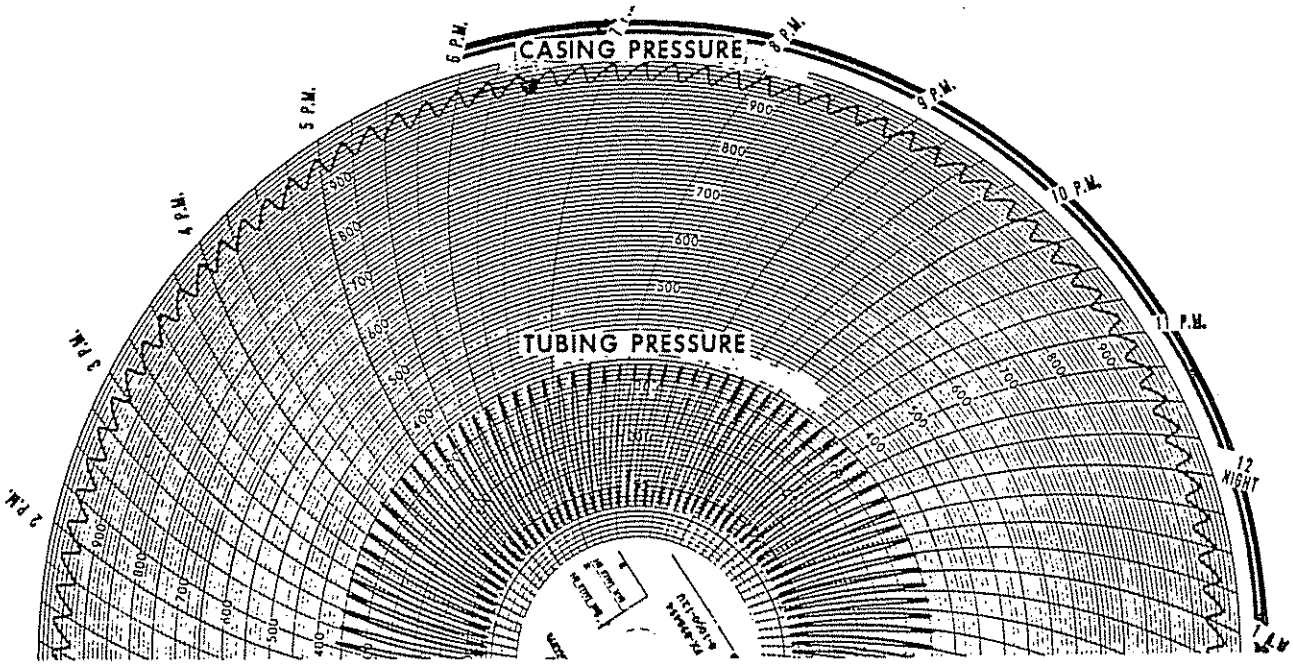


Fig. 3.621 (59)-B Two Pen Pressure Recorder. Fluid Valves-Casing Pressure Increase to 950 PSI

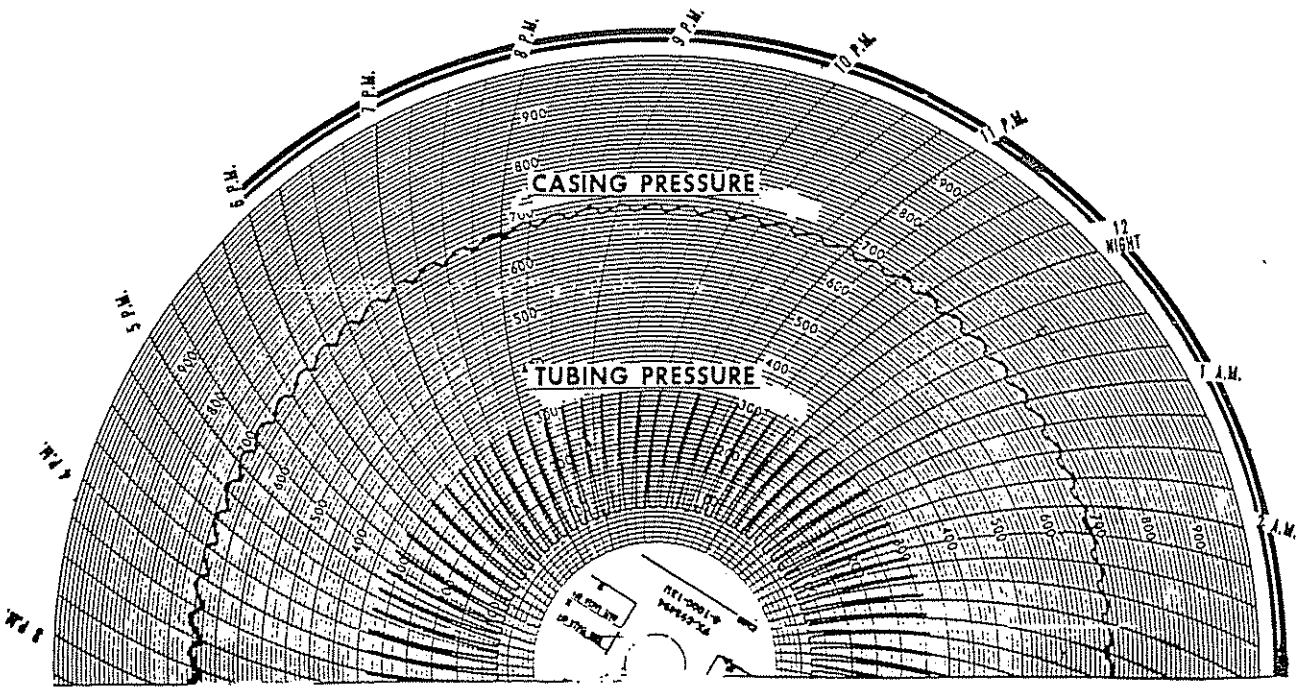


Fig. 3.621 (59)-C Two Pen Pressure Recorder. Fluid Valves-Casing Pressure Decrease to Approximately 675 PSI

flexibility and ability of fluid valves to operate under widely varying casing operating pressures. This is desirable for common annulus duals or wells where line pressure fluctuation is a problem. The fluid valves in this installation were designed to operate with 800 psi operating pressure. Figure 3.621(59)A shows the casing pressure is approximately 800 psi and the operation is satisfactory. Notice in Figure 3.621(59)B the line pressure was allowed to rise to 950 psi but the operation was essentially unaffected. The pressure subsequently dropped to 675 psi. (Figure 3.621(59)C). Although the rate fell off somewhat, the well continued to produce efficiently. This particular installation produced efficiently over an operating pressure range of 275 psi.

3.638 Surface wellhead pressure

3.6381 Introduction

The surface well head pressure or surface back pressure of a gas lift installation is extremely important in controlling total fluid production. As noted by Beadle, Harlan and Brown:

> Serious loss of production occurs when gas lift wells are produced against excessive back pressures at the surface. Surface back pressure is considered to be the pressure found inside the Christmas tree just upstream from the flowline. This excessive back pressure is generally caused by: (1) high separator pressure; (2) choke in the Christmas tree or flowline; (3) long flowline; (4) inadequate flowline size; (5) restrictions in the flowline such as paraffin, scale deposits, crimped flowline, etc.; and (6) a nonstreamlined Christmas tree incorporating a large number of sharp bends.

The high surface back pressure can generally be economically corrected. The changing of a flowline from 2" to 4" will normally pay out in a short period of time.

3.6382 Effect of back-pressure for continuous flow

In continuous flow the surface tubing pressure directly affects the flowing BHP. For high productivity wells an increase in tubing pressure of 100 psi could result in a decreased production of several hundred b/d. High surface back pressure decreases the fluid production and increases the gas requirements per barrel of lifted fluid. Refer to Figure 3.622A for a plot showing how a lowering of the tubing pressure allows a lower flowing BHP to be created and a lower injection GLR. Figure 3.622B shows how the same flowing BHP can be maintained with less gas as the surface flowing tubing pressure is lowered.

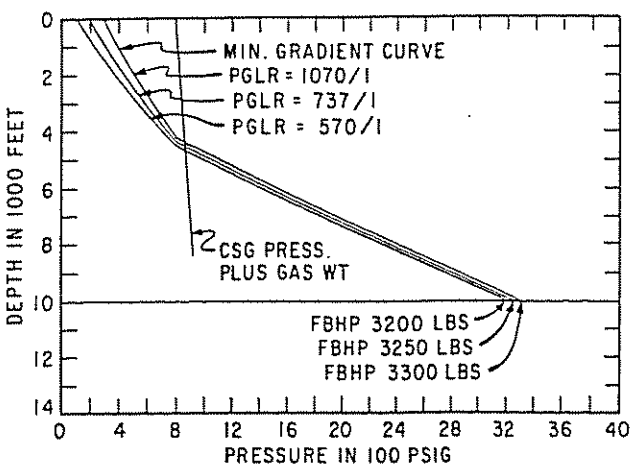


Fig. 3.622·A Effect of Surface Back Pressure on Production Rate

Beadle et al.,[7] ran a series of field tests to show the effect of increased back pressure on an actual field well. The following tabulation shows these results:

TABLE 3.61
EFFECT OF SURFACE CHOKE ON RATE

Choke size (in.)	Oil (B/D)	Liquid (B/D)	Surface back pressure (psig)	Flowing BHP (psig)
1	118	1745	175	3666
¾	74	1209	190	3678
⅝	60	852	200	3684
½	3	633	225	3682

A complete removal of all restrictions plus a 4 in. flowline installation allowed this well to make 2375 b/d of liquid (160 bo/d). The limiting factor was a required 100 psig separator pressure.

The following table was given by Beadle, Harlan and Brown showing vivid results of removing surface restrictions:

TABLE 3.62
EFFECT OF REDUCING BACK PRESSURE

Well no.	BOPD before back pressure was reduced	BOPD after back pressure was reduced	% increase
1	41	135	229
2	58	102	76
3	17	21	24
4	19	23	21
5	114	333	192
6	35	58	66
7	89	129	45
8	39	52	33
9	43	94	119
10	7	16	129
11	31	133	329
12	74	160	116
	567 Total	1256	122
		−567	Average
		689 Increase	

These observations show that production can be increased by eliminating surface back pressure and surface restrictions. The rate can be more than doubled in many instances.

The remedy for high back pressure in many instances requires the installation of larger flowlines. However, from an economic standpoint a change from a 2" to a 4" flowline can be easily justified in many cases. Figure 3.622C shows the effect of flowline diameter and gas injection rate on liquid production rate, while

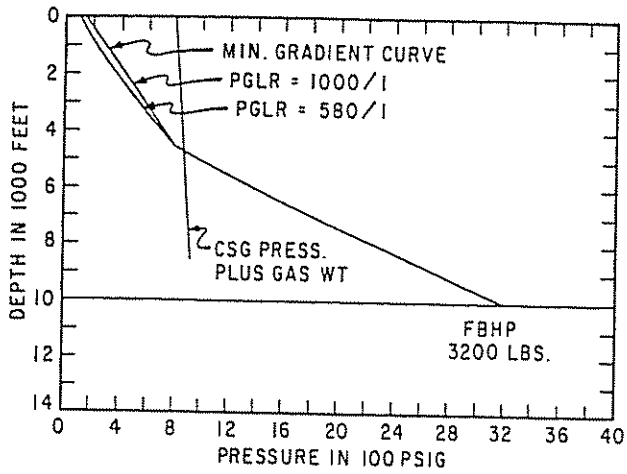


Fig. 3.622-B Effect of Surface Back Pressure on Injected Gas Volume

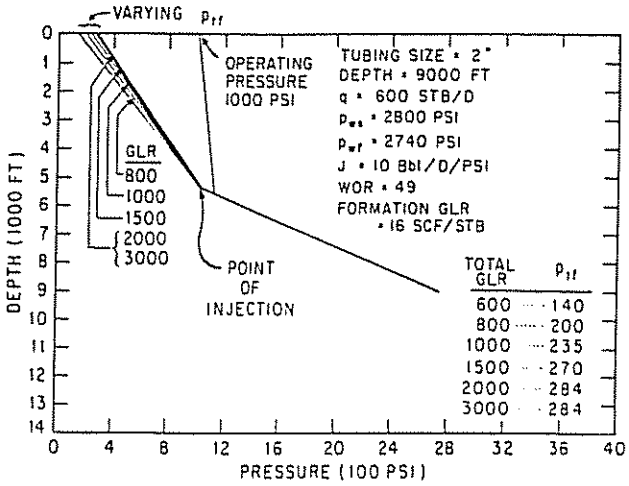


Fig. 3.622-E Effect of Wellhead Pressure on Gas Liquid Ratio

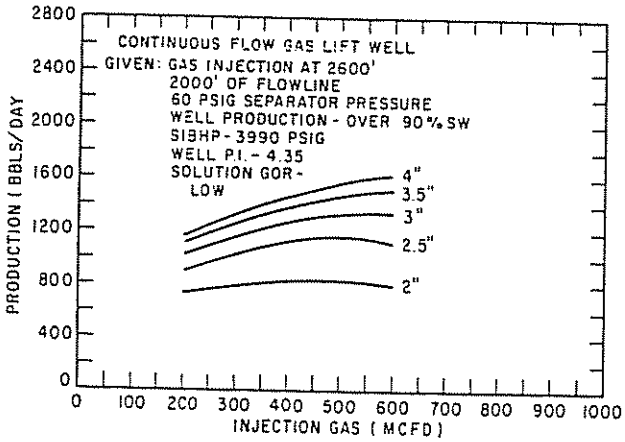


Fig. 3.622-C Effect of Flow Line Diameter and Gas Injection Rate on Liquid Production

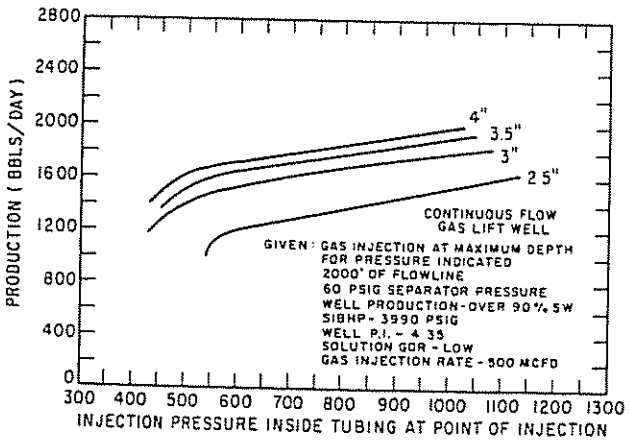


Fig. 3.622-D Effect of Flowline Diam. and Gas Injection Pressure on Liquid Production

Figure 3.622D shows the effect of flowline diameter and gas injection pressure on the liquid production rate. Figure 3.622E shows how the same rate of 600 B/D can be maintained with different values of wellhead pressure requiring different volumes of gas. The following table illustrates this effect:

TABLE 3.63
EFFECT OF WELLHEAD PRESSURE ON GAS-LIQUID RATIO

Wellhead pressure (psig)	Required total GLR (scf/B)
140	600
200	800
235	1000
270	1500
284	2000
284	3000

The lower wellhead pressures require less gas than the higher pressures.

Figure 3.622F:

Figure 3.622F shows a case on an offshore platform in the Cook Inlet of Alaska, illustrating a change in rate from 4600 B/D to 5000 B/D by lowering the wellhead pressure from 140 psi to 100 psi.

Figure 3.622G:

Figure 3.622G shows the difference in production rates by gas lift for wellhead pressures of 100 and 200 psi. This is a high productivity well. The following tabulation shows rates possible at different static pressures and for 4" tubing and 2⅞" × 7" annular flow:

TABLE 3.64
EFFECT OF WELLHEAD PRESSURE ON RATE

Flow configuration	Wellhead pressure	$\bar{P}_R$	Rate B/D
4"	200	2600	9400
4"	100	2600	10600
4"	200	1400	4200
4"	100	1400	5100
2⅞" × 7"	200	1400	7700
2⅞" × 7"	100	1400	9100

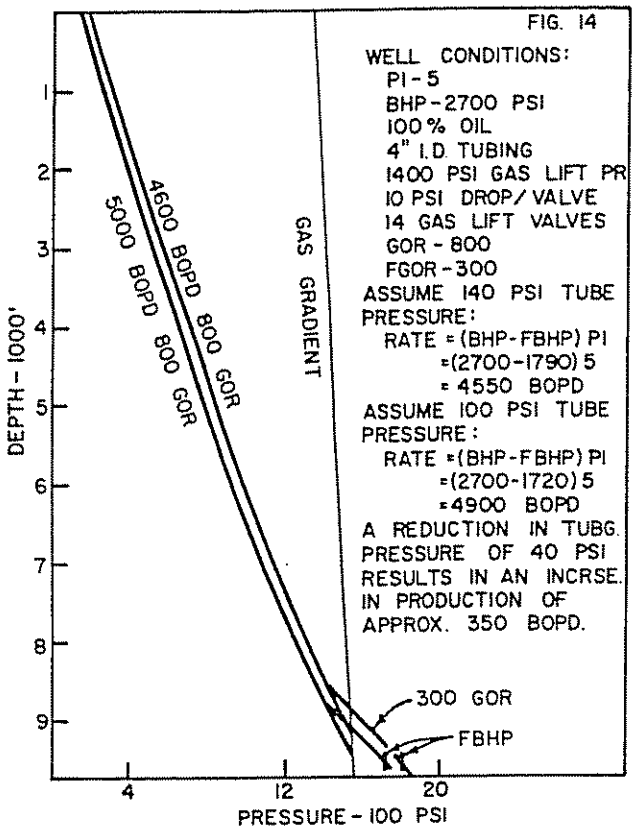


Fig. 3.622-F Effects of Tubing Pressure on Producing Rates

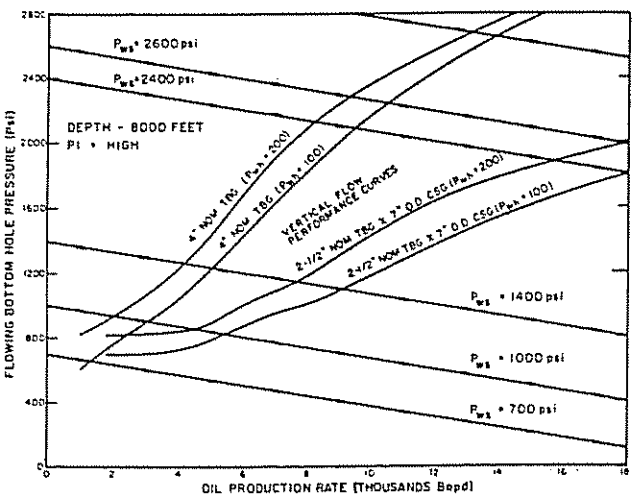


Fig. 3.622-G Effect of Wellhead Pressure

As noted, a wellhead pressure of 100 psi will cause the well to produce at a much higher rate compared to a wellhead pressure of 200 psi.

Figures 3.622H(1) and 3.622H(2):

Figure 3.622H(2) shows the changes that were made in an offshore platform production system similar to Figure 3.622H(1) The 4" header was changed to 12",

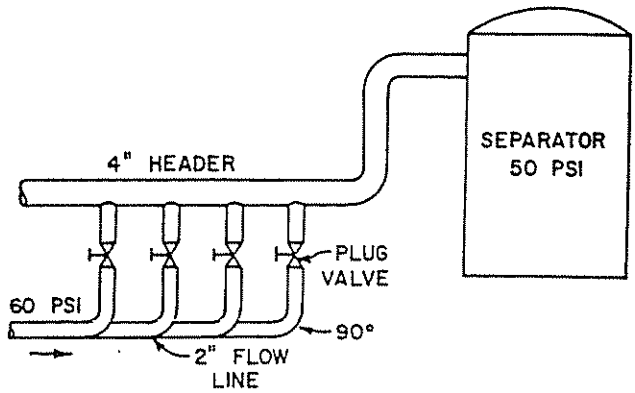


Fig. 3.622-H (1) Typical Gas Lift Installation, Header & Separator System

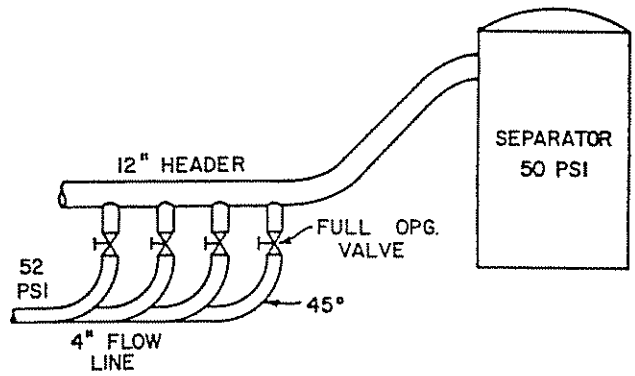


Fig. 3.622-H (2) Gas Lift Installation after Modifications, Header & Separator System

the 2" flowlines to 4", the plug valves for full opening valves, and the 90° turns were eliminated. These changes only made an 8 psi change in pressure at the wellhead but paid for themselves immediately in increased production from high PI wells.

3.6383 Effect of back-pressure for intermittent flow

Surface back pressure and surface restrictions can also materially affect the production rate from an intermittent well. For example, Beadle, Harlan, and Brown showed that the percentage recovery of an initial slug could vary from 35 - 73% in producing against a ⅜" choke at the surface as compared to a 1½" surface choke. Figure 3.622(I) shows a comparison of slugs being produced against a ½" and a 1½" surface choke. Figure 3.622(J) shows a 24-minute recording for the same slug produced against ⅜", ½", ¾" and 1½" surface chokes.

Evidentally, the 1½" surface choke allows a greater percentage of the slug to be recovered. Figures 3.622K(1) and 3.622K(2) show pressure recordings down the hole for typical intermittent slugs against differing surface choke sizes. Note the change in percent recovery from 35% for a ⅜" choke to 66% for a 1½" choke.

The following conclusions and recommendations were given:[7]

In considering the size of surface choke to install for intermittent lift, the important factor to be con-

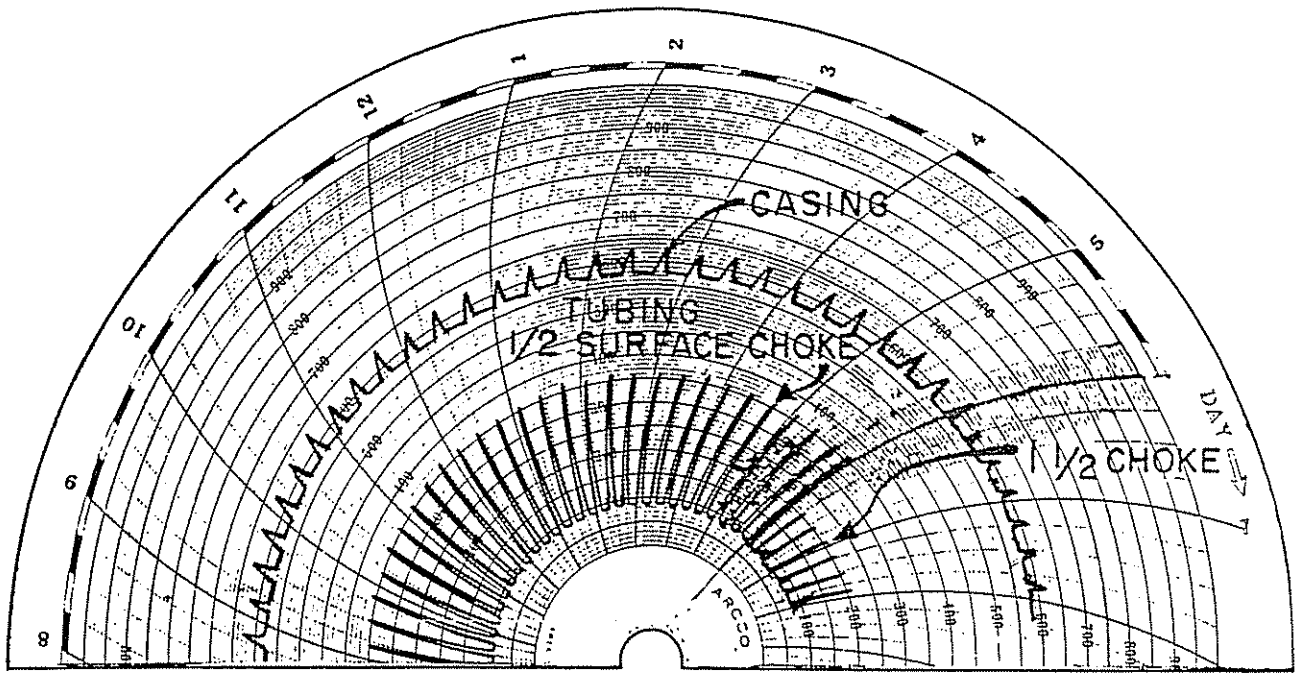


Fig. 3.622-I Typical Recording of Casing and Tubing Pressure—$^{7}/_{16}$" Diameter Port from 4072 ft., (½" and 1½" Surface Chokes)

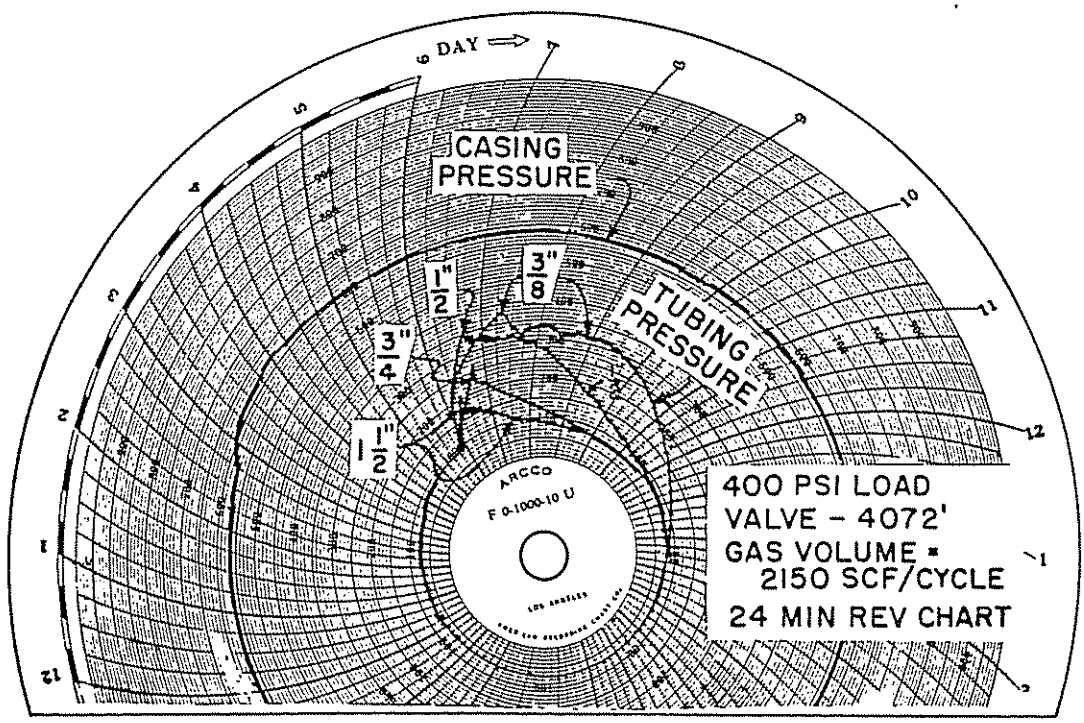


Fig. 3.622-J Surface Tubing Recording for Varying Surface Chokes

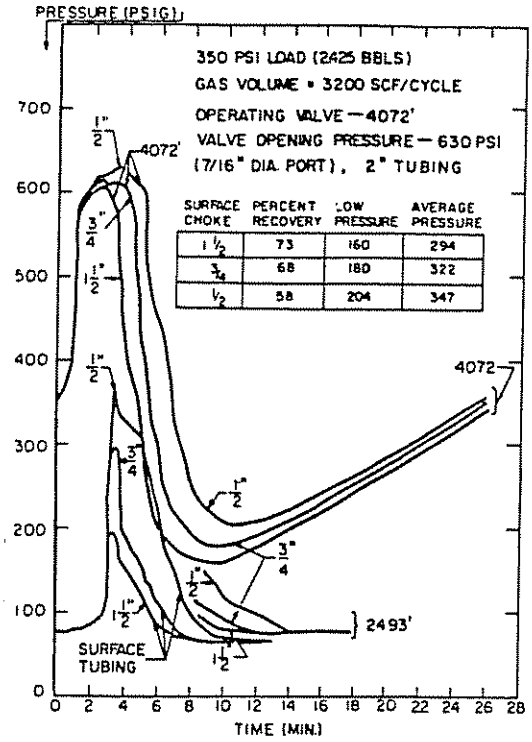


Fig. 3.622-K (1) Pressure Recordings for Varying Surface Choke Sizes

sidered is the maximum sized surface choke that will allow protection to all surface equipment.

In many instances the surface equipment is not being utilized properly so that a minimum restriction can be taken at the Christmas tree. As soon as it has been decided that a well is to go on intermittent gas lift, the following factors should be immediately investigated:

(1) The size of seat in the liquid dump at the separator should be checked and a maximum permissible size installed;

(2) The size of orifice in the gas measurement meter run should be checked and a size installed that is capable of passing the peak tail gas behind the liquid slug. In many instances the entire meter run may have to be changed to a larger size in order to ensure that the gas volumes can be handled. It may be necessary to install a by-pass as a safety device to prevent separator damage.

There are various ways to improve intermittent installations that must use a surface choke to protect surface equipment. If a well must be choked to protect the separator, it may be beneficial to place the choke at the separator instead of at the well head. If the flowline is fairly long, this will allow most of the slug to leave the vertical tubing string and enter the horizontal flowline before encountering the surface choke. This will prevent the additional fallback of liquids that will occur if the choke is at the top of the vertical tubing string. If the installation is equipped with a short flowline before reaching the separator, a container similar to a separator but with no gas outlet can be installed in the flowline to hold the liquids and allow their transport to the separator at a safe rate.

Once an intermittent well has reached a low bottom hole pressure, it cannot tolerate a high surface back pressure without serious loss in production.

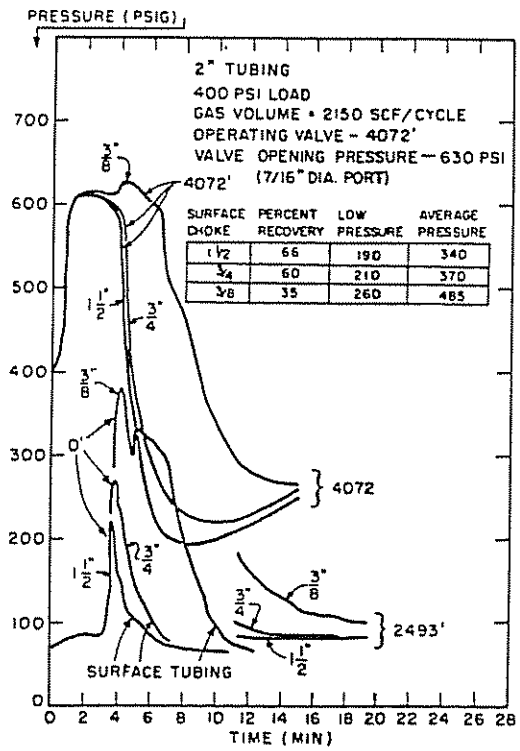


Fig. 3.622-K (2) Typical Pressure Recordings for Varying Surface Choke Sizes

Figure 3.622L:

Figure 3.622L shows how a surface restriction can appear as a good liquid slug. The ⅜" surface choke causes the tubing kick to reach a pressure of 500 psi, indicating a good recovery which, in reality, is very poor recovery.

Figures 3.622M(1) and 3.622M(2) show how the surface choke size affects all factors such as GLR and percent recovery in the actual production of an intermittent slug from 4000 ft.

Figure 3.622M(3) shows an additional plot that illustrates a minimum surface choke size of 1" diam. In all cases a rather sharp change in slope of the line occurs for choke sizes of ¾" diameter and less. Any choke size less than 1" diameter can be expected to reduce the total fluid recovery from the well. This surface restriction immediately reflects upon the average flowing BHP, thereby causing a decrease in production. Table 3.65 shows the behavior of surface chokes ranging in size from 1½"-⅜".

3.639 Injection gas pressure

One of the first checks on any installation with unsatisfactory performance is on the injection gas pressure utilized at the well head. A low operating pressure for continuous flow wells may explain low production

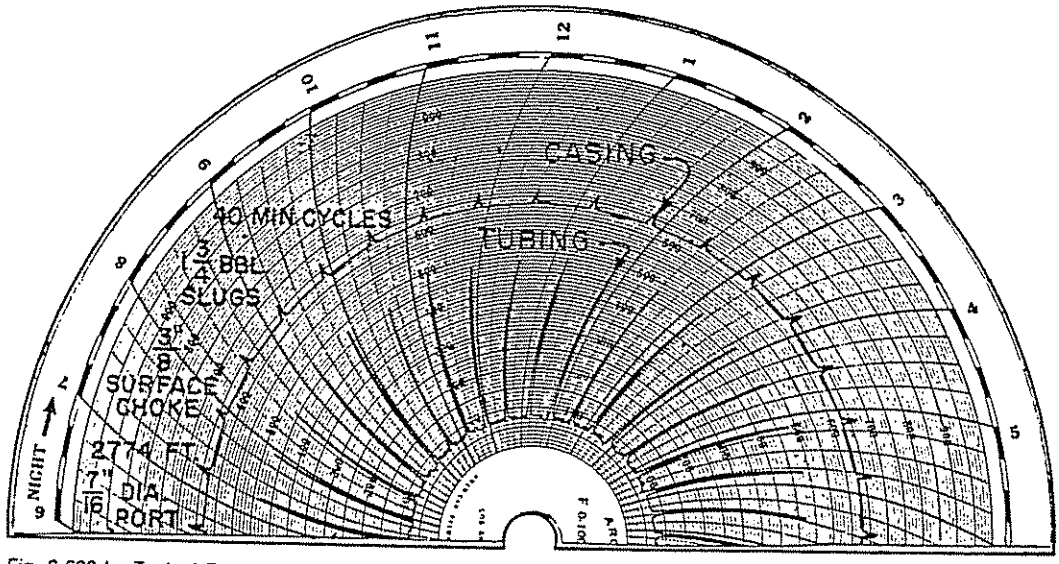


Fig. 3.622-L Typical Recording of Casing and Tubing Pressure (7/16-in. diam. port from 2774 ft. depth; 3/8-in. surface choke)

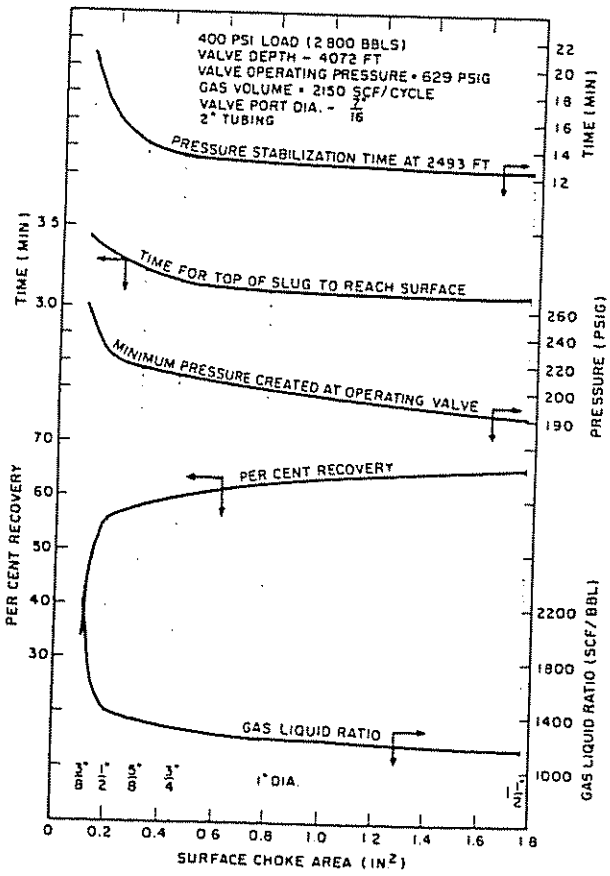


Fig. 3.622-M (1) Surface Choke Evaluation for 400 PSI Constant Load and Constant Gas Volume of 2150 SCF/Cycle

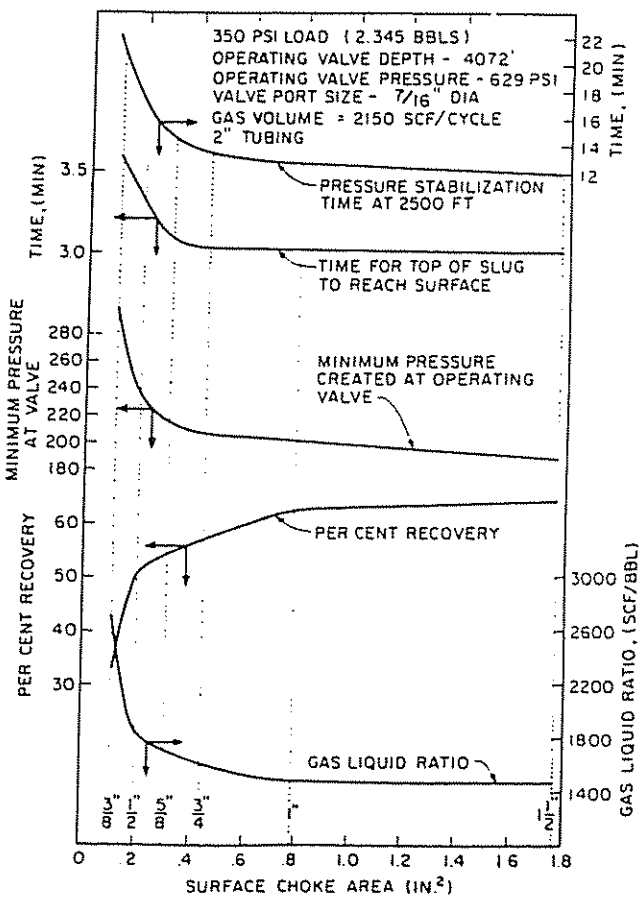


Fig. 3.622-M (2) Surface Choke Evaluation for 350 PSI Load and Constant Gas Volume of 2150 SCF/Cycle

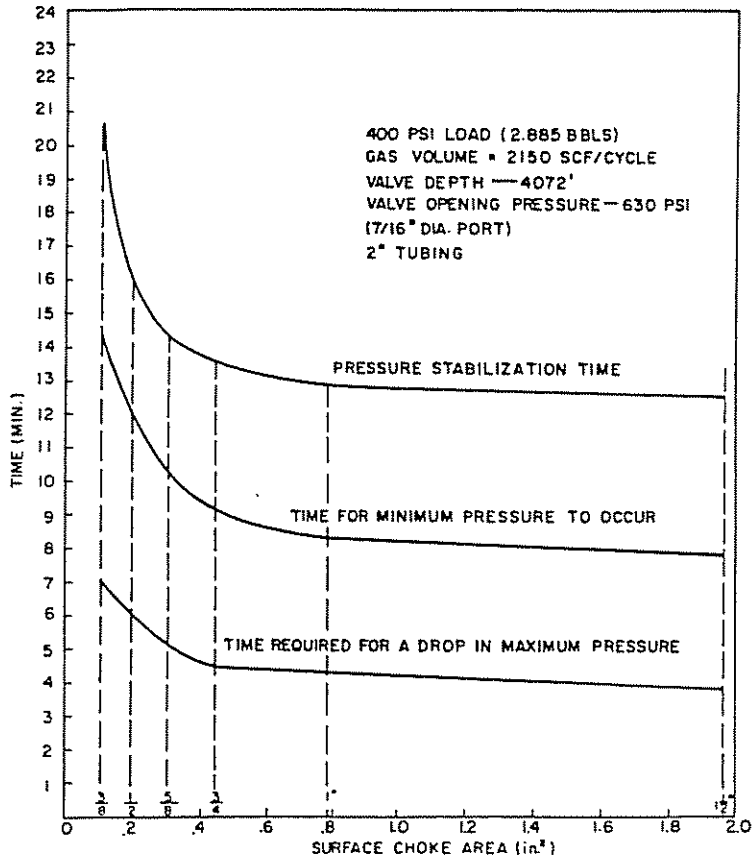


Fig. 3.622-M (3) Time Evaluation for Intermittent Slug

TABLE 3.65
SURFACE CHOKE TABULATION

Surface choke diam (in.)	Recovery (%)	GLR (scf/B)	Minimum pressure at operating valve (psig)	Time for slug to reach surface (min)	Pressure stabilization time at 2493 ft (min)
1½	66	1160	190	3.09	12½
1	63	1220	200	3.11	13
¾	60	1300	210	3.14	14
⅝	58	1350	215	3.18	14
½	56	1400	224	3.22	16
⅜	35	2200	260	3.40	21

rates. The same can be true for deep intermittent flow wells. A low operating pressure from a well 8000 ft or deeper can explain low recovery per cycle. A dropping-off of casing pressure for intermittent wells indicates leaking tubing or valves. Some of these indications are shown on the recorder charts.

In many cases the injection gas pressure available limits the production rate. Provisions must be made to increase the pressure in order to increase the production.

Figure 3.623A

Figure 3.623A shows the effect of increasing the injection gas pressure on production rate. This is a typical high productivity well (PI=6). The rate changes from 4700 B/D for 1000 psi to 5800 B/D for 1400 psi operating pressure.

Figures 3.623(B) and 3.623(C):

Figure 3.623(B) shows an original installation of compressors on an offshore platform furnishing 900 psi to the wells. Figure 3.623(C) shows the addition of a compressor which gives 1300 psi available at each well.

3.6310 Injection gas volumes

Every gas lift installation should be equipped with a means of measuring the input or injection GLR. This need not be a permanent meter installation. A meter

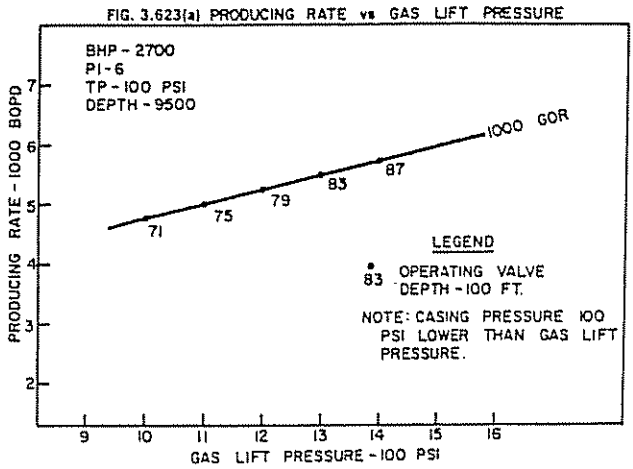


Fig. 3.623-A Producing Rate vs. Gas Lift Pressure

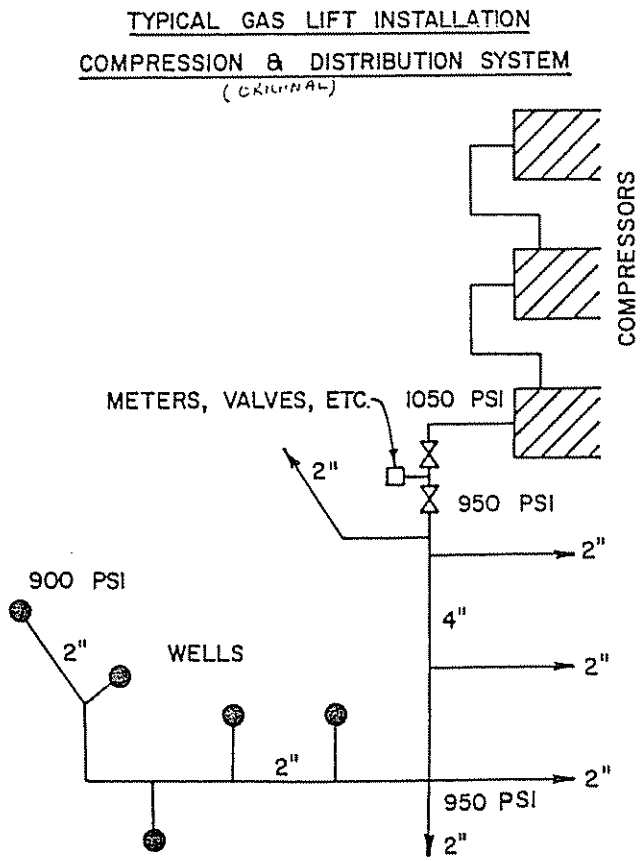


Fig. 3.623-B Typical Gas Lift Installation, Compression & Distribution System

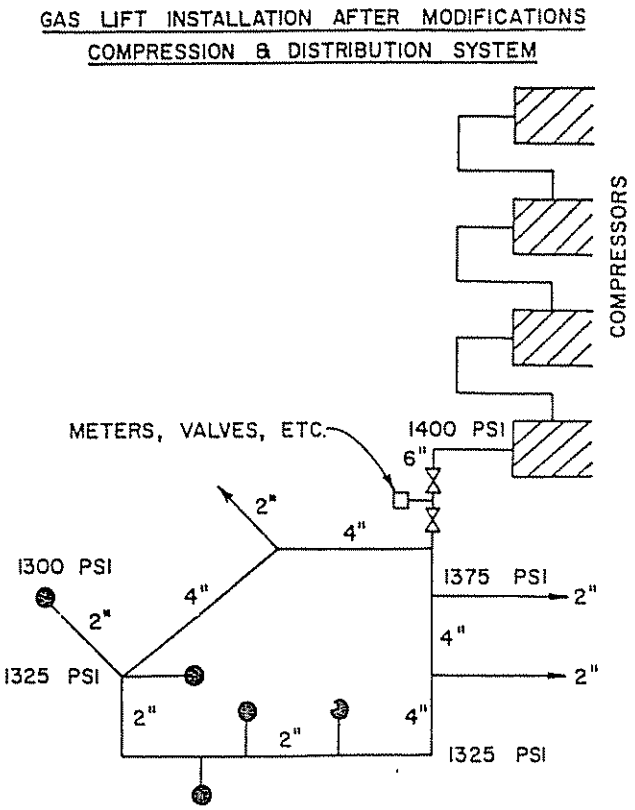


Fig. 3.623-C Gas Lift Installation after Modifications, Compression & Distribution System

run may be installed on the straight section of the injection line and a portable meter used for measuring the gas. For several wells in any one field, the portable meter will provide an economical savings. The following comments were offered by Brown and Canalizo:[5]

Normally, a continuous flow well will offer no problems in gas measurement. However, a well on intermittent flow will make recordings on a gas meter chart that are difficult to interpret. This is particularly true for a 24-hour or 7-day revolution clocks. Therefore, provisions should be made to obtain combination clocks on gas meters than can be easily converted to 24-minute clocks (one complete revolution in 24 minutes).

The gas injected per cycle should be recorded on this 24-minute chart; in turn, enough separate "kicks" on the chart should be obtained to secure an average reading. The gas per cycle can then be calculated. The total number of kicks per day may be counted from the 24-hour chart or obtained from the injection clock settings. By multiplying the gas used per cycle by the number of cycles per day a very close approximation can be obtained for the injection gas-liquid ratios. Refer to Figure 3.624 for typical 24-hour clock recordings as compared to a 24-minute revolution clock.

Computer aided gas measurement devices should be used for present-day measurement. One of the best checks for intermittent gas volumes is to record the change in casing pressure between the opening and closing of the gas lift valve. By knowing the volume of the annular space this change in pressure can be converted to scf used per cycle. Gas volumes used should be obtained periodically so that an increase in gas consumption can be immediately detected. There can be any number of reasons why gas consumption would increase. For example, a continuous flow well on casing control (constant gas pressure maintained at surface) might lower itself down to the next lower operating valve. The casing pressure would then attempt to adjust itself to the lower valve setting, but

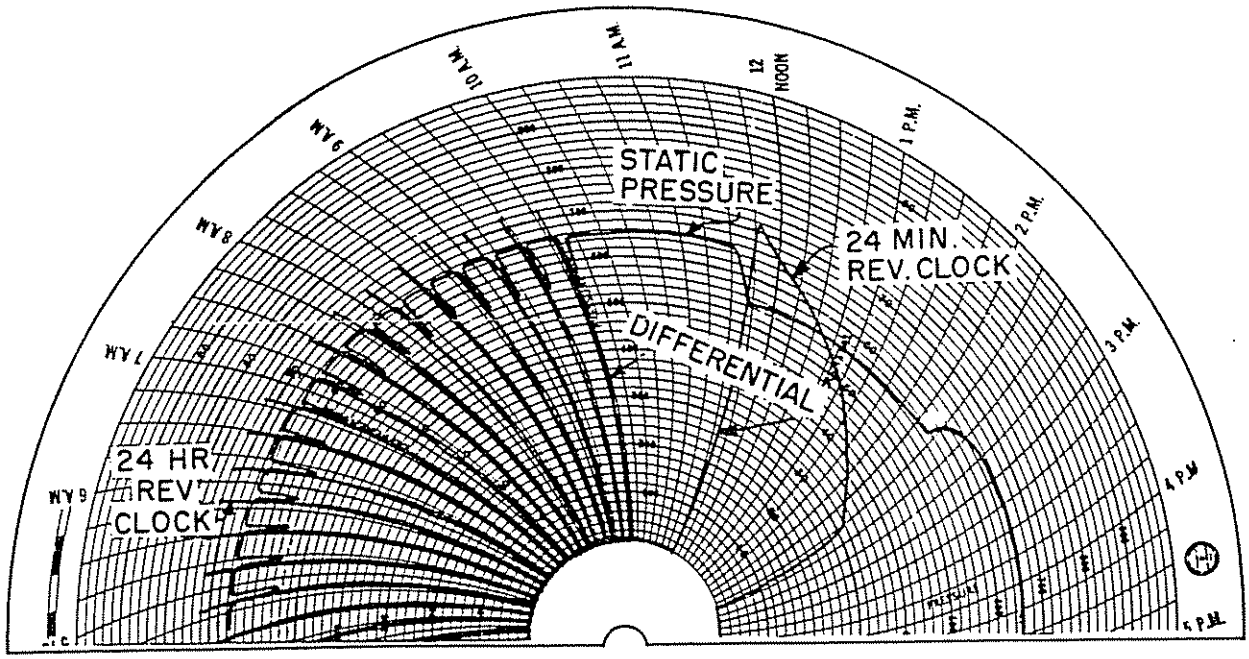


Fig. 3.624 Comparison of Gas Measurement—24-hour revolution clock and 24-minute revolution clock

the regulator at the surface would attempt to maintain a constant higher casing pressure. This would result in high gas consumption with the operation of two or more gas lift valves.

Other troubles would be a leak in the tubing, leaking packer, leak in the Christmas tree pack-off, etc.

The frequency and length of gas injection periods may be a source of trouble for intermittent wells. This problem can generally be found by analyzing two-pen recordings of the casing and tubing pessure. If this information is not available, trial and error settings will be necessary to obtain the correct cycle frequency as well as the correct volume of gas to be injected per cycle. After each adjustment, the well should stabilize for at least eight hours and possibly 24 hours before making another change.

3.6311 Total output gas volumes

This would merely be a supplement to the preceding topic, "Injection Gas Volumes," and, of course, the total output gas less the input gas provides a measure of the formation GOR. Where several wells are coming into one battery, one meter run is usually set up to measure all gas from the separator. However, for gas lift operations a test separator and a test meter run should always be set up so individual wells can be tested. The total gas output is then the sum of injection gas plus solution gas. If total gas output minus solution gas differs appreciably from measured injection gas, another test should be run. If the difference still exists, the meters should be checked.

3.6312 Total fluid recovery

One of the quickest methods for determining trouble is a loss in oil production. Keeping daily production records is important. If a loss in production occurs, several things should be checked:

(1) Check for correct gas injection volumes.
(2) Check for excessive back pressure.
(3) Finally, a pressure or temp survey may be necessary to locate the trouble.

3.6313 Temperature of the flowline and Christmas tree

Many wells have been pulled for trouble down the hole when a simple check of the Christmas tree tubing pack-off would have solved the problem. If a casing pressure decline is detected, the first thing to do is note whether or not the Christmas tree is cold and sweating, especially near the tubing or pack-off. There may be a leak in the tree. The coldness, of course, is caused by the expansion of the high-pressure gas.

In the absence of leakage, a cold tree may also be caused by a severe pressure drop of the gas coming up the tubing from the first valve because of a plug in the tubing below that point. It could also be caused by a plugged tube lower down or by a severe paraffin deposit in the tubing, but the effect might be less obvious. If a tubing gage run on a wire line finds nothing, a pressure survey should be run.

3.6314 Miscellaneous

In the absence of any surveying equipment, a few things can be done at the well head in an attempt to analyze a troublesome installation. If the casing pressure is declining, indicating some type of leakage, the following procedure may be followed to distinguish between a valve failure and a hole in the tubing:

(1) Pressure-up both the tubing and casing to the same pressure.
(2) Bleed off the casing pressure, which in turn should close the reverse checks on the gas lift

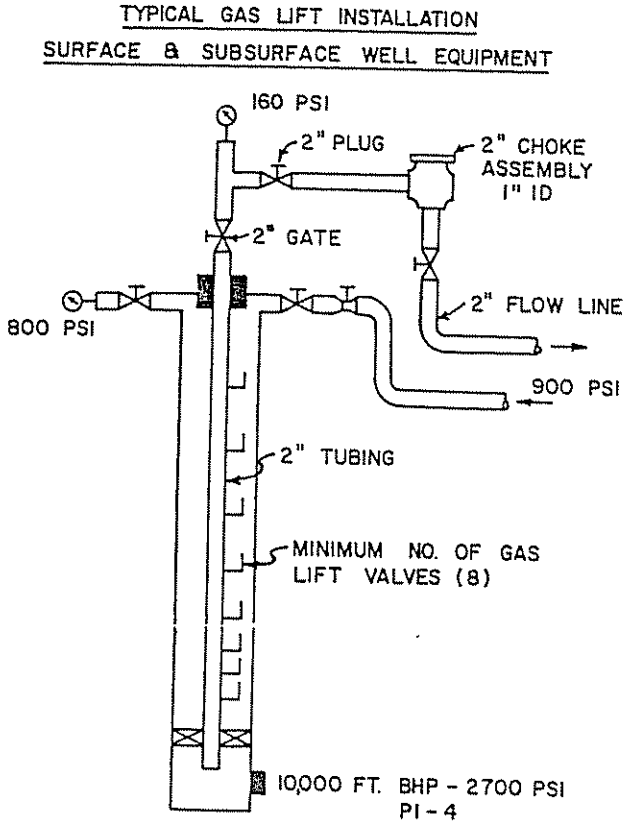


Fig. 3.625-A Typical Gas Lift Installation, Surface & Subsurface Well Equipment

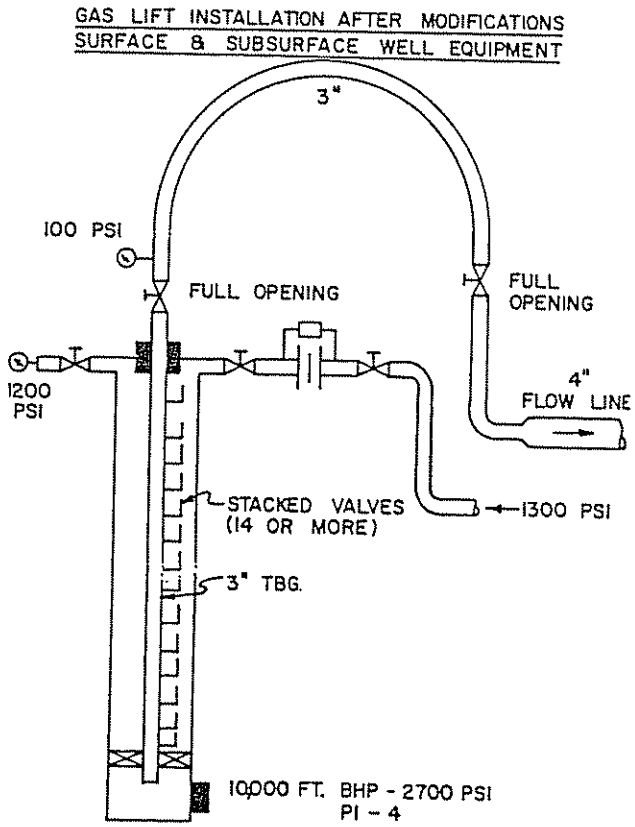


Fig. 3.625-B Gas Lift Installation after Modifications, Surface & Subsurface Well Equipment

valves, assuring that no gas will pass by the valves.

(3) If the tubing pressure remains constant, this indicates a bad valve.

(4) If the tubing pressure declines as the casing pressure is bled off, a hole in the tubing is indicated; it is improbable that a bad valve and a bad reverse check would occur together.

Another trouble that may sometimes be remedied by manipulation is where a gas lift valve is plugged. The well should be rocked. The rocking of a well simply means the successive bleeding off of first the casing pressure to a point below the tubing pressure then bleeding the tubing pressure to a point below the casing pressure. The tubing and casing pressure should at first be equalized. This simple procedure is often successful in unplugging a valve, eliminating pulling the valve or possibly the tubing.

3.6315 Field case of improved operations

Figure 3.625A shows a typical field case that was not producing the desired production rate.

Figure 3.625(B) shows the changes made to increase the production rate. The following changes were made:

(1) Changed tubing from 2" to 3"

(2) Increased the number of valves from 8 to 14

(3) Increased the operating pressure from 800 psi to 1200 psi

(4) Changed the flowline from 2" to 4"

(5) Removed all restrictions at the surface and streamlined the Christmas tree and related surface connections

3.6316 Summary

In summary, the field man must do everything possible to keep a gas lift well producing. Some things that he must check right away are the behavior of the tubing and casing pressures. This cannot be overstressed. Such things as high back pressure, freezing gas lines, tubing leaks, or gas lift valve leaks can be found. Once the field man has done everything that he can to get the well back on production, he may then want to call an engineer who specializes in the area of gas lift. The engineer may be able to help, but the field operating people are the ones who must operate the wells and recognize and report "trouble signs" from which consultation may benefit.

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Appendix 3

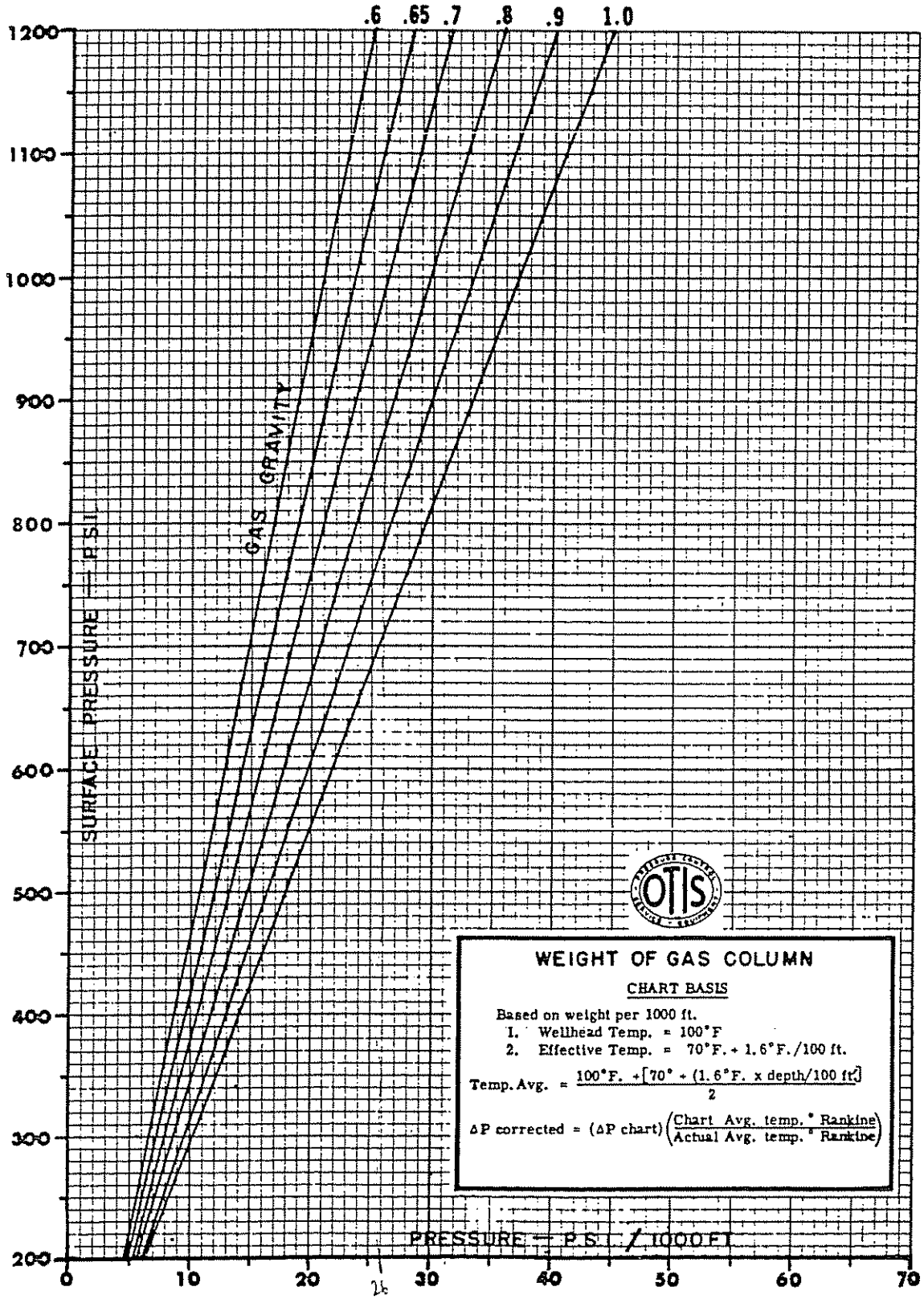


Fig. 3A-1 Weight of Gas Column (Courtesy Otis Engineering Corp.)

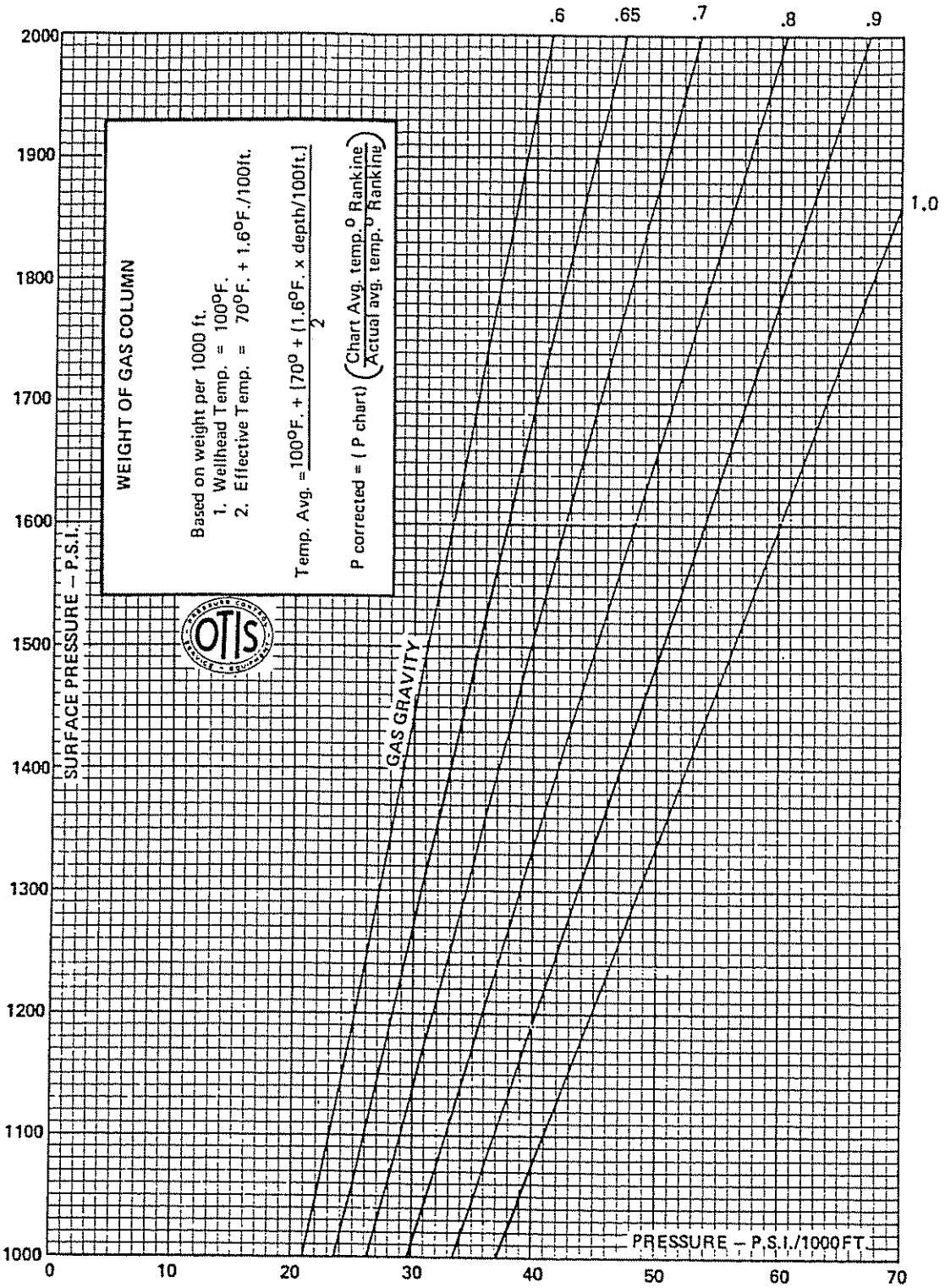


Fig. 3A-2 Weight of Gas Column (Courtesy Otis Engineering Corp.)

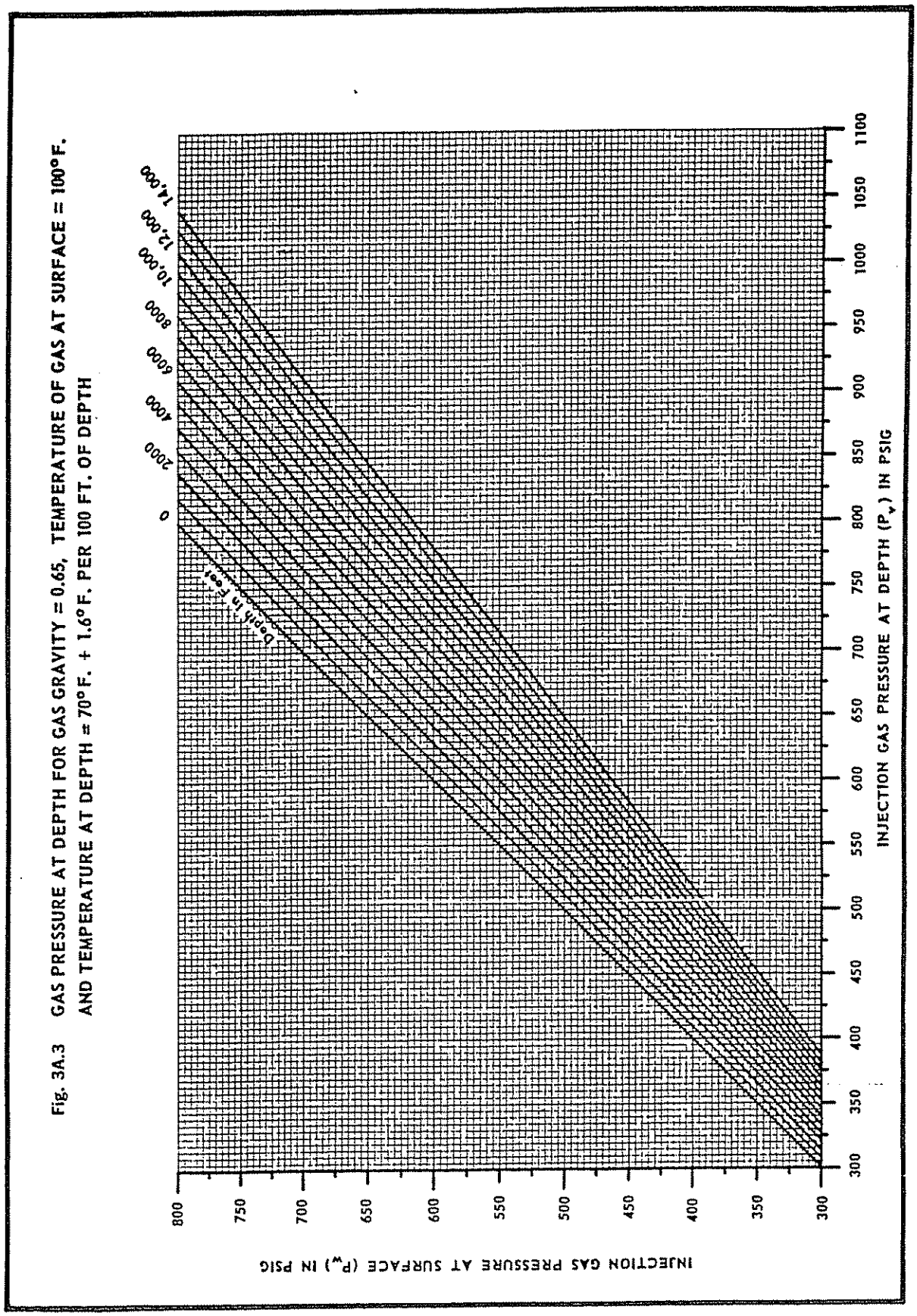


Fig. 3A.3 GAS PRESSURE AT DEPTH FOR GAS GRAVITY = 0.65, TEMPERATURE OF GAS AT SURFACE = 100°F. AND TEMPERATURE AT DEPTH = 70°F. + 1.6°F. PER 100 FT. OF DEPTH

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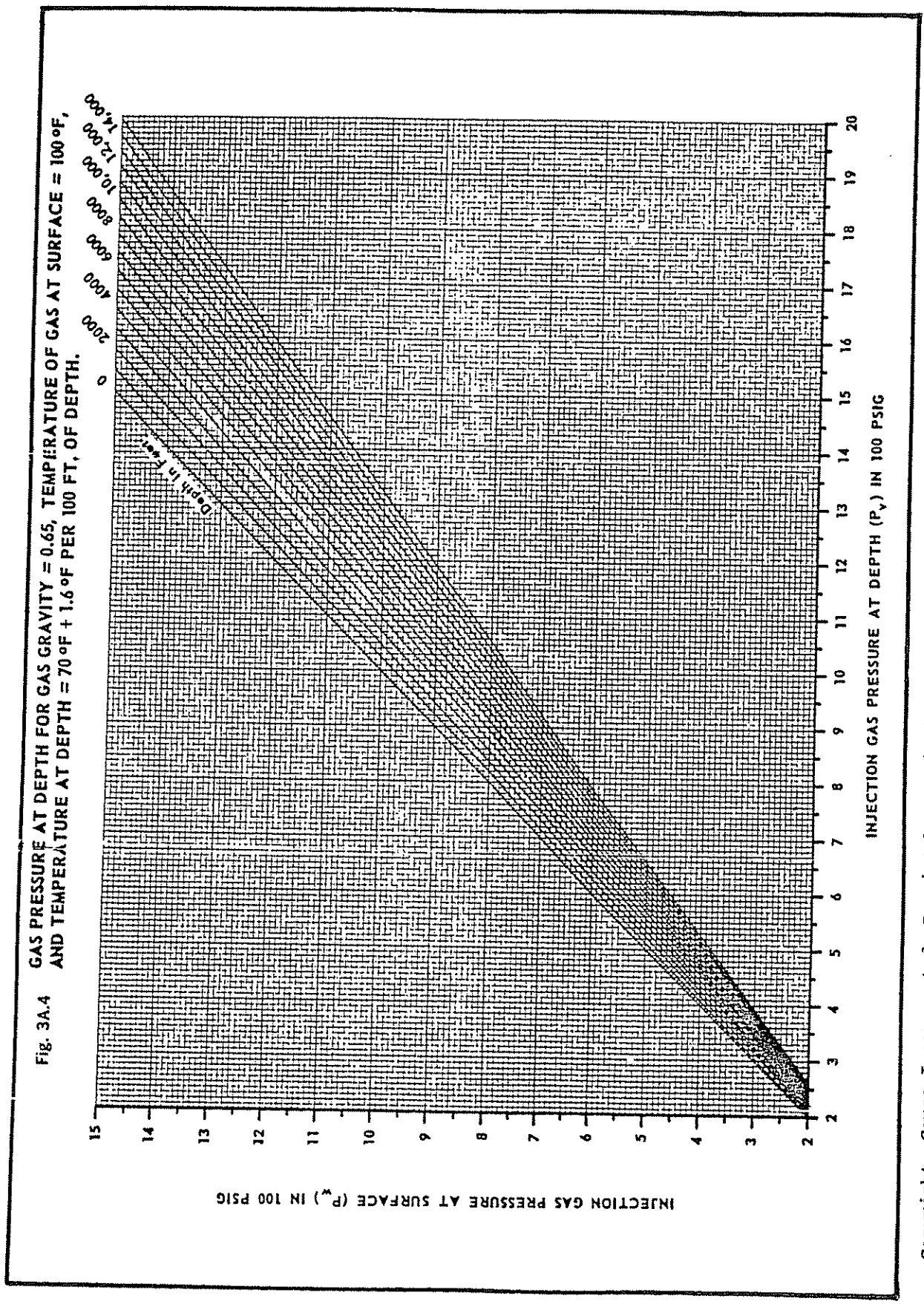
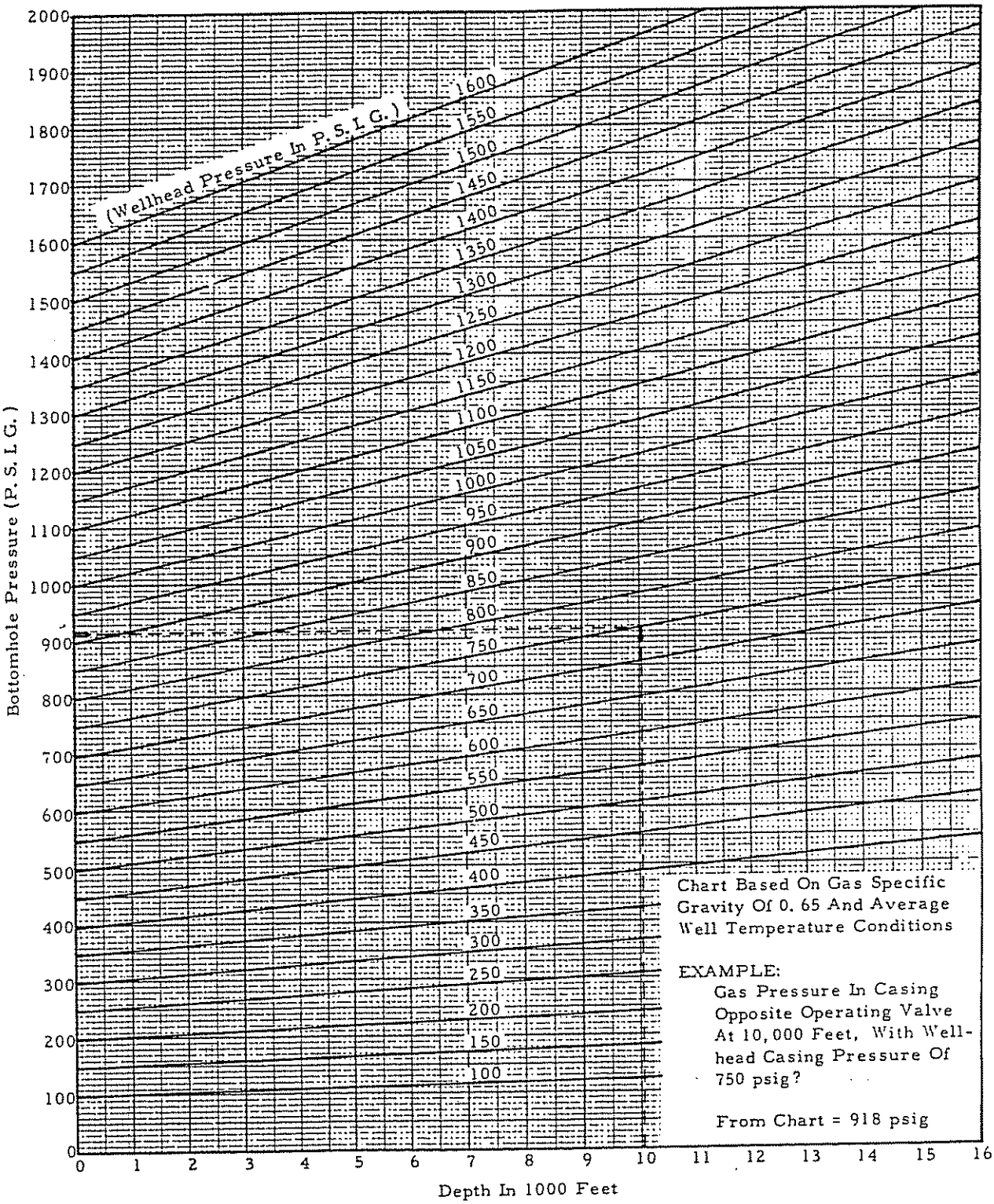
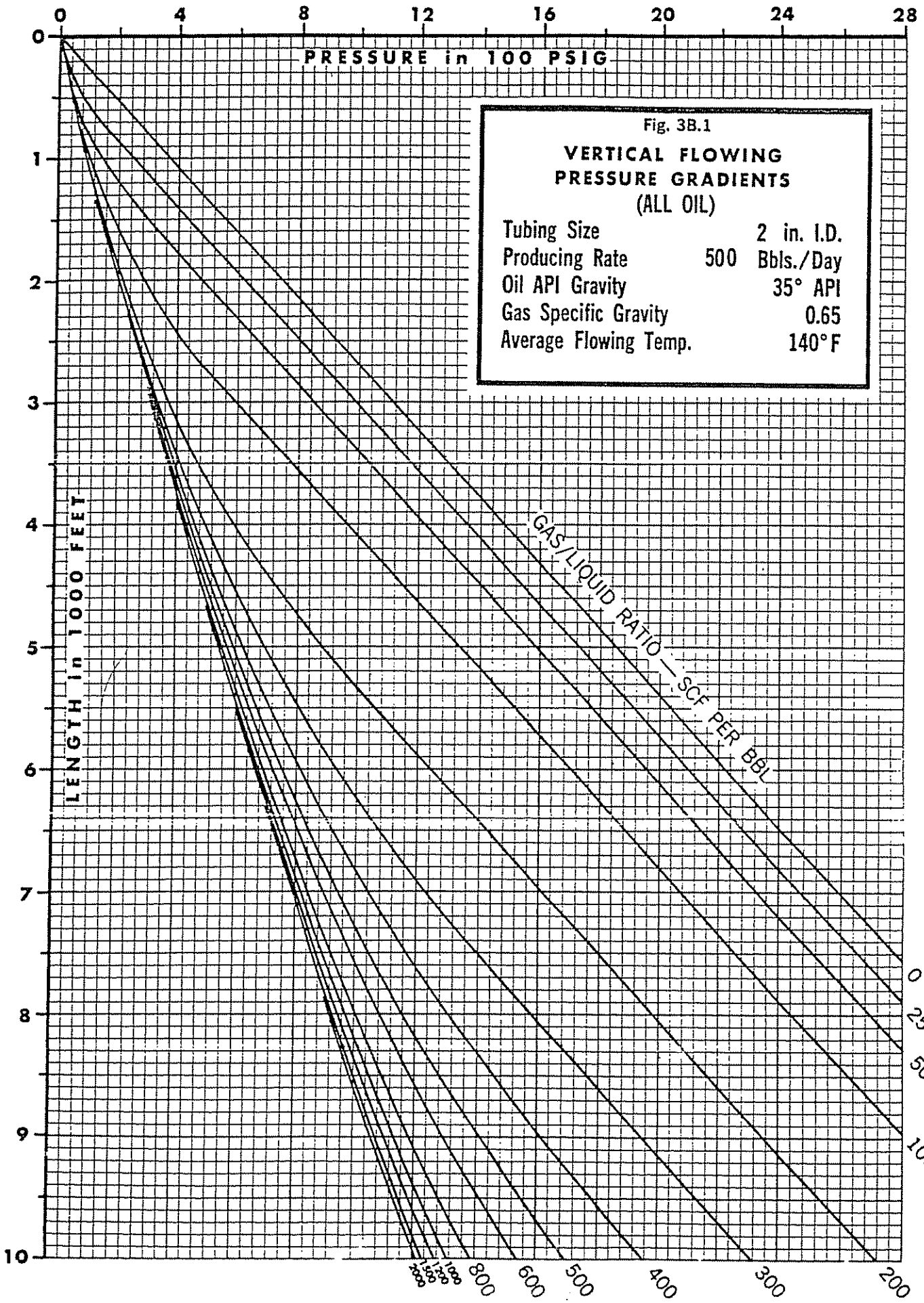


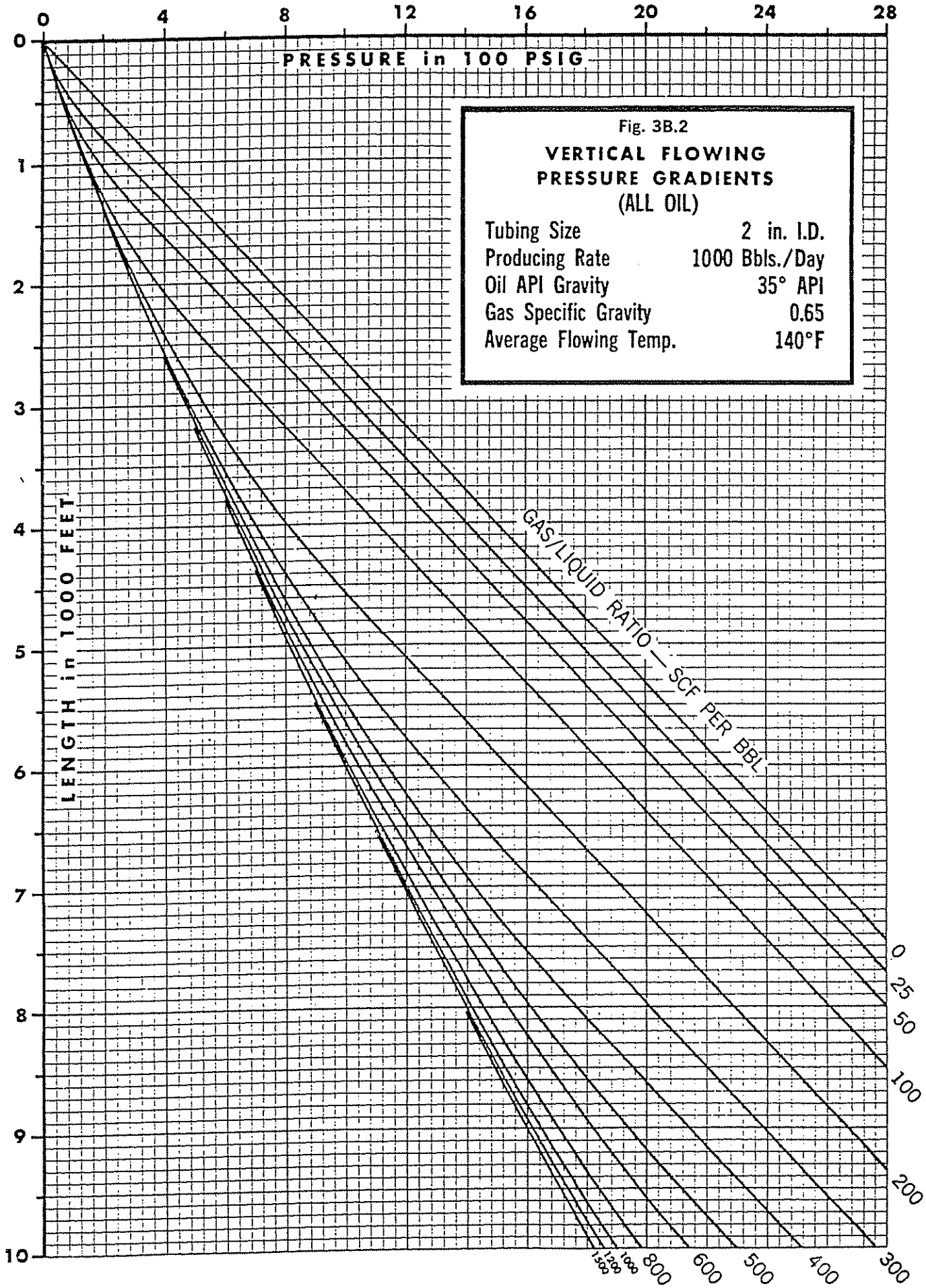
Fig. 3A.4

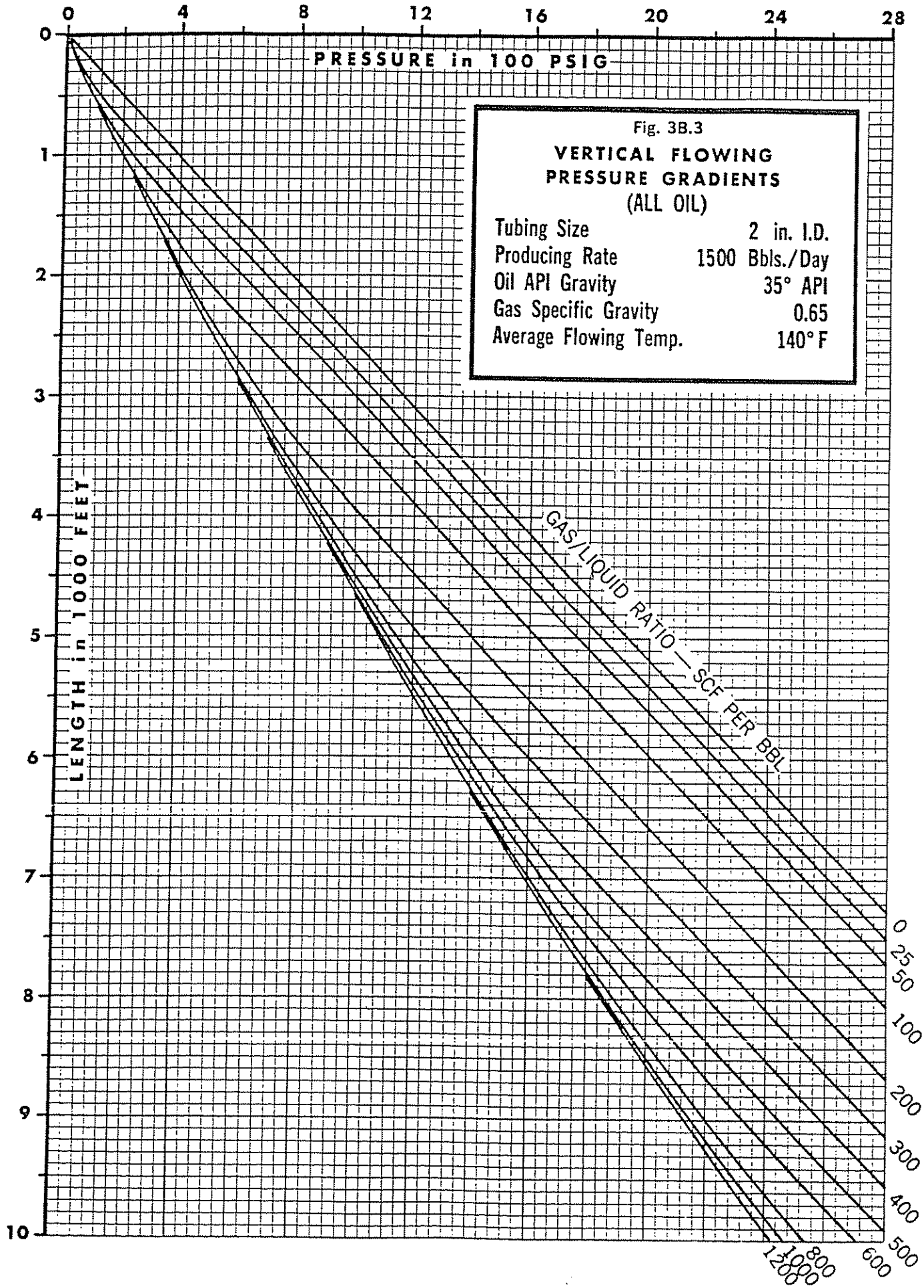
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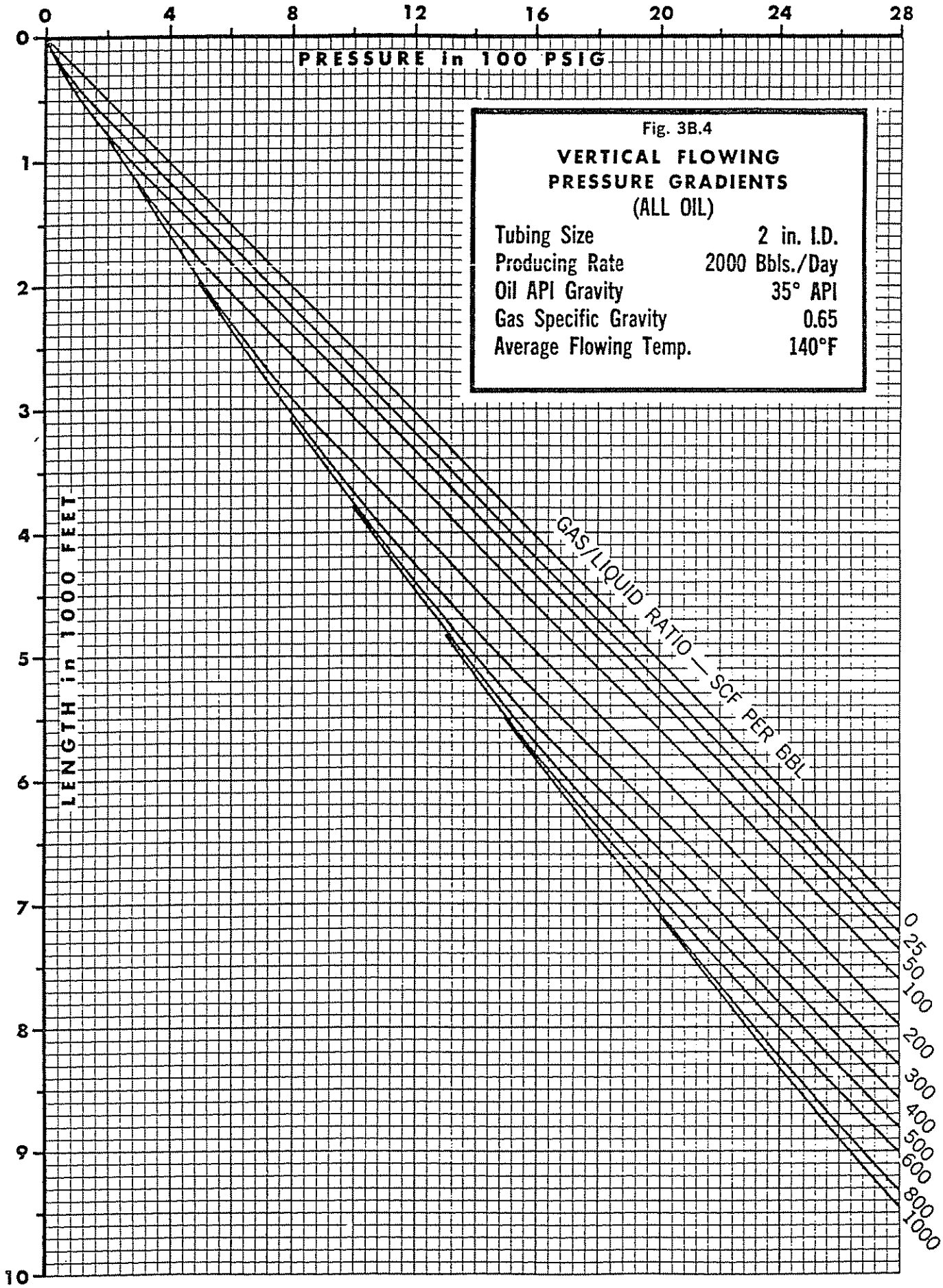
Fig. 3A.5 — PRESSURE DUE TO GAS COLUMN WEIGHT




0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.1
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL OIL)
Tubing Size 2 in. I.D.
Producing Rate 500 Bbls./Day
Oil API Gravity 35° API
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800


0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.2
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL OIL)
Tubing Size 2 in. I.D.
Producing Rate 1000 Bbls./Day
Oil API Gravity 35° API
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000
1200
1500


0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.3
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL OIL)
Tubing Size 2 in. I.D.
Producing Rate 1500 Bbls./Day
Oil API Gravity 35° API
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000
1200


0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.4
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL OIL)
Tubing Size 2 in. I.D.
Producing Rate 2000 Bbls./Day
Oil API Gravity 35° API
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000

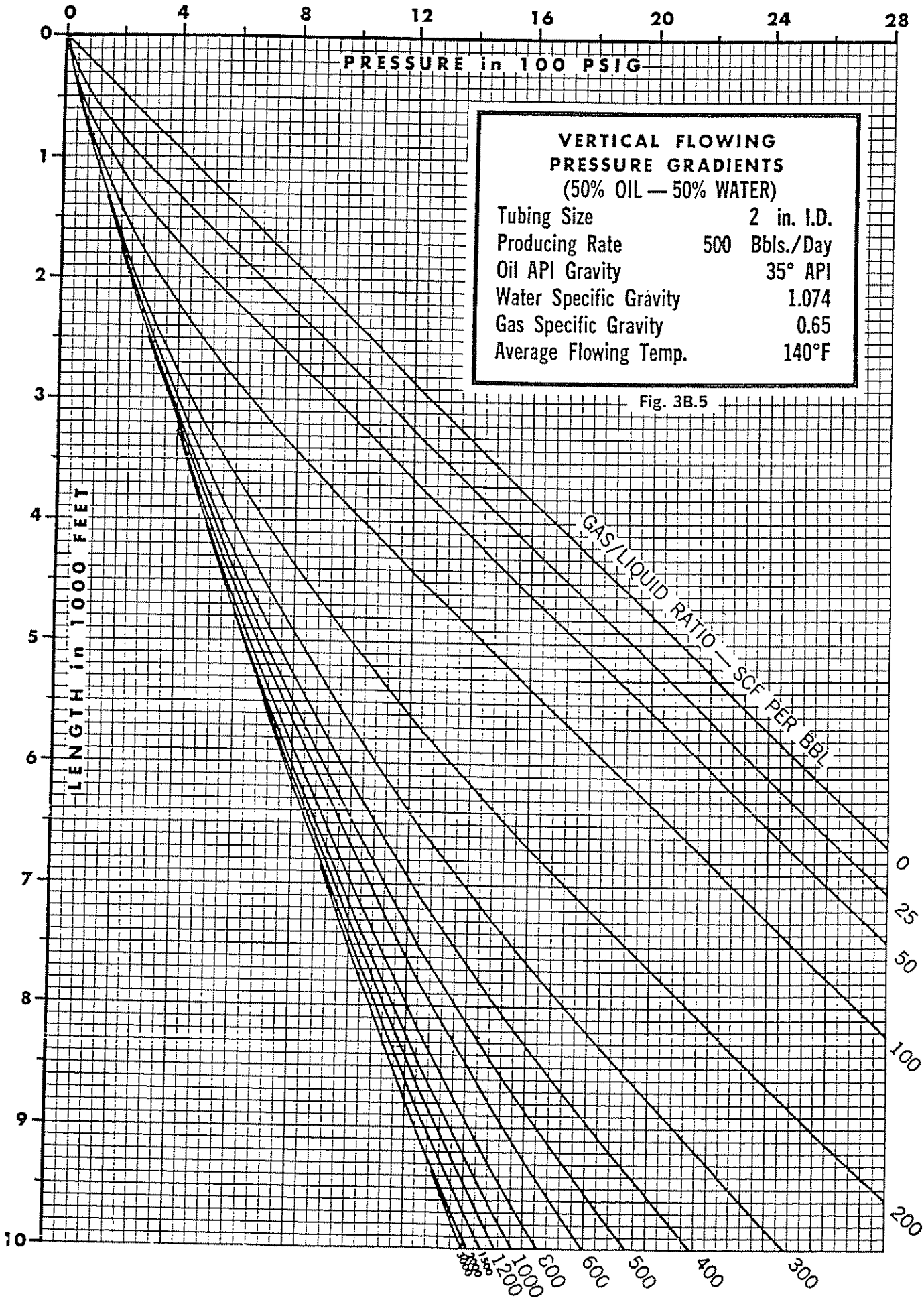
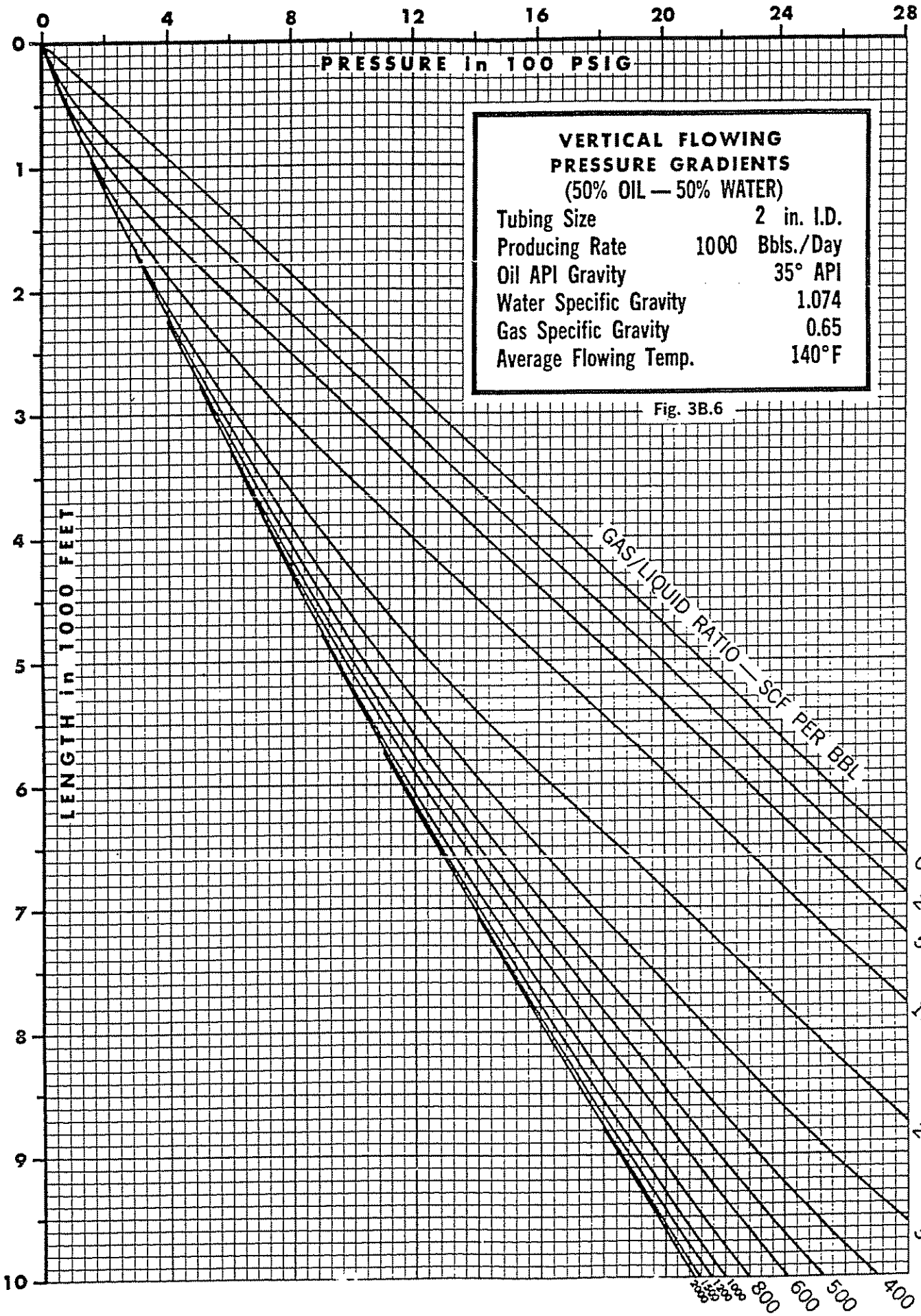
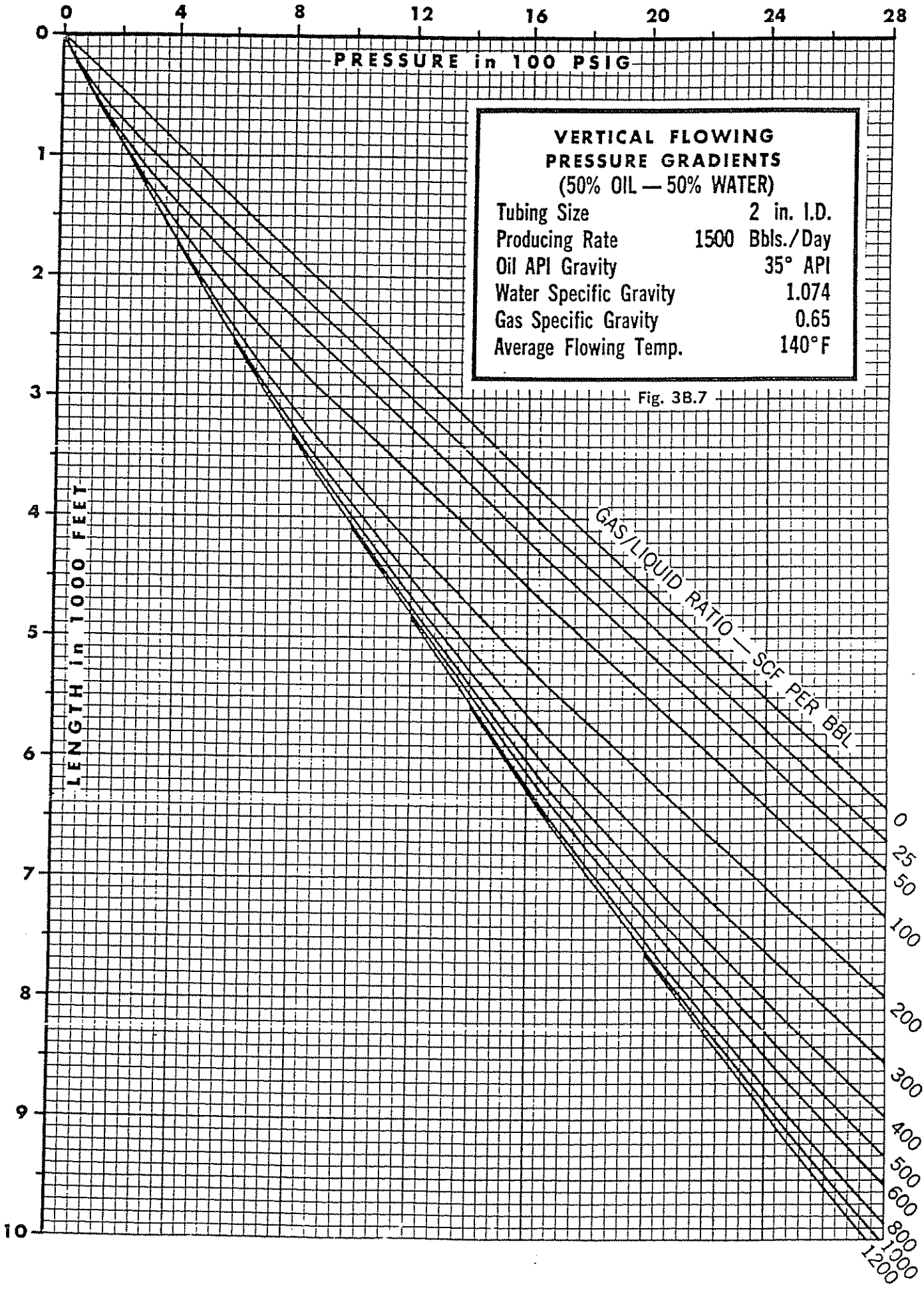
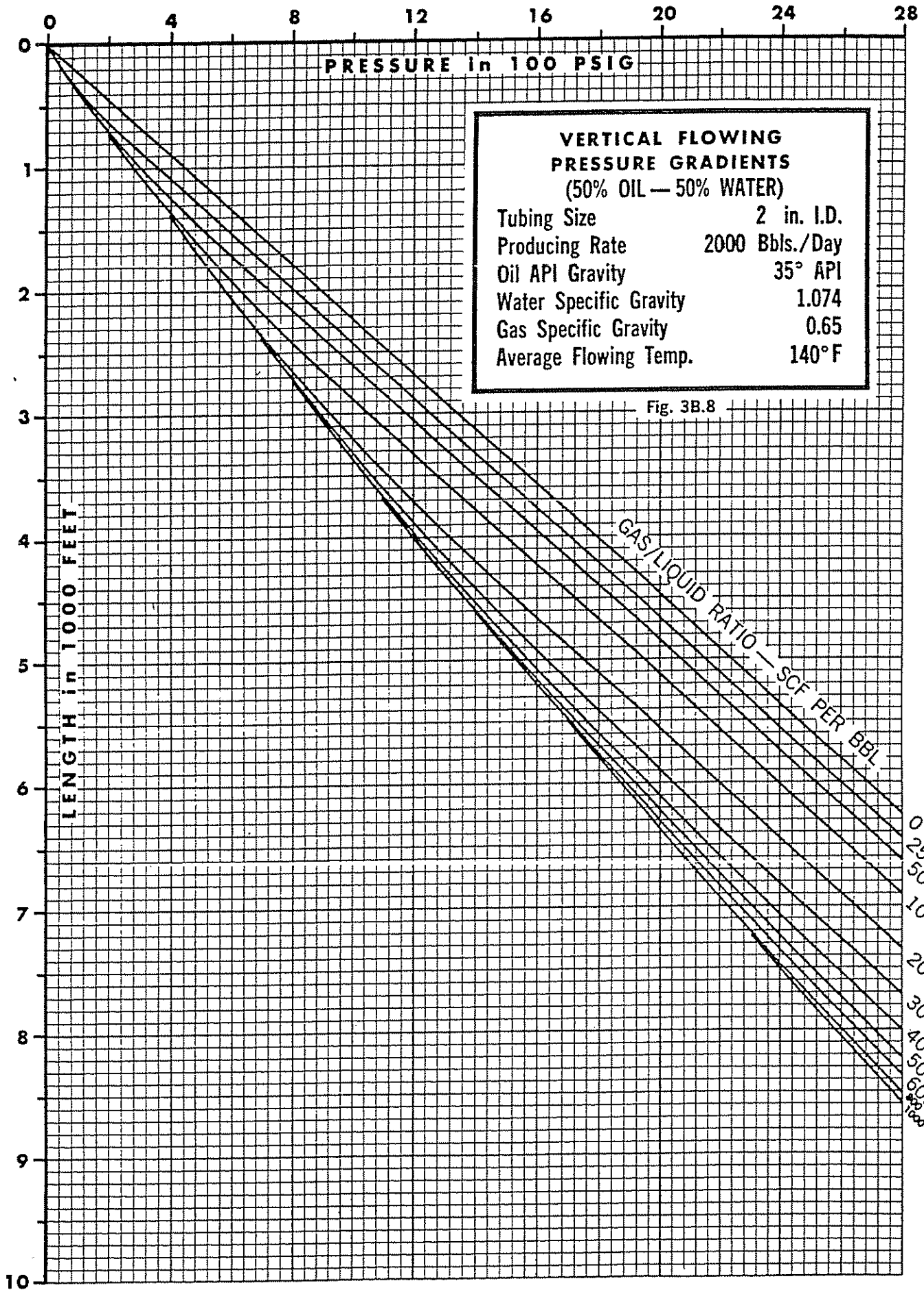
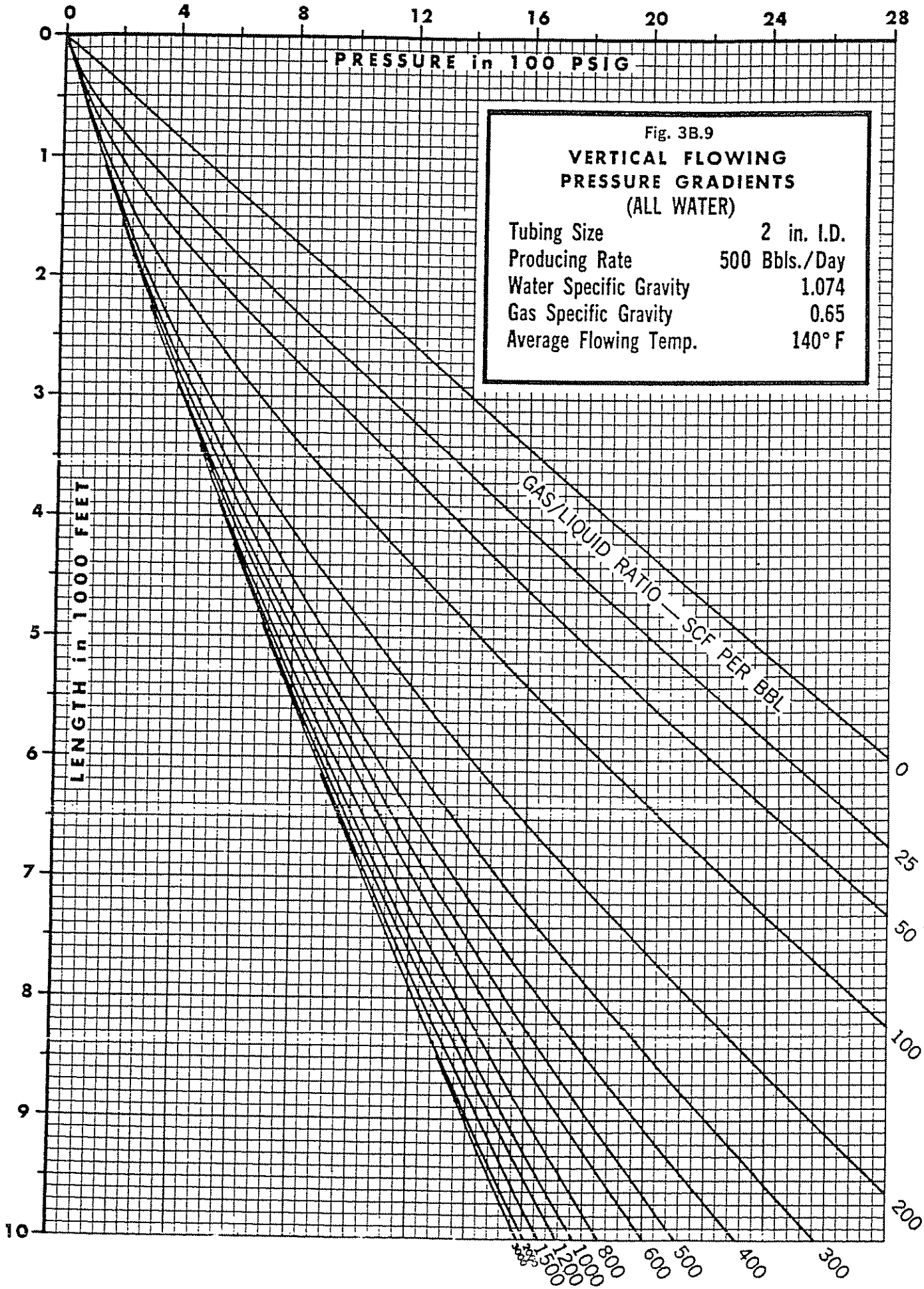

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 2 in. I.D.
Producing Rate 500 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000
1200

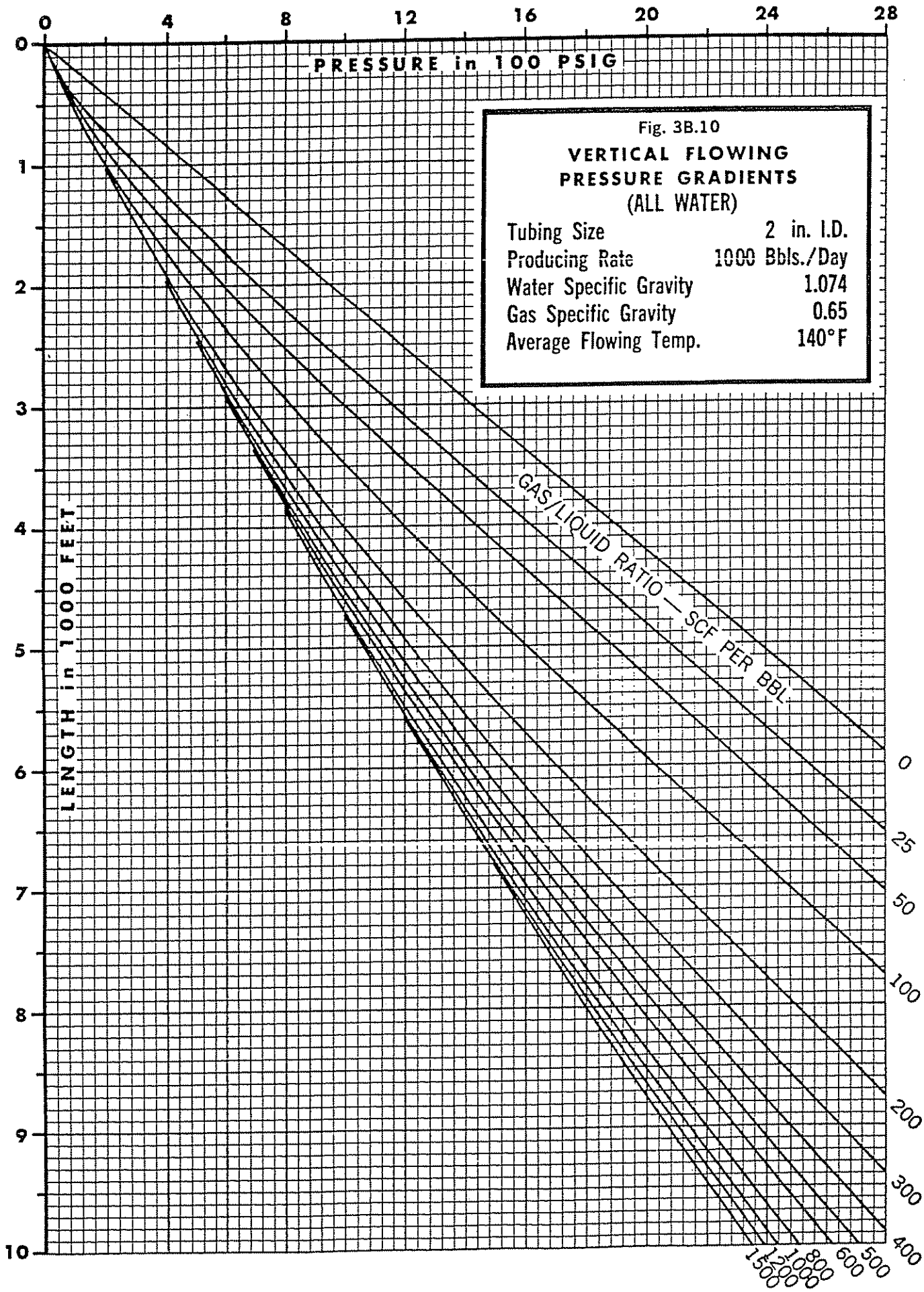
Fig. 3B.5

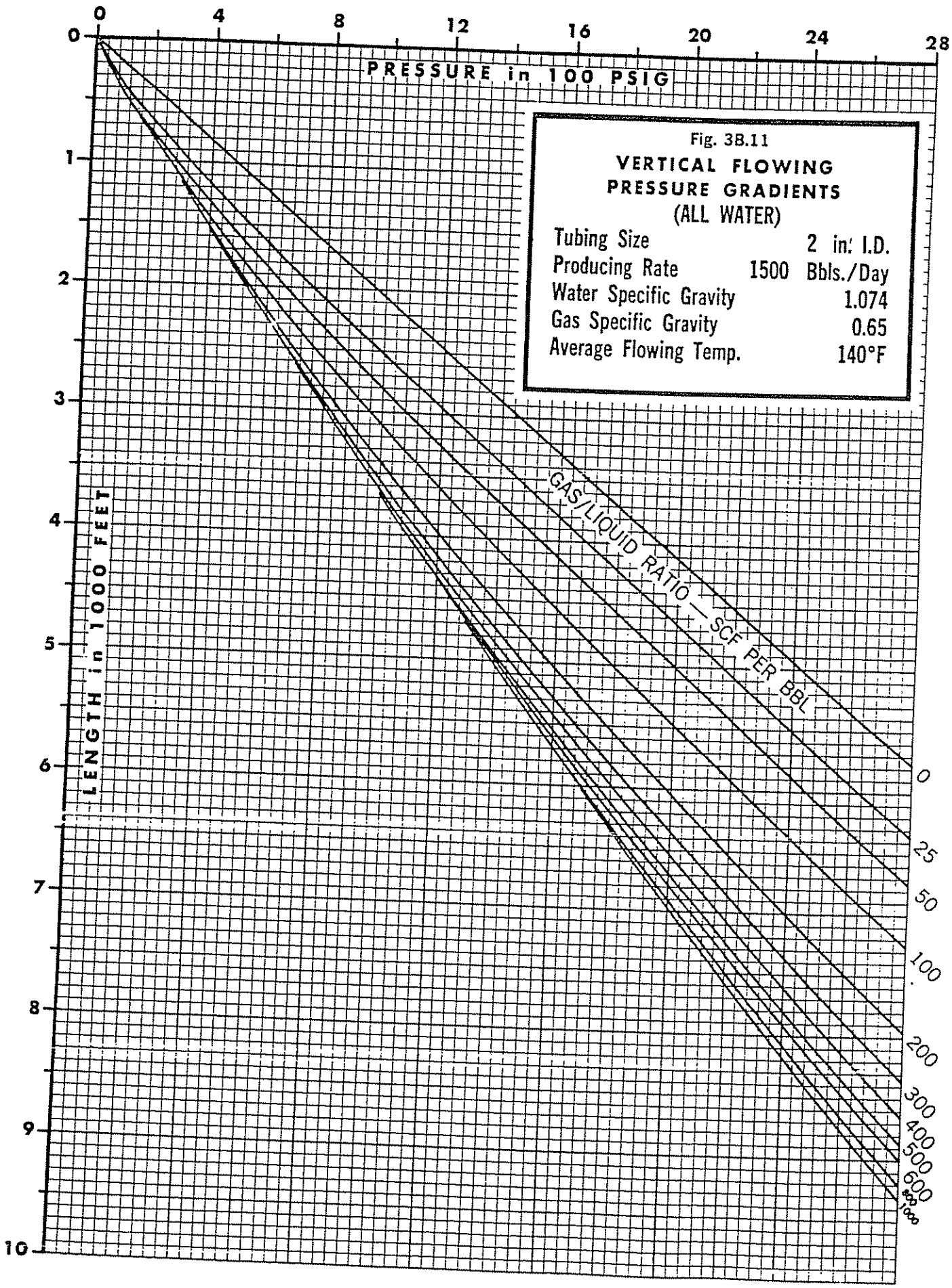

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 2 in. I.D.
Producing Rate 1000 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F
Fig. 3B.6
LENGTH in 1000 FEET
0
1
2
3
4
5
6
7
8
9
10
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800

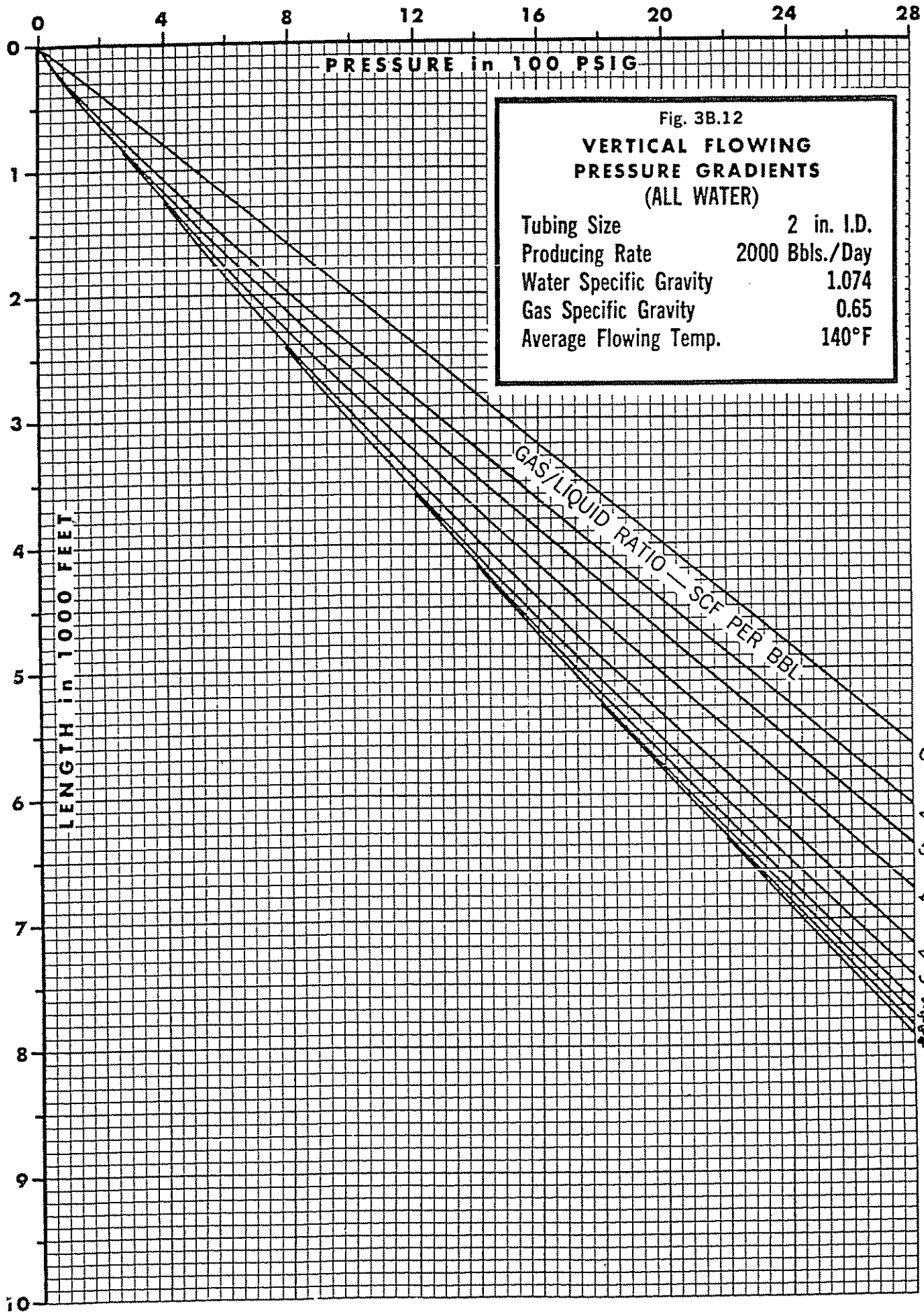

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 2 in. I.D.
Producing Rate 1500 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F
Fig. 3B.7
LENGTH in 1000 FEET
0
1
2
3
4
5
6
7
8
9
10
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000
1200

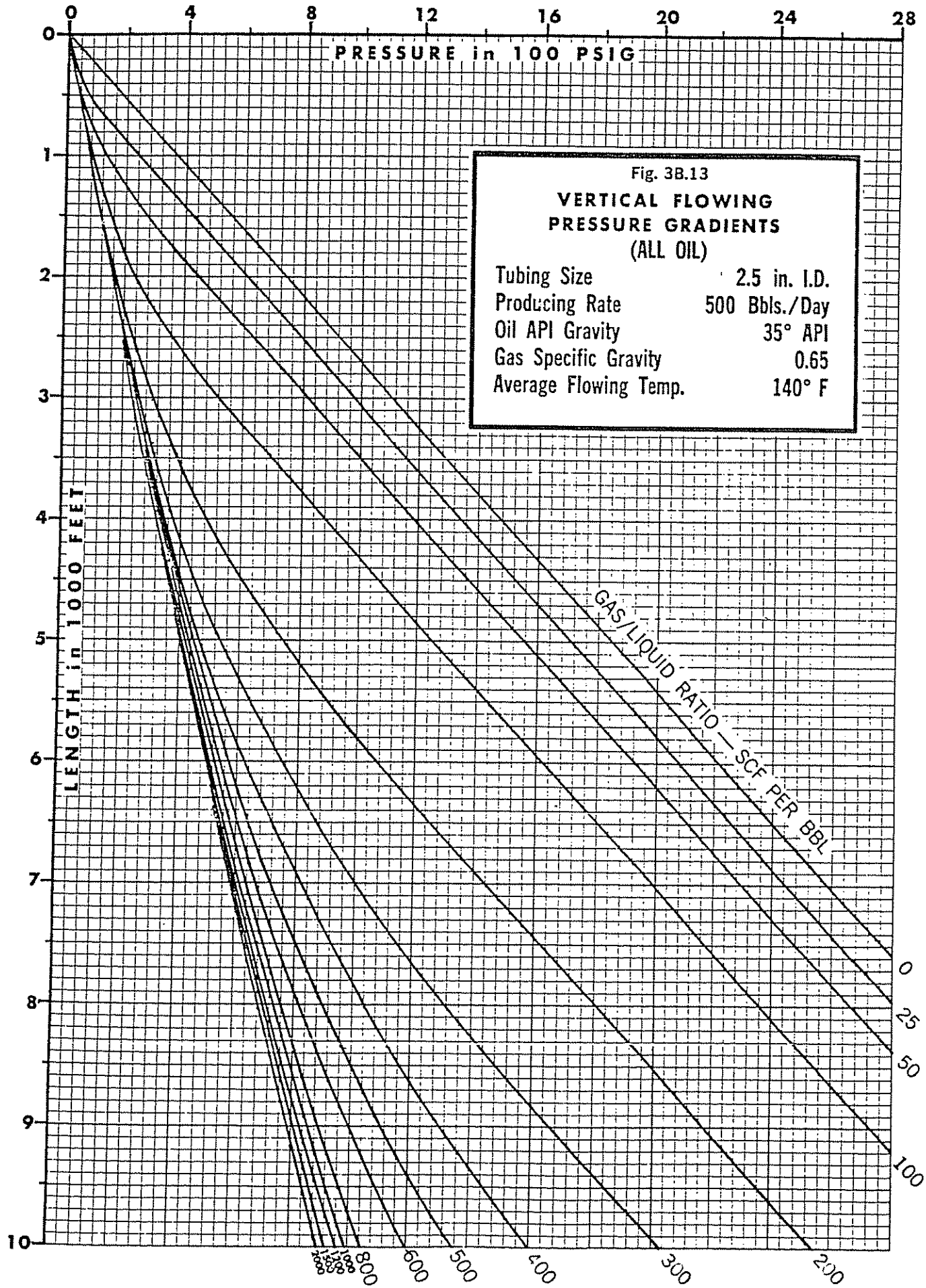

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 2 in. I.D.
Producing Rate 2000 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F
Fig. 3B.8
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
1000

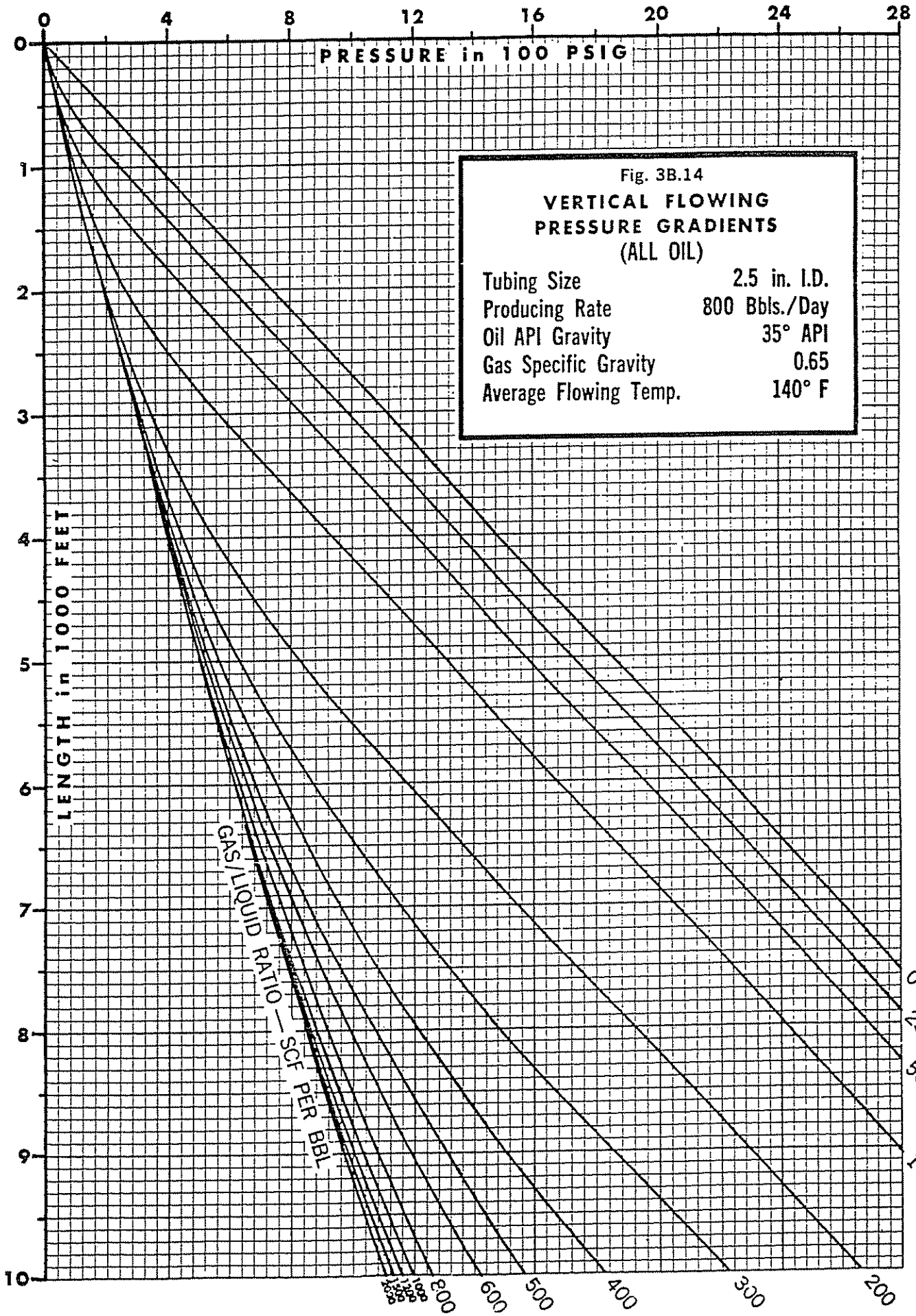

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.9
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Tubing Size 2 in. I.D.
Producing Rate 500 Bbls./Day
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000
1200
1500

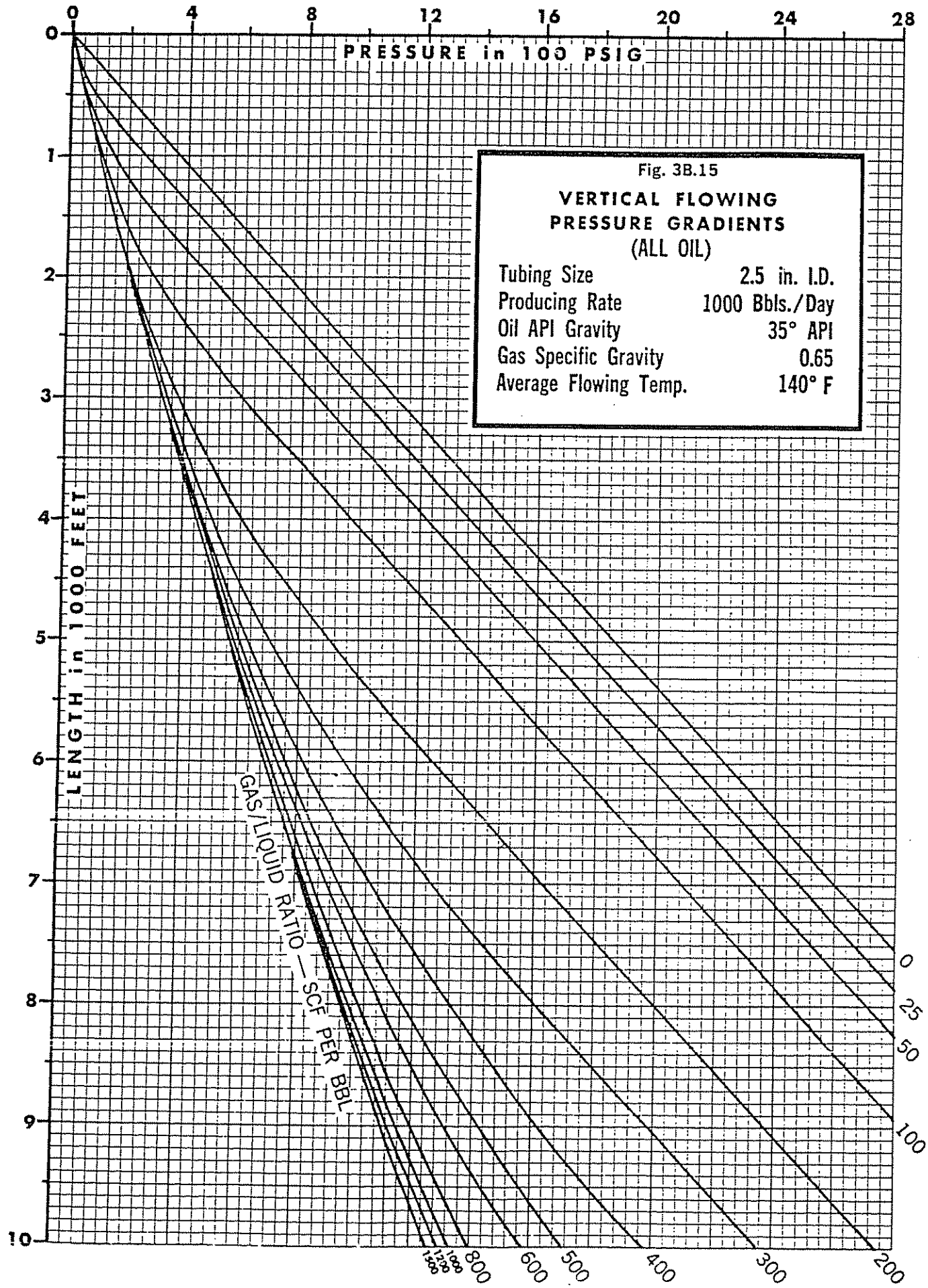

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.10
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Tubing Size 2 in. I.D.
Producing Rate 1000 Bbls./Day
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F
GAS/LIQUID RATIO — SCF PER BBL.
0
25
50
100
200
300
400
500
600
800
1000
1200
1500

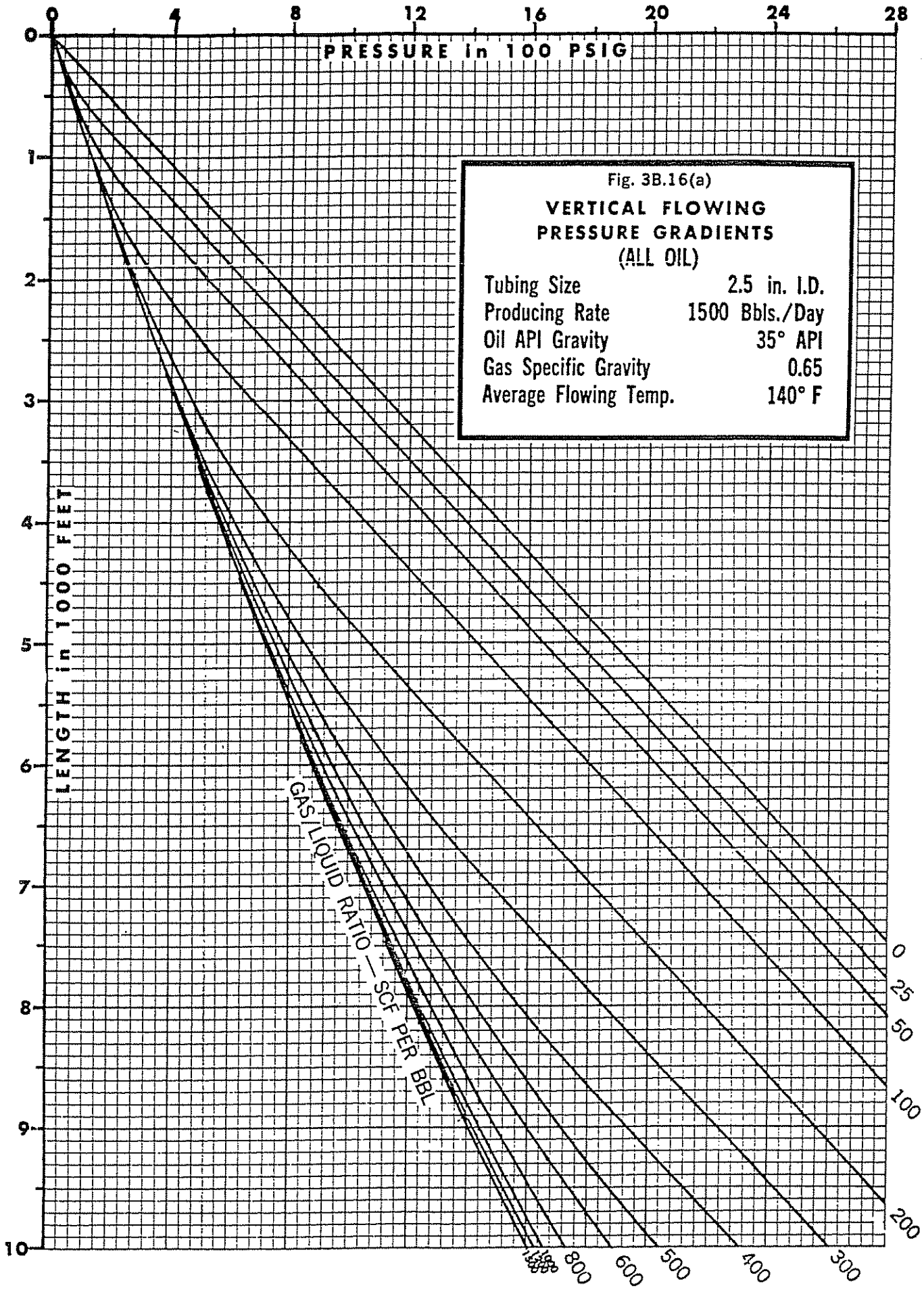

Fig. 3B.11
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Tubing Size 2 in. I.D.
Producing Rate 1500 Bbls./Day
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F
PRESSURE in 100 PSIG
0
4
8
12
16
20
24
28
LENGTH in 1000 FEET
1
2
3
4
5
6
7
8
9
10
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000

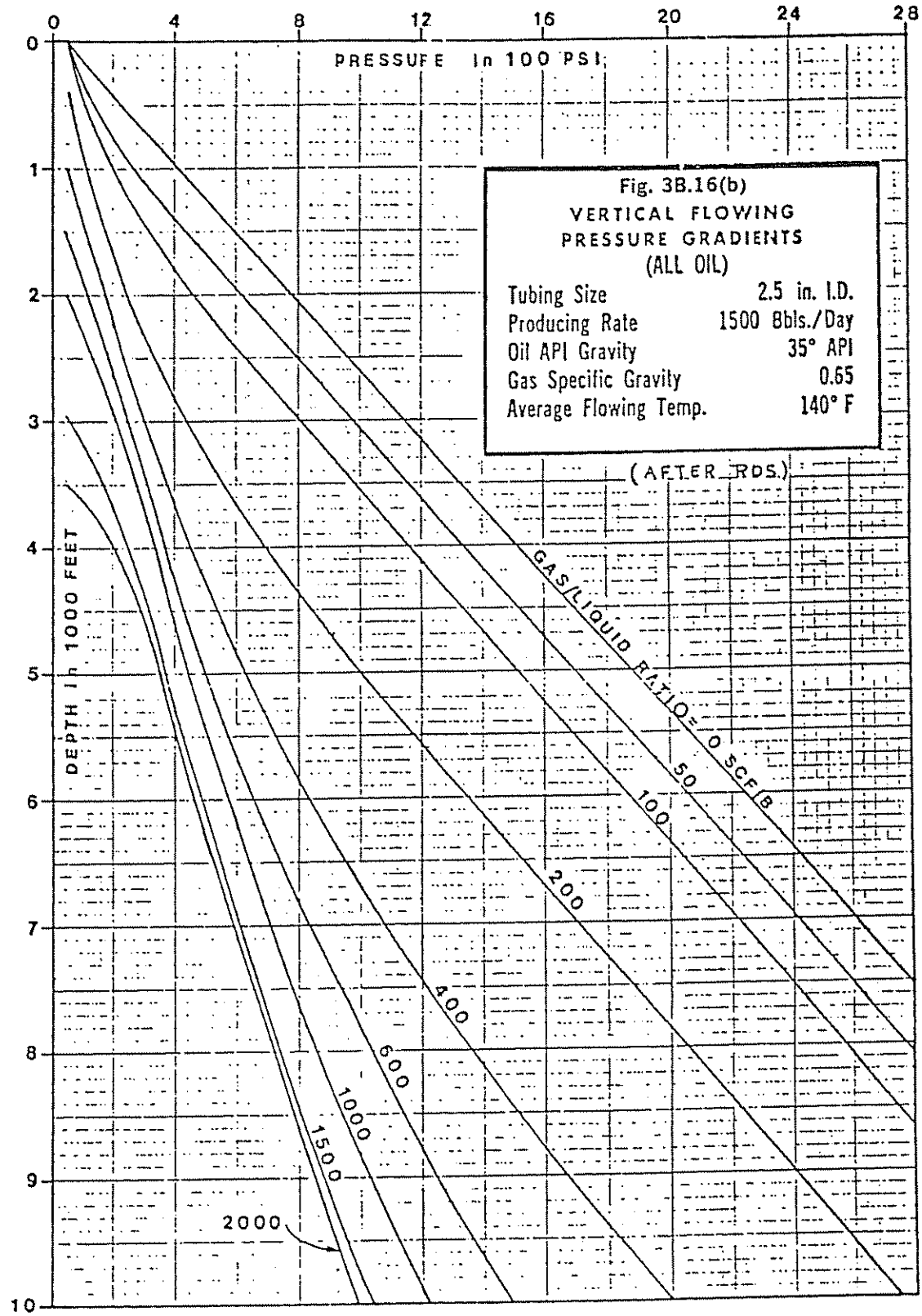

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.12
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Tubing Size 2 in. I.D.
Producing Rate 2000 Bbls./Day
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
∞

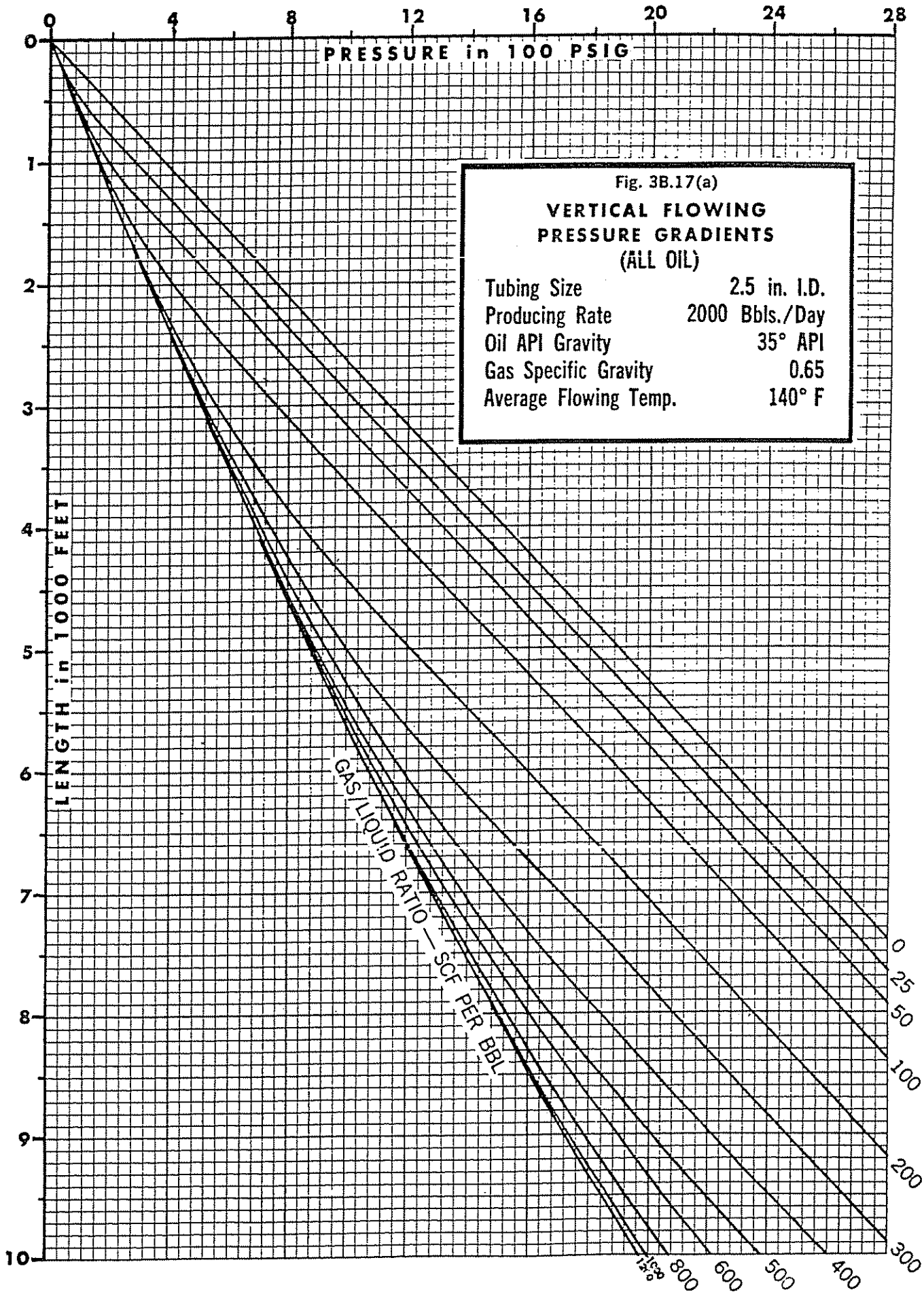

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.13
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL OIL)
Tubing Size 2.5 in. I.D.
Producing Rate 500 Bbls./Day
Oil API Gravity 35° API
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800

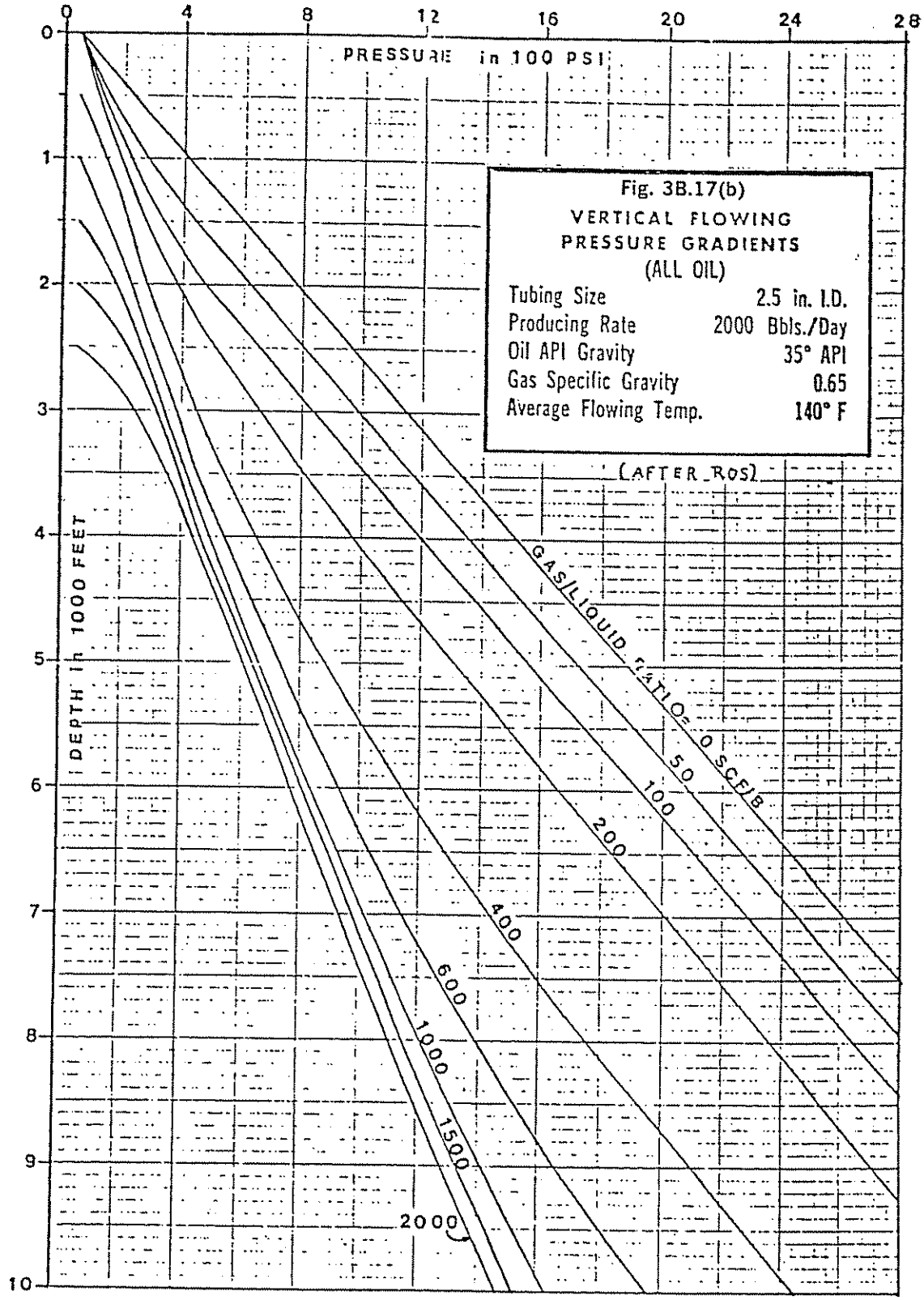

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.14
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL OIL)
Tubing Size 2.5 in. I.D.
Producing Rate 800 Bbls./Day
Oil API Gravity 35° API
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800

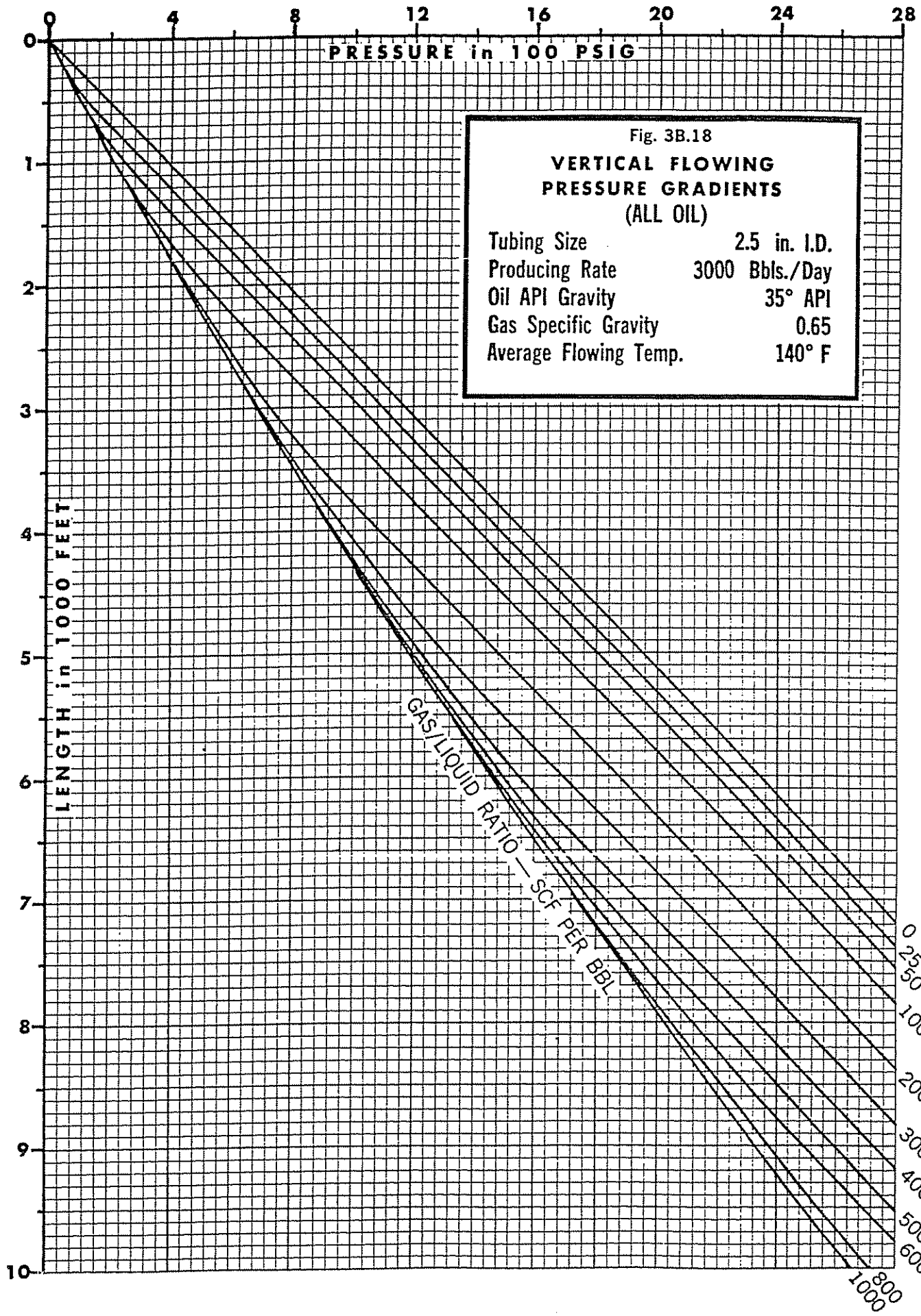

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.15
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL OIL)
Tubing Size 2.5 in. I.D.
Producing Rate 1000 Bbls./Day
Oil API Gravity 35° API
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000
1250
1500

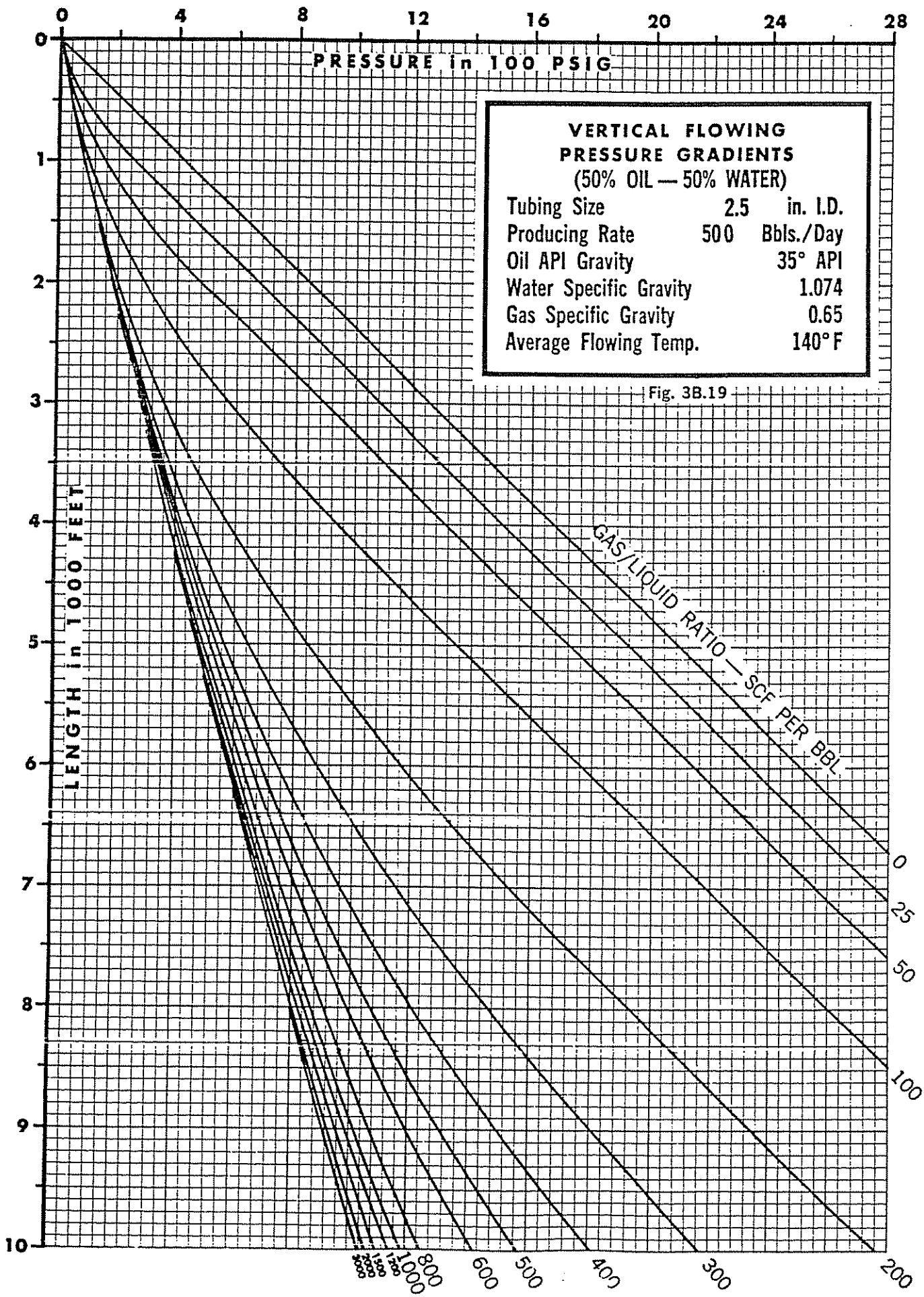

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.16(a)
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL OIL)
Tubing Size 2.5 in. I.D.
Producing Rate 1500 Bbls./Day
Oil API Gravity 35° API
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000

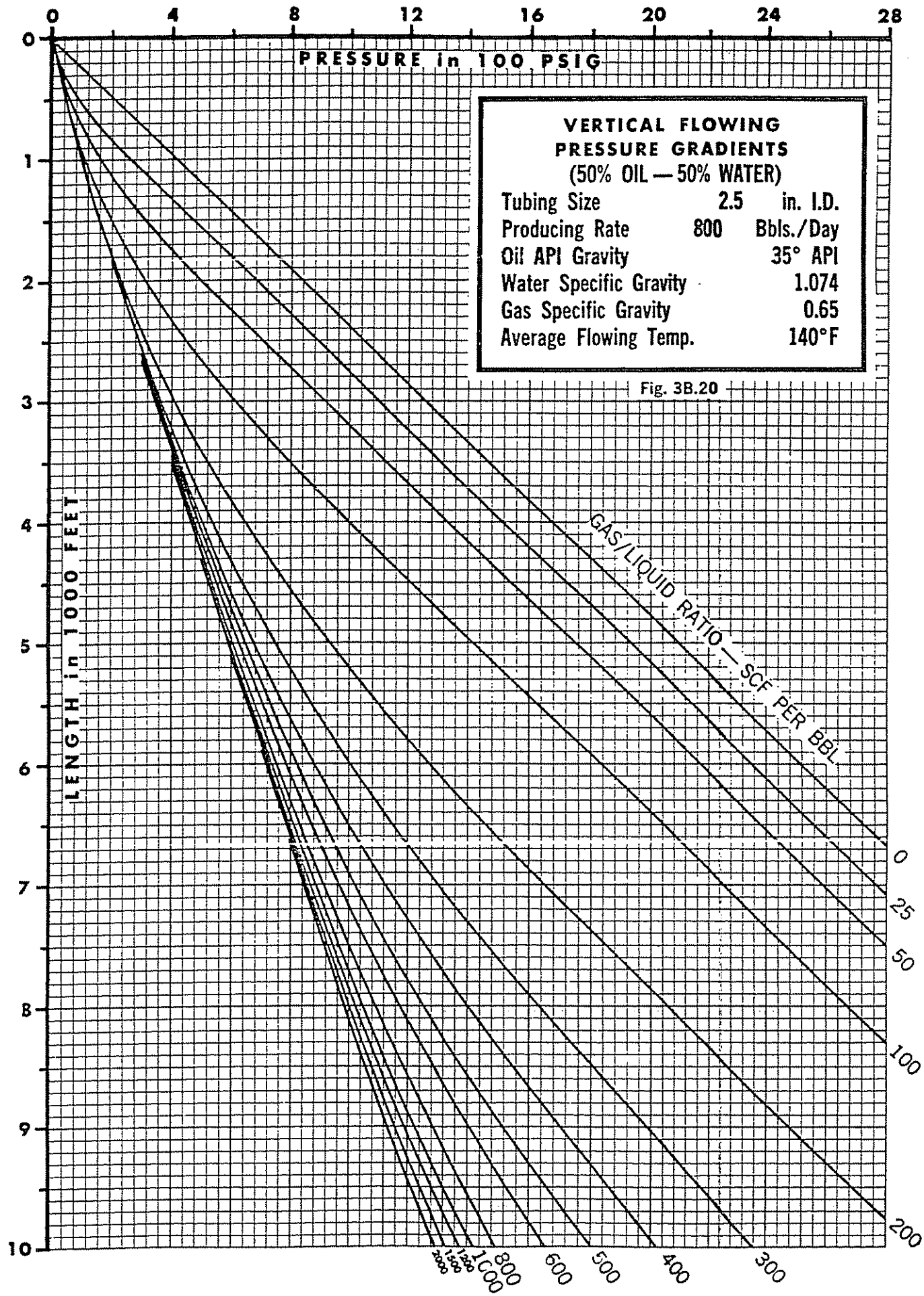

0
4
8
12
16
20
24
28
PRESSURE In 100 PSI
DEPTH In 1000 FEET
0
1
2
3
4
5
6
7
8
9
10
Fig. 3B.16(b)
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL OIL)
Tubing Size 2.5 in. I.D.
Producing Rate 1500 Bbls./Day
Oil API Gravity 35° API
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
(AFTER RDS.)
GAS/LIQUID RATIO = 0 SCF/B
50
100
200
400
600
1000
1500
2000


0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.17(a)
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL OIL)
Tubing Size 2.5 in. I.D.
Producing Rate 2000 Bbls./Day
Oil API Gravity 35° API
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1,000
1,250


0
4
8
12
16
20
24
28
PRESSURE in 100 PSI
DEPTH in 1000 FEET
0
1
2
3
4
5
6
7
8
9
10
Fig. 3B.17(b)
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL OIL)
Tubing Size 2.5 in. I.D.
Producing Rate 2000 Bbls./Day
Oil API Gravity 35° API
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
(AFTER ROS)
GAS/LIQUID RATIOS 0 SCF/B
50
100
200
400
600
1000
1500
2000


0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.18
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL OIL)
Tubing Size 2.5 in. I.D.
Producing Rate 3000 Bbls./Day
Oil API Gravity 35° API
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000


0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 2.5 in. I.D.
Producing Rate 500 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
Fig. 3B.19
LENGTH in 1000 FEET
0
1
2
3
4
5
6
7
8
9
10
GAS/LIQUID RATIO — SCF PER BBL.
0
25
50
100
200
300
400
500
600
800
1000


0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 2.5 in. I.D.
Producing Rate 800 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F
Fig. 3B.20
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000

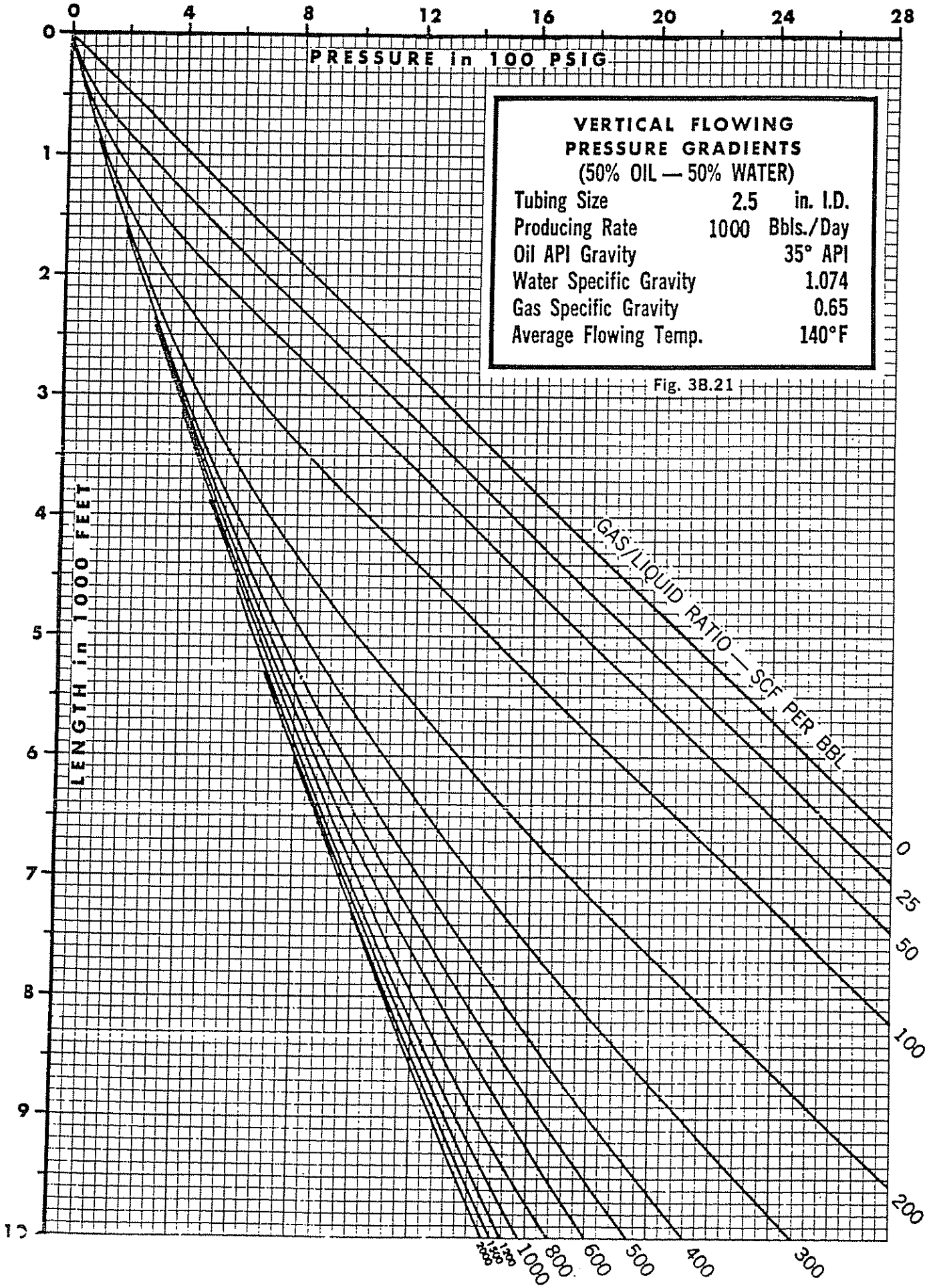
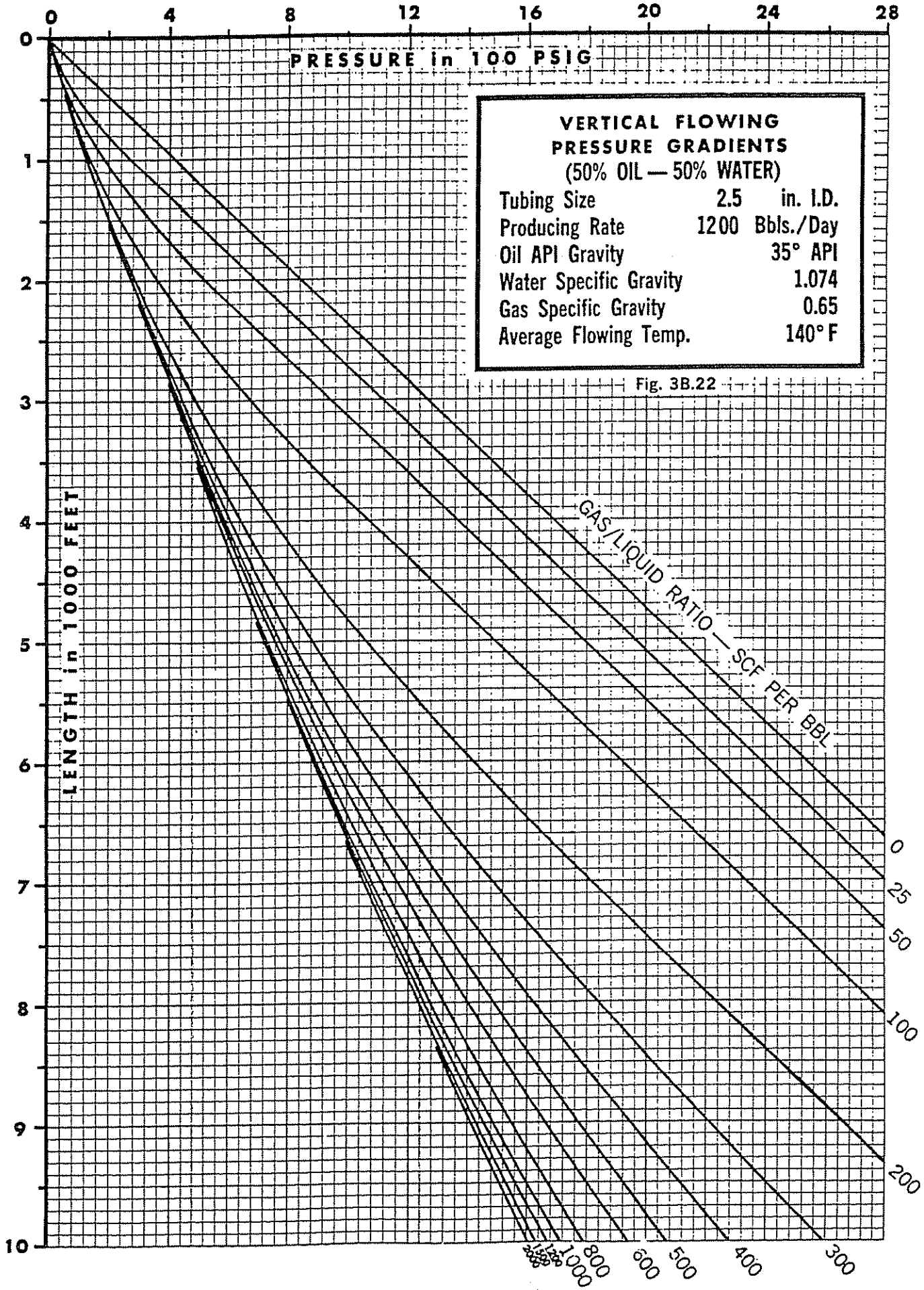
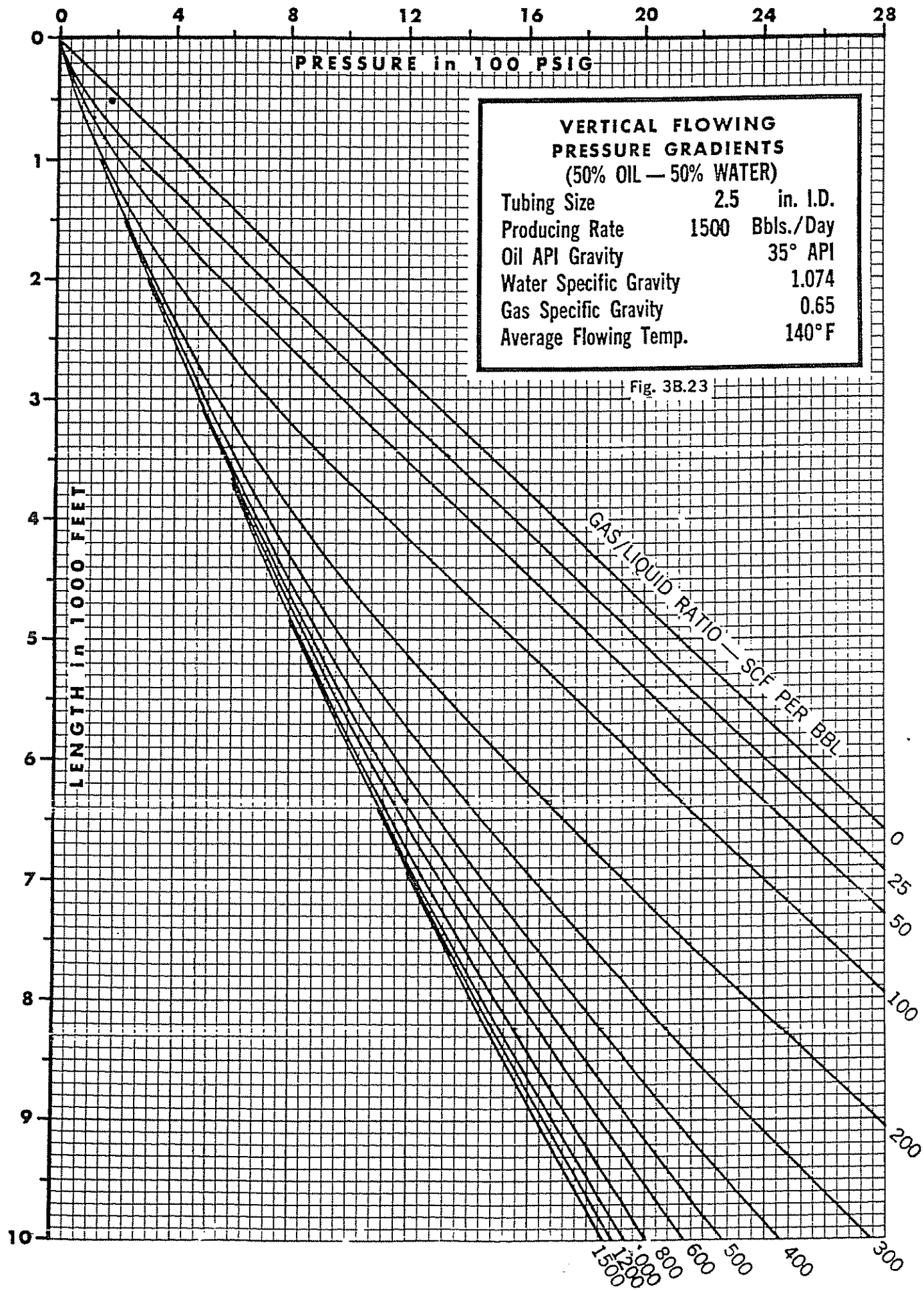
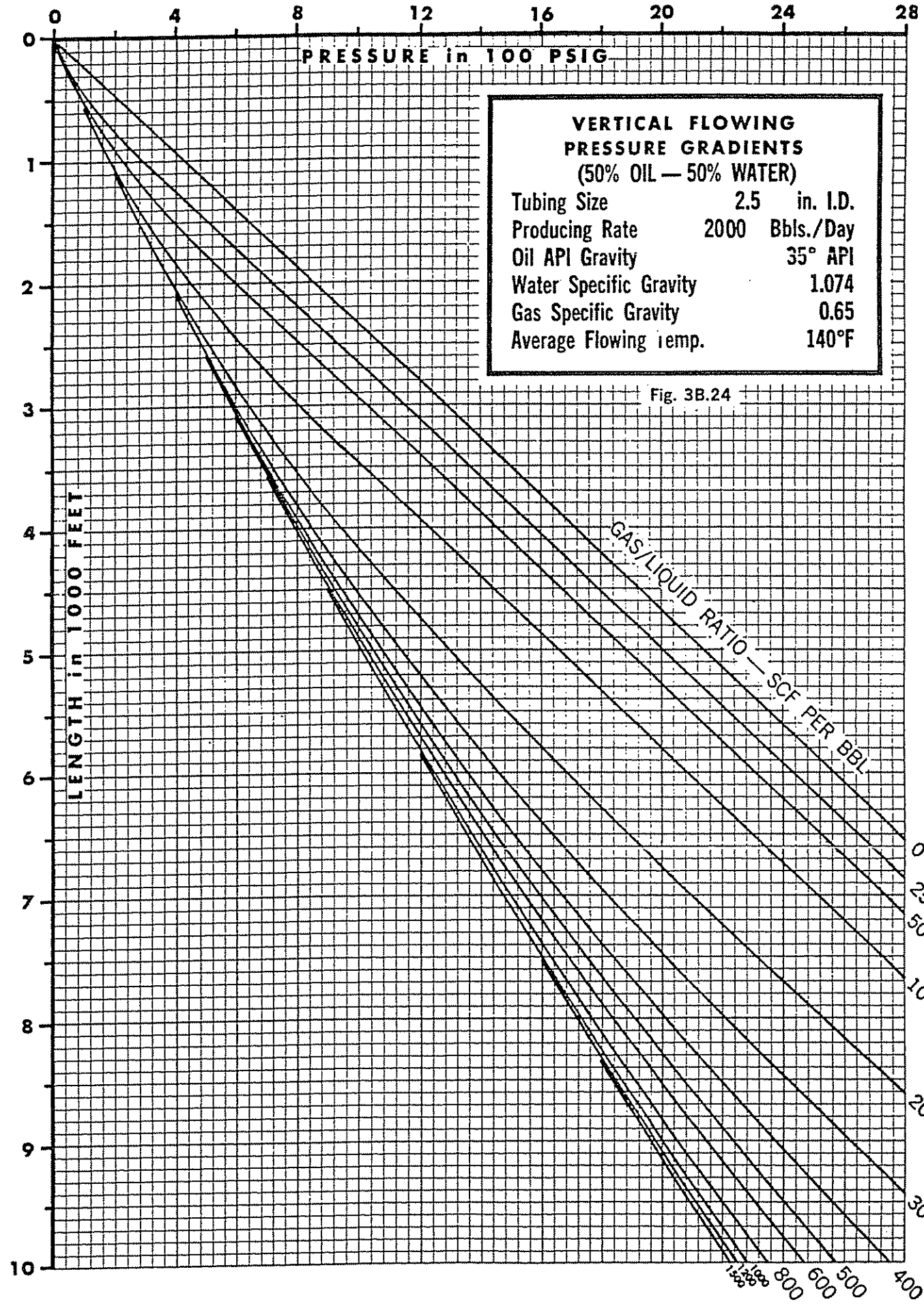
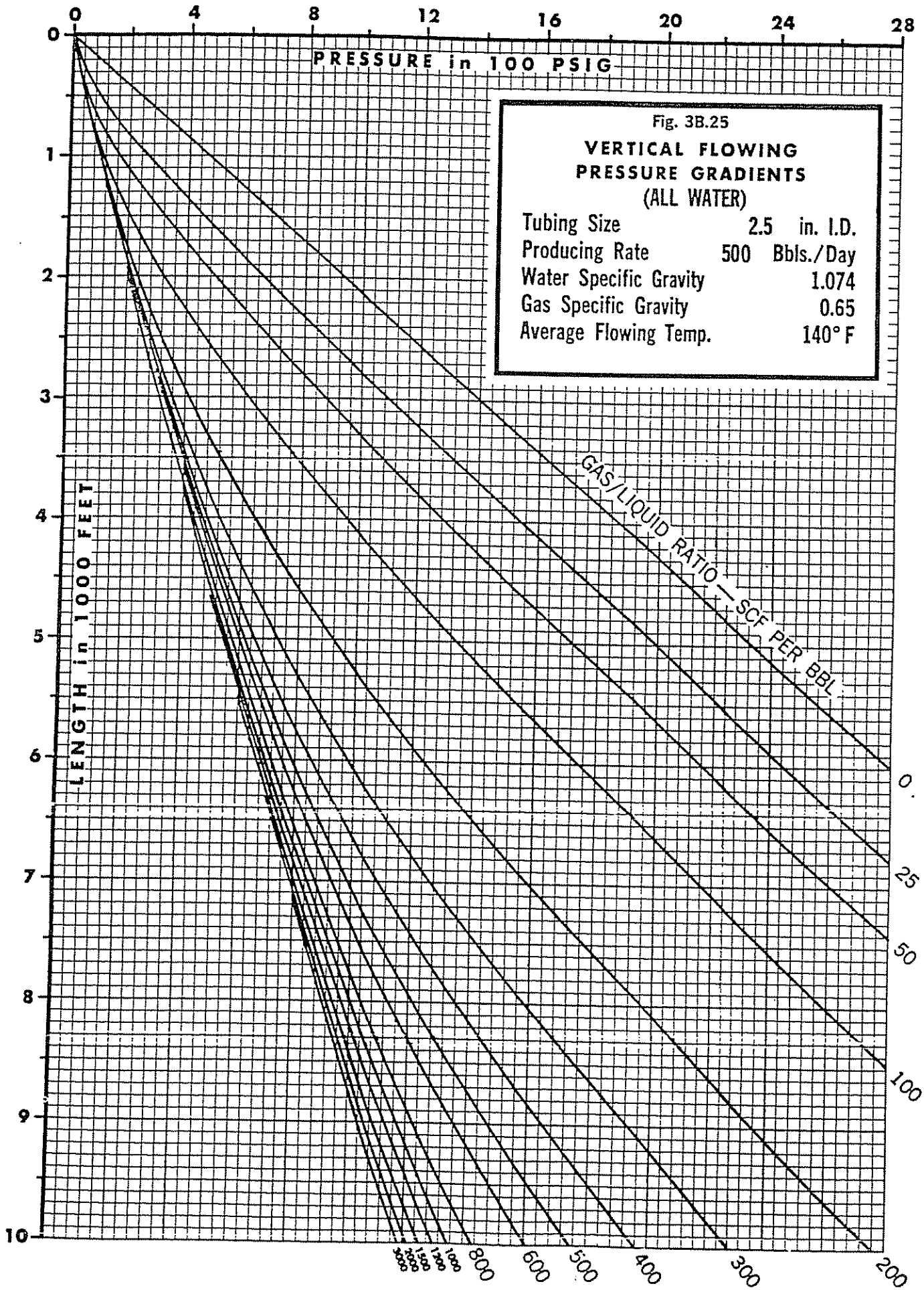

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 2.5 in. I.D.
Producing Rate 1000 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000
1200

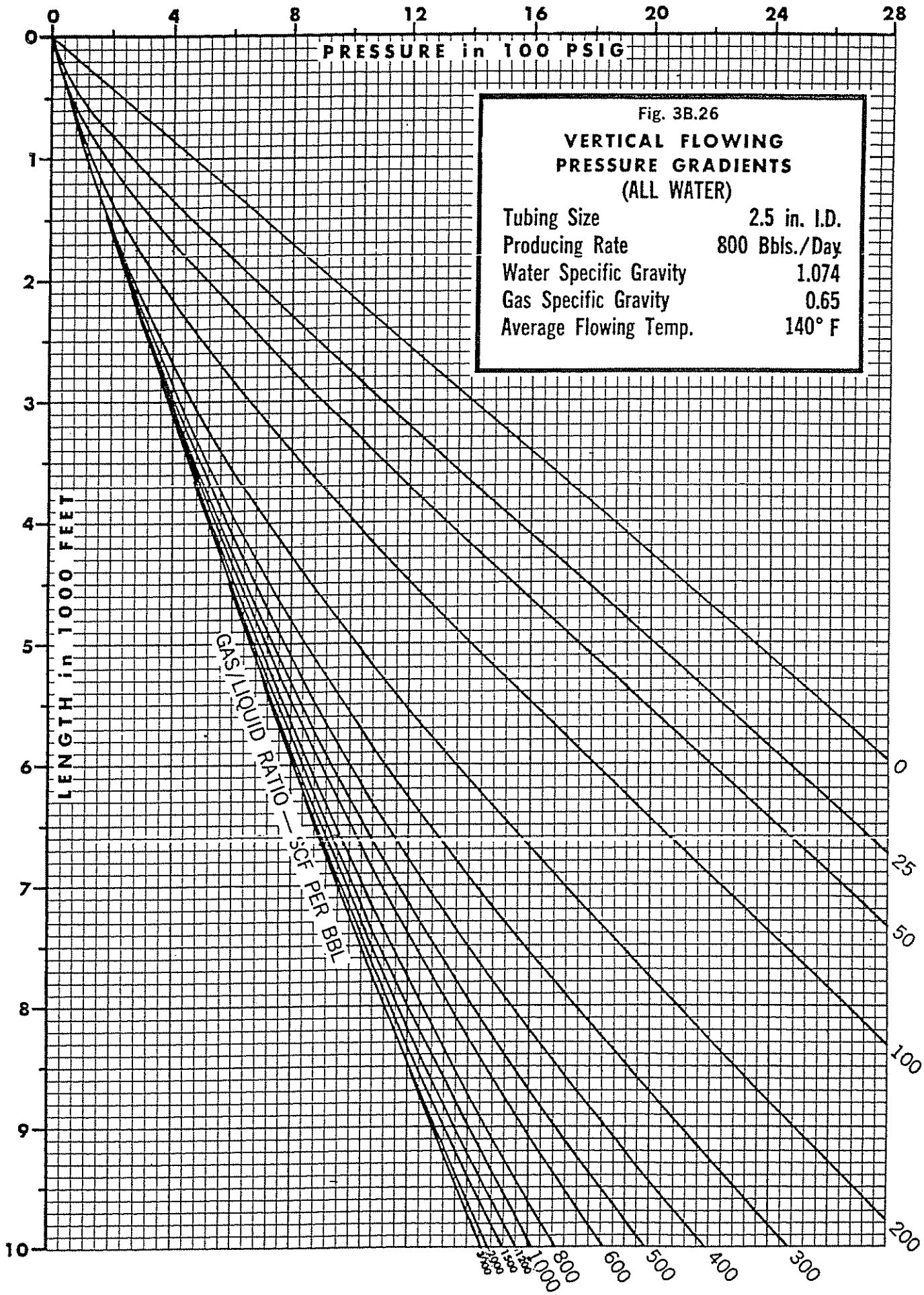
Fig. 38.21


0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 2.5 in. I.D.
Producing Rate 1200 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
Fig. 3B.22
LENGTH in 1000 FEET
0
1
2
3
4
5
6
7
8
9
10
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000


0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 2.5 in. I.D.
Producing Rate 1500 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
Fig. 3B.23
LENGTH in 1000 FEET
0
1
2
3
4
5
6
7
8
9
10
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000
1200
1500


0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 2.5 in. I.D.
Producing Rate 2000 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F
Fig. 3B.24
LENGTH in 1000 FEET
0
1
2
3
4
5
6
7
8
9
10
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000
1200
1400


0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.25
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Tubing Size 2.5 in. I.D.
Producing Rate 500 Bbls./Day
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800


0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.26
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Tubing Size 2.5 in. I.D.
Producing Rate 800 Bbls./Day
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000
1250
1500
2000
3000

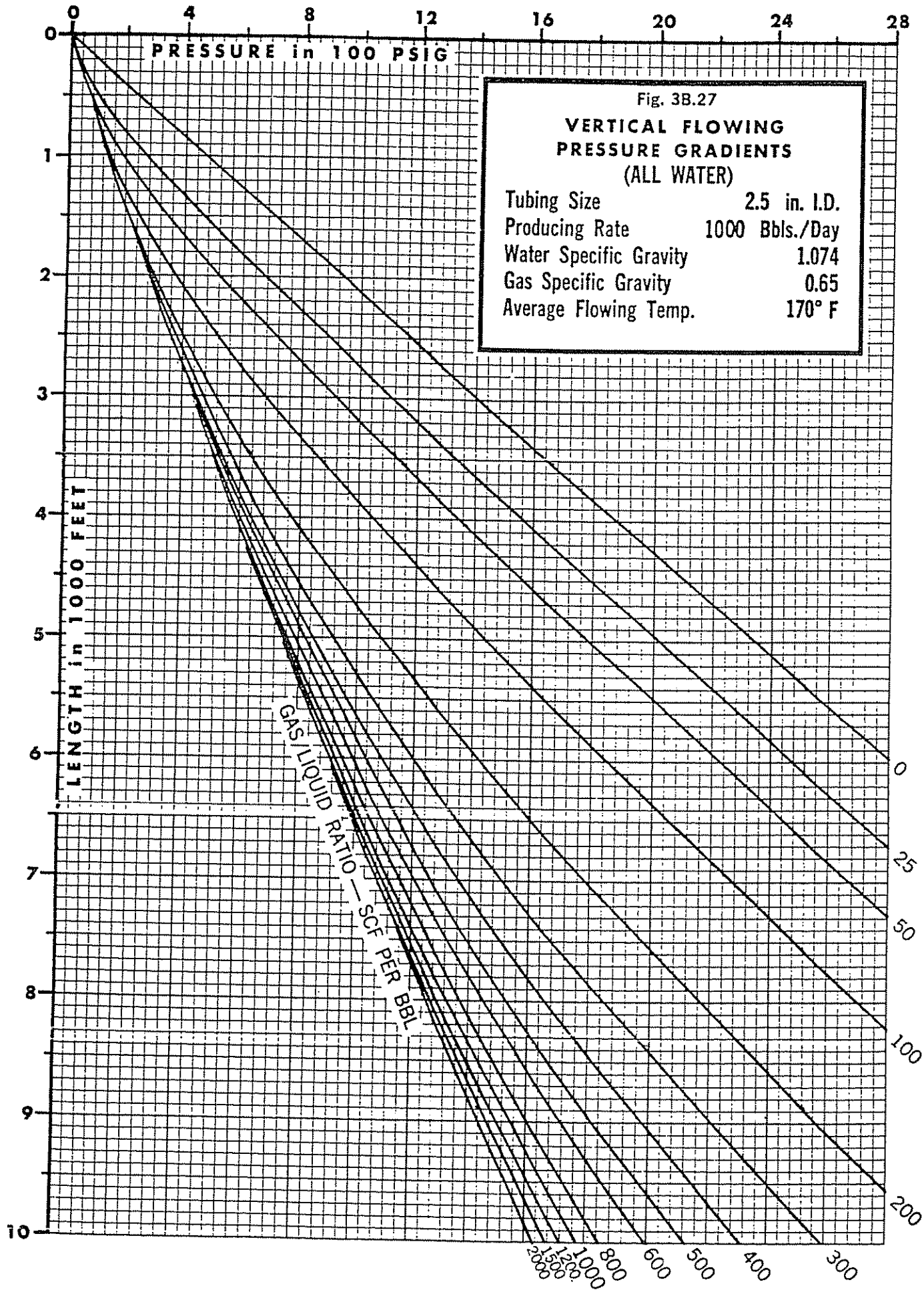
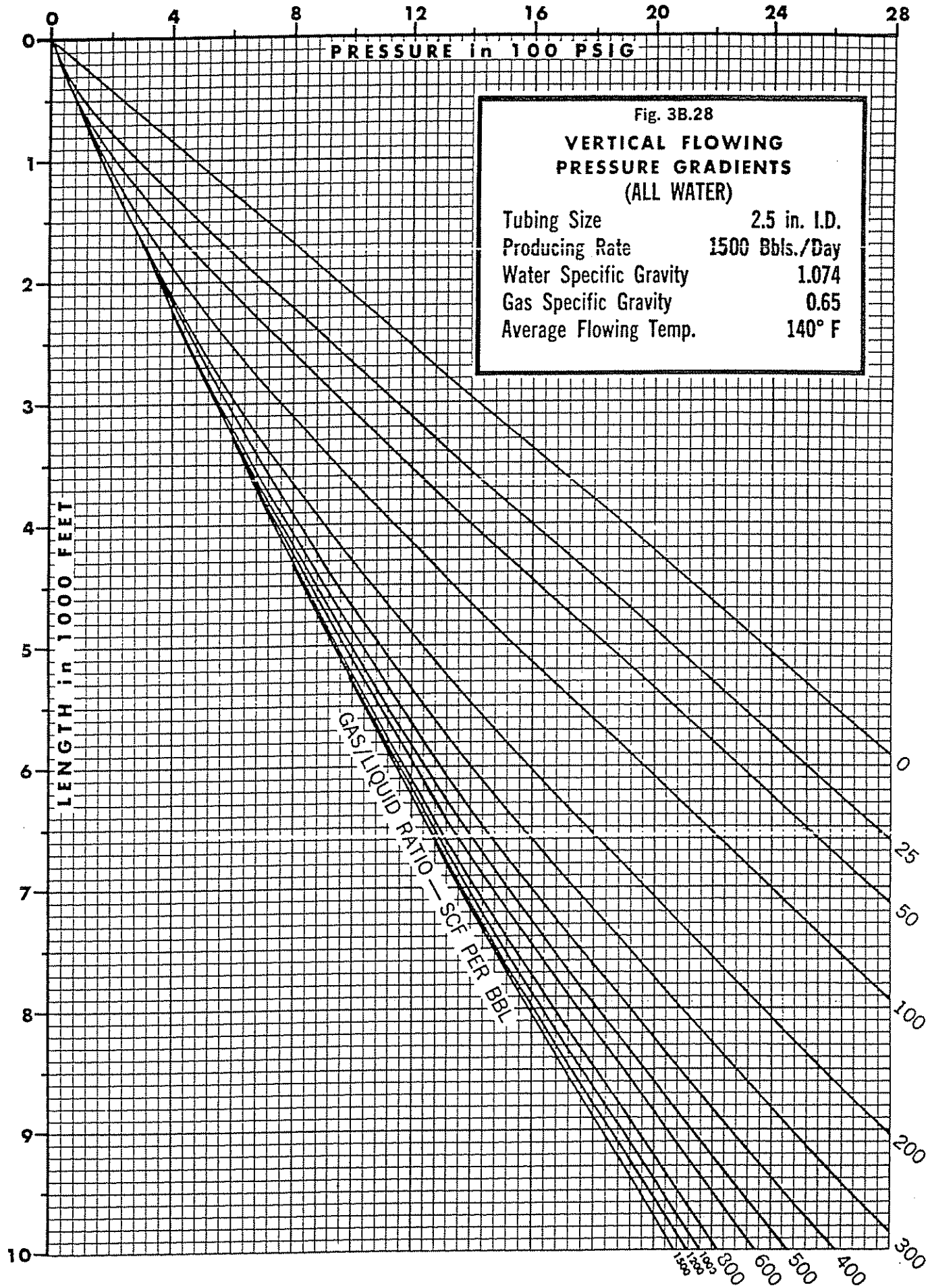
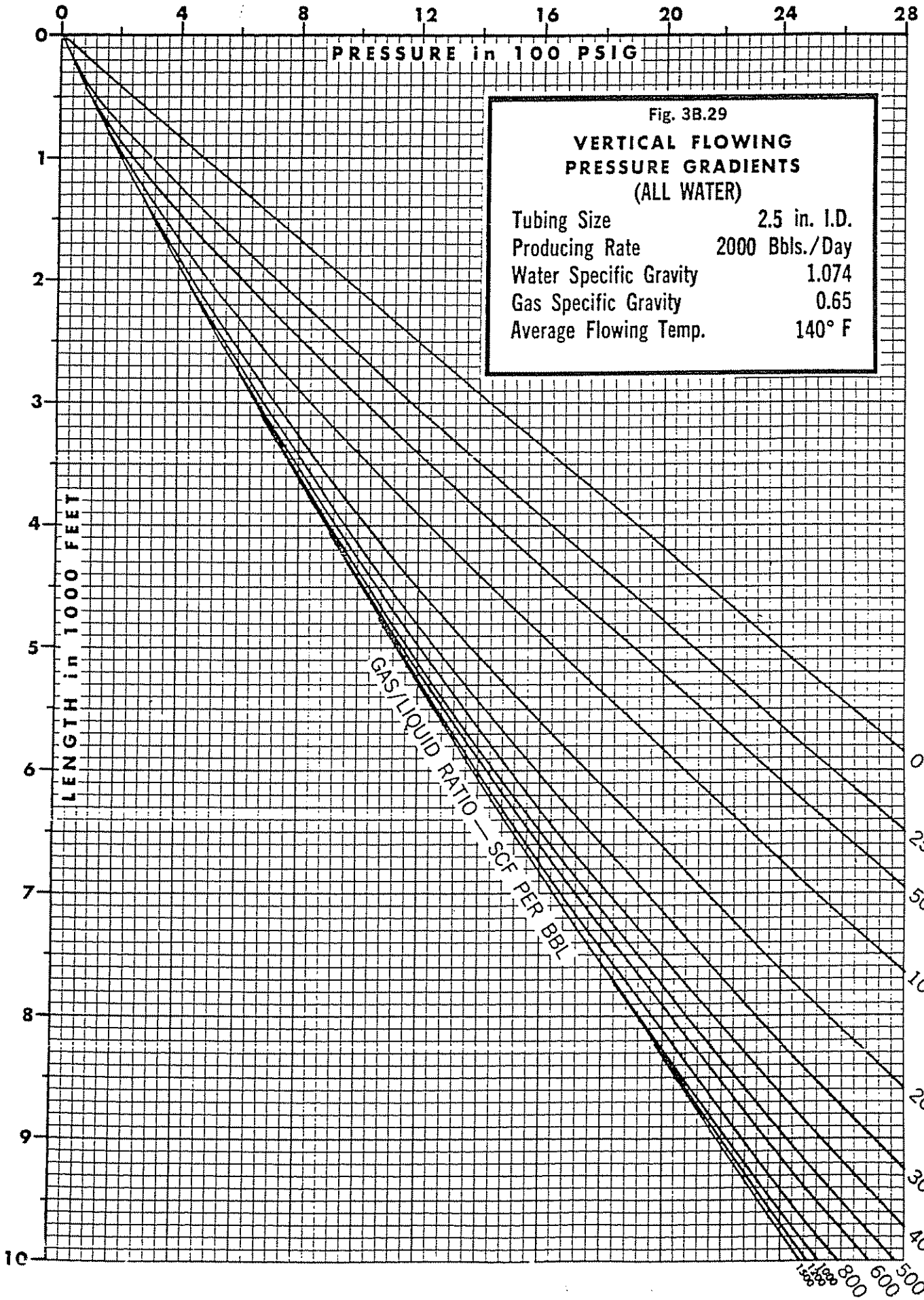


Fig. 3B.27

VERTICAL FLOWING PRESSURE GRADIENTS
(ALL WATER)

Tubing Size	2.5 in. I.D.
Producing Rate	1000 Bbls./Day
Water Specific Gravity	1.074
Gas Specific Gravity	0.65
Average Flowing Temp.	170° F


0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.28
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Tubing Size 2.5 in. I.D.
Producing Rate 1500 Bbls./Day
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000
1200
1400


0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000
1200
1400
Fig. 3B.29
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Tubing Size 2.5 in. I.D.
Producing Rate 2000 Bbls./Day
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F

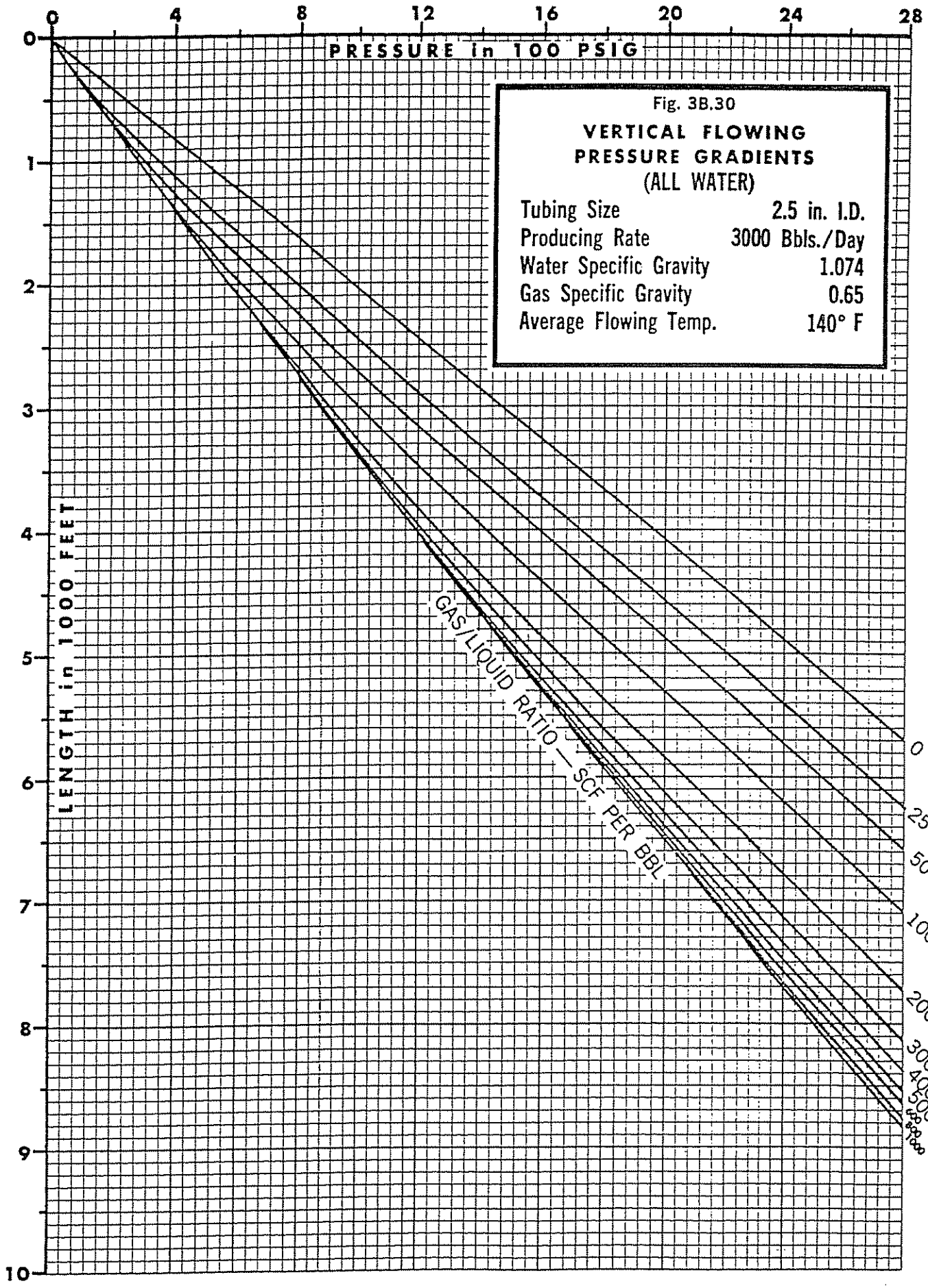
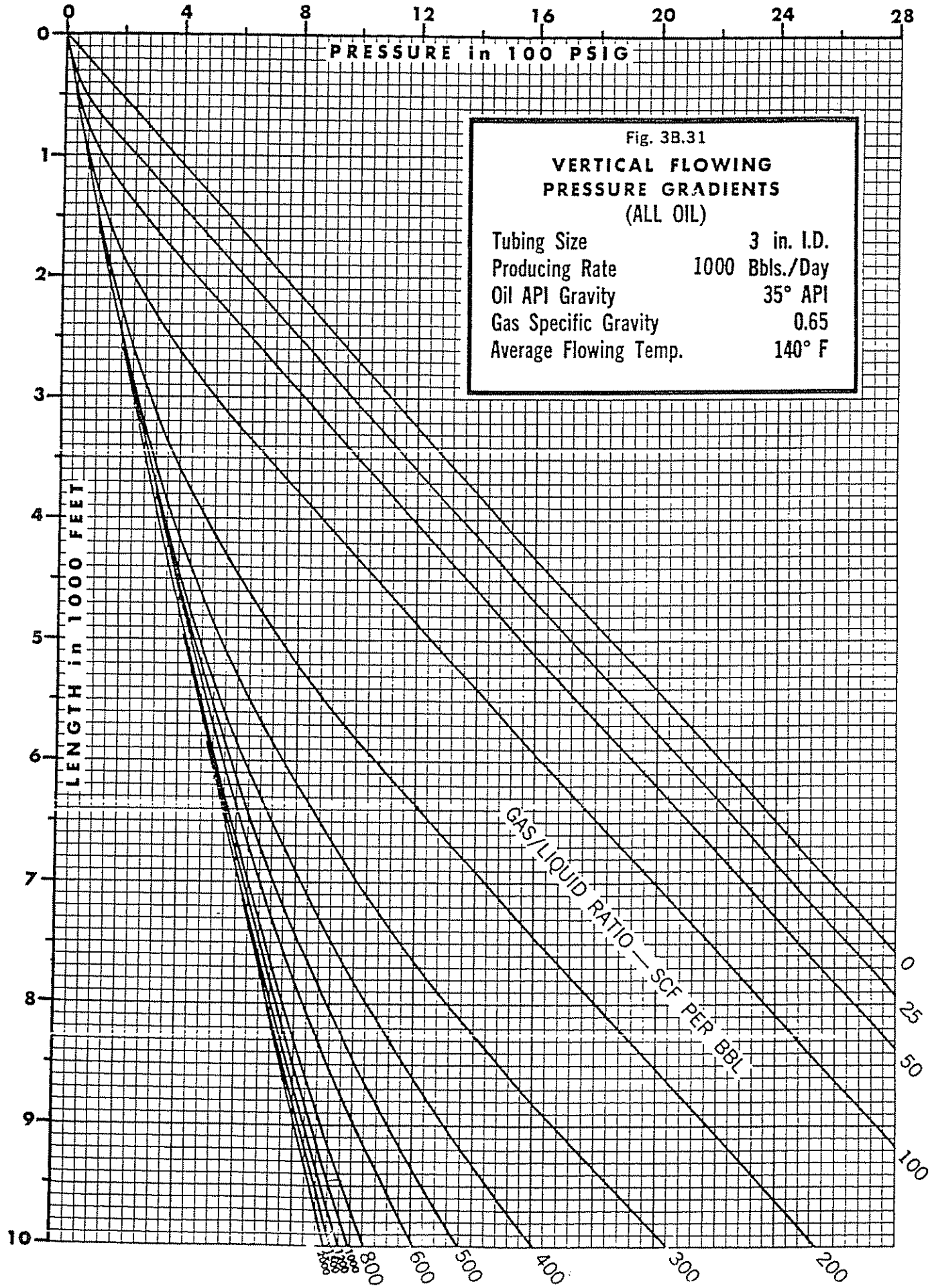
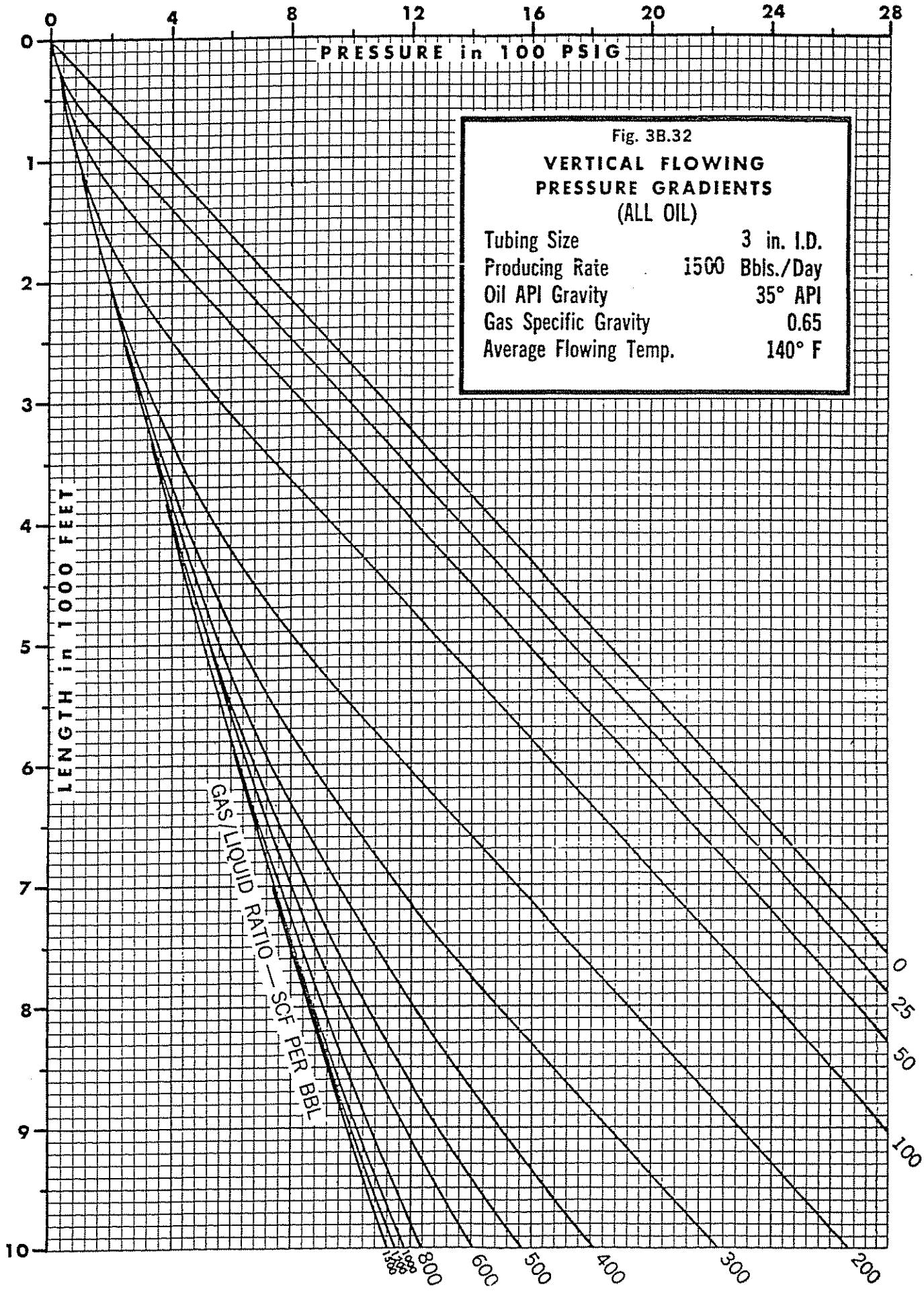


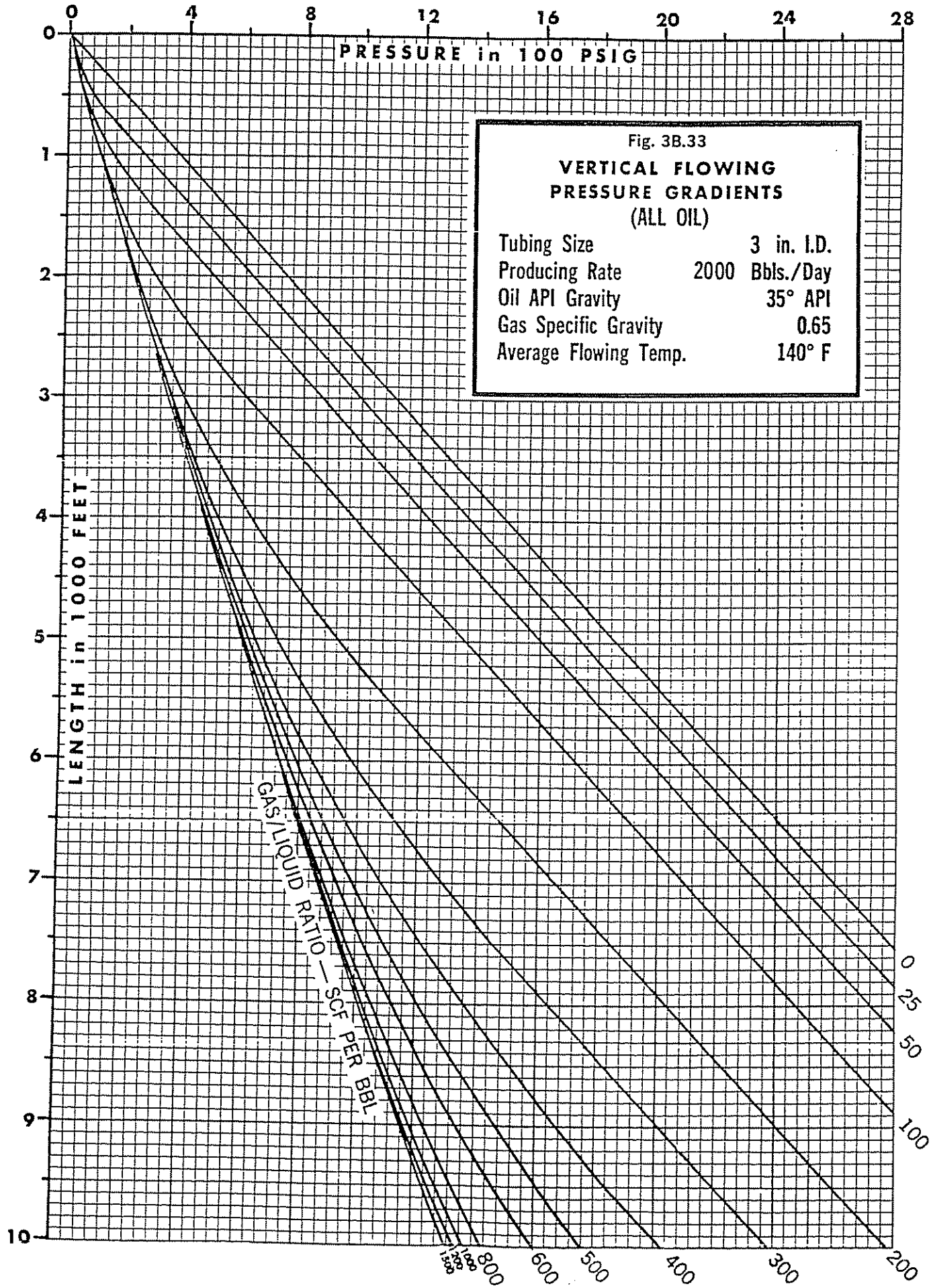
Fig. 3B.30

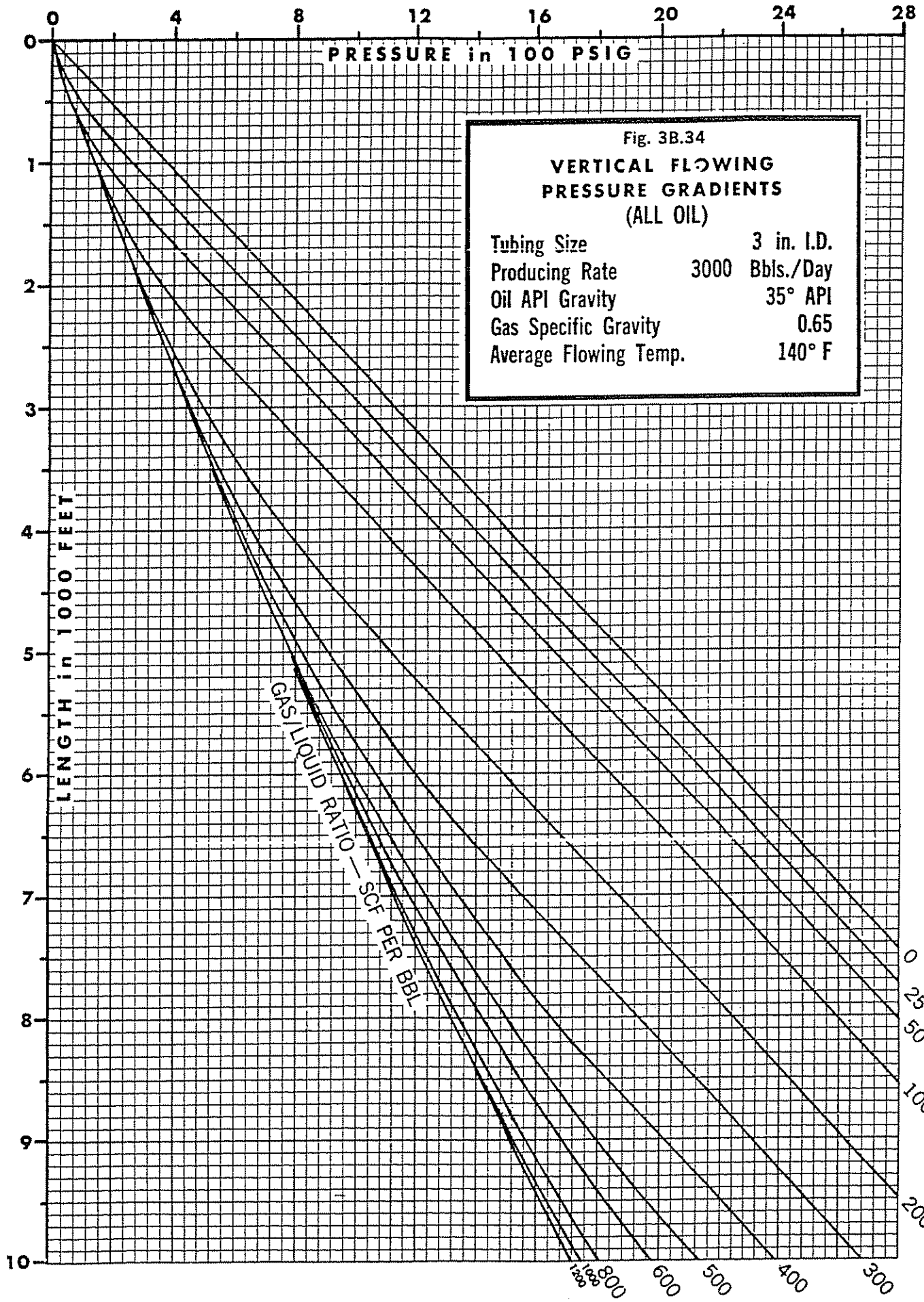
VERTICAL FLOWING PRESSURE GRADIENTS
(ALL WATER)

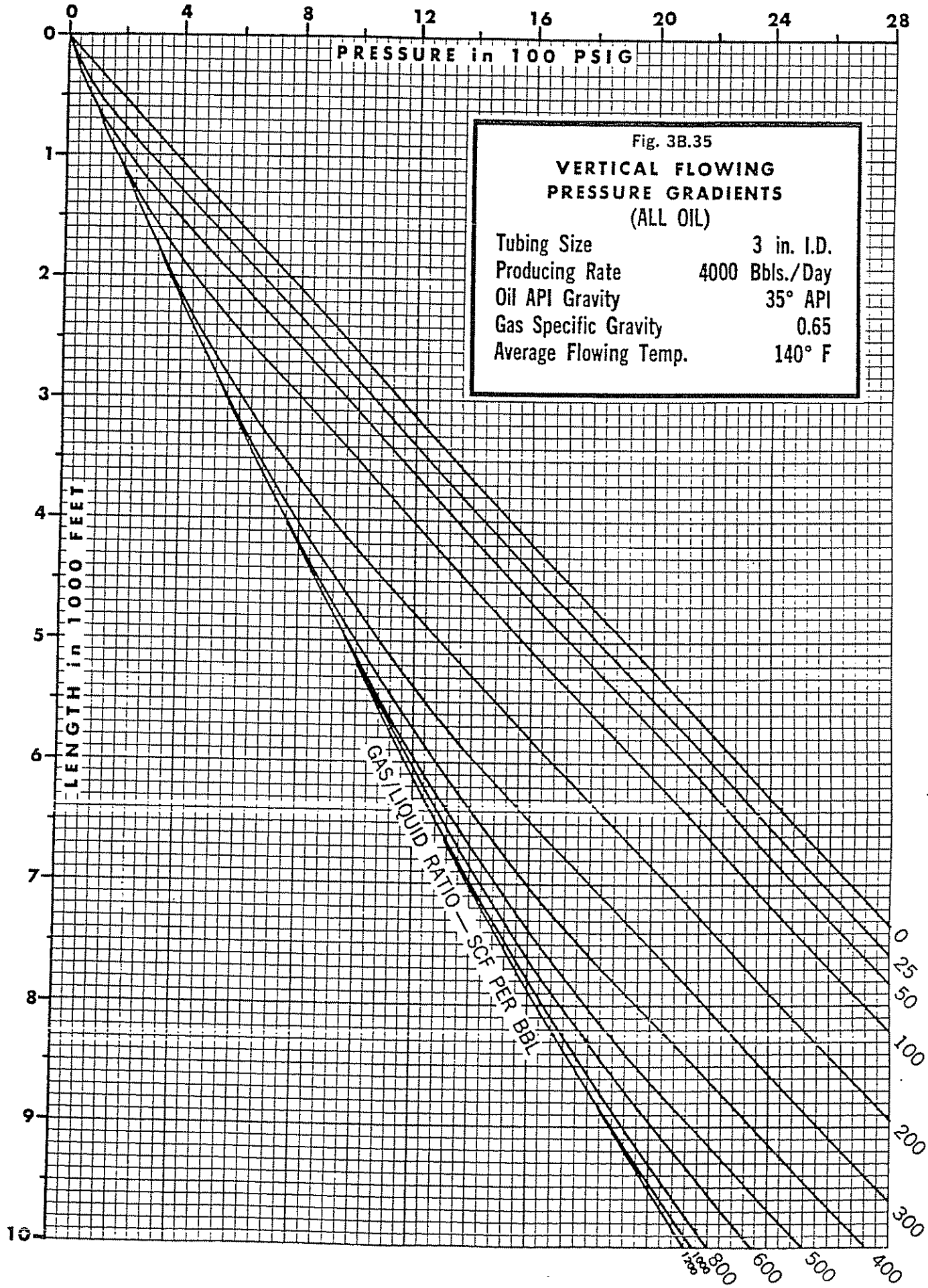
Tubing Size	2.5 in. I.D.
Producing Rate	3000 Bbls./Day
Water Specific Gravity	1.074
Gas Specific Gravity	0.65
Average Flowing Temp.	140° F

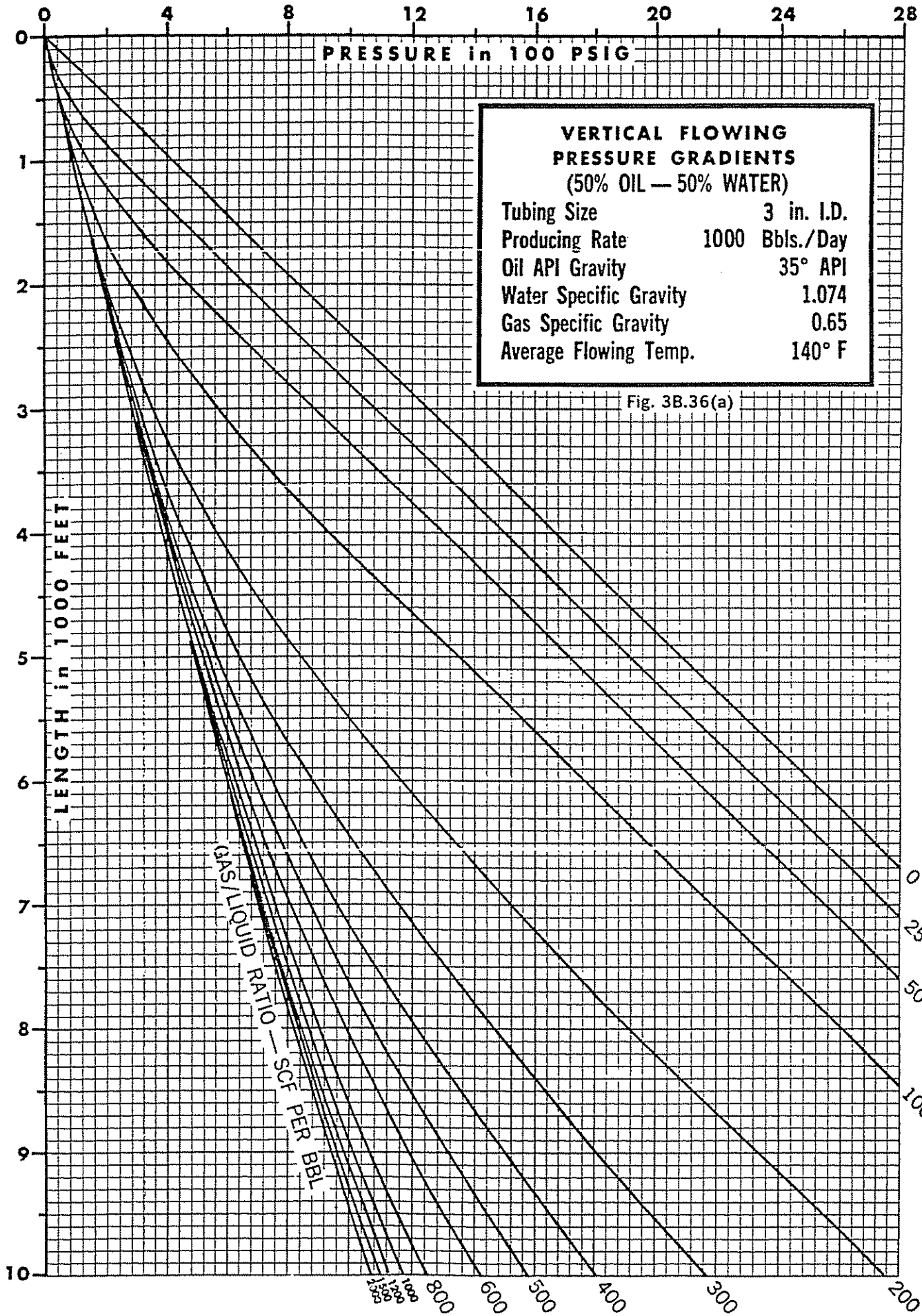

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.31
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL OIL)
Tubing Size 3 in. I.D.
Producing Rate 1000 Bbls./Day
Oil API Gravity 35° API
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800

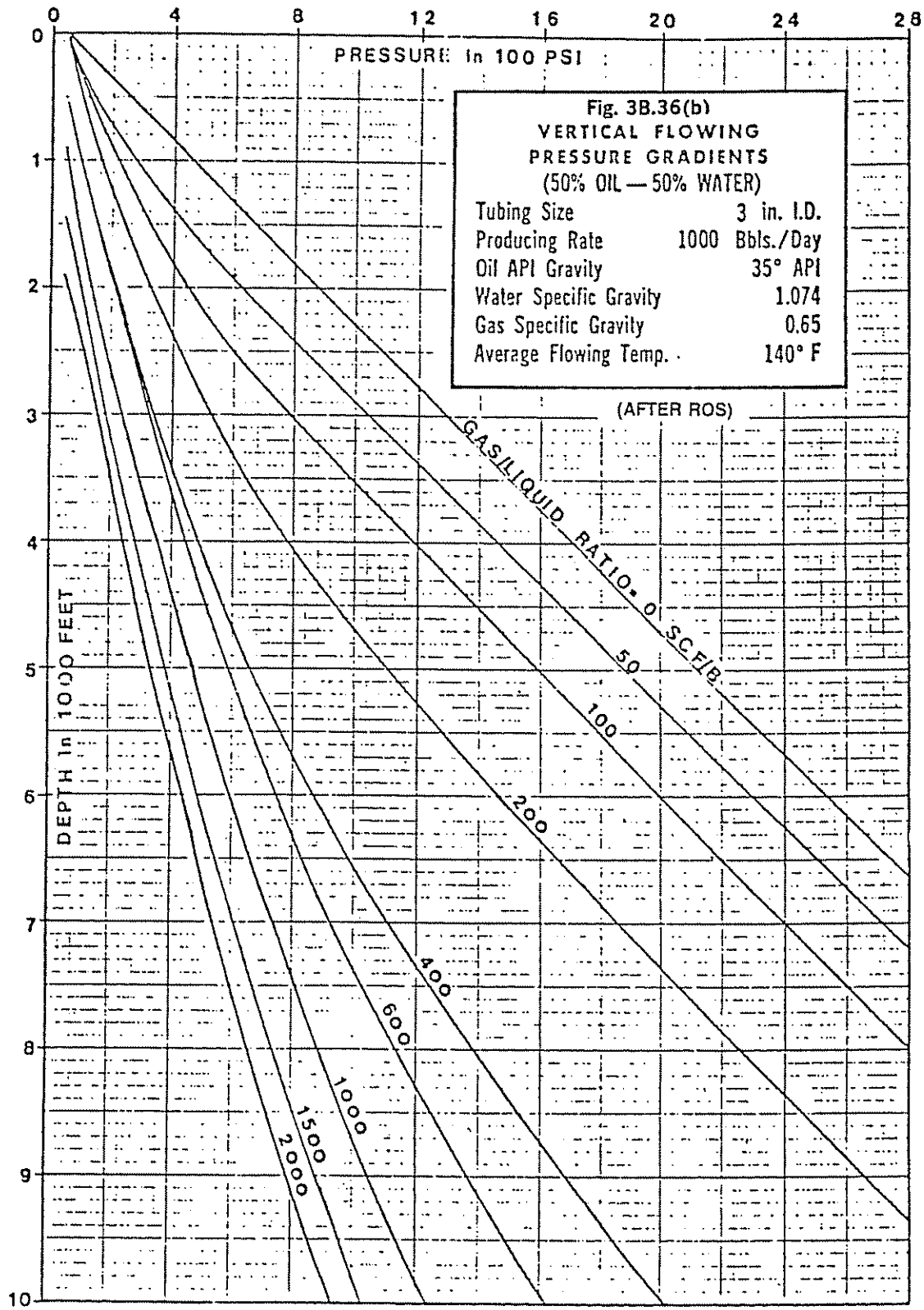

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.32
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL OIL)
Tubing Size 3 in. I.D.
Producing Rate 1500 Bbls./Day
Oil API Gravity 35° API
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
1500
1000
800
600
500
400
300
200
0
25
50
100

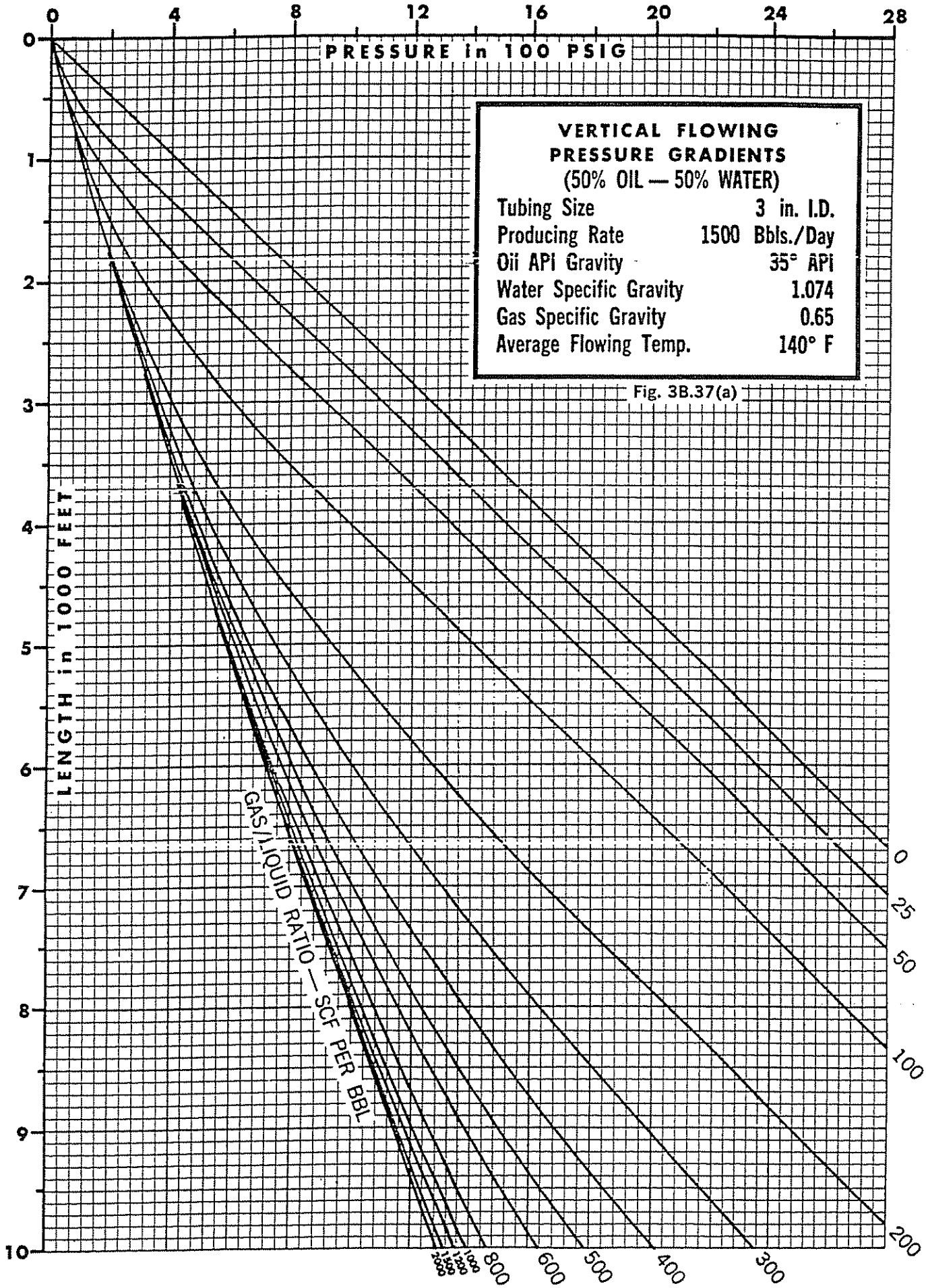

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.33
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL OIL)
Tubing Size 3 in. I.D.
Producing Rate 2000 Bbls./Day
Oil API Gravity 35° API
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000
1200
1500

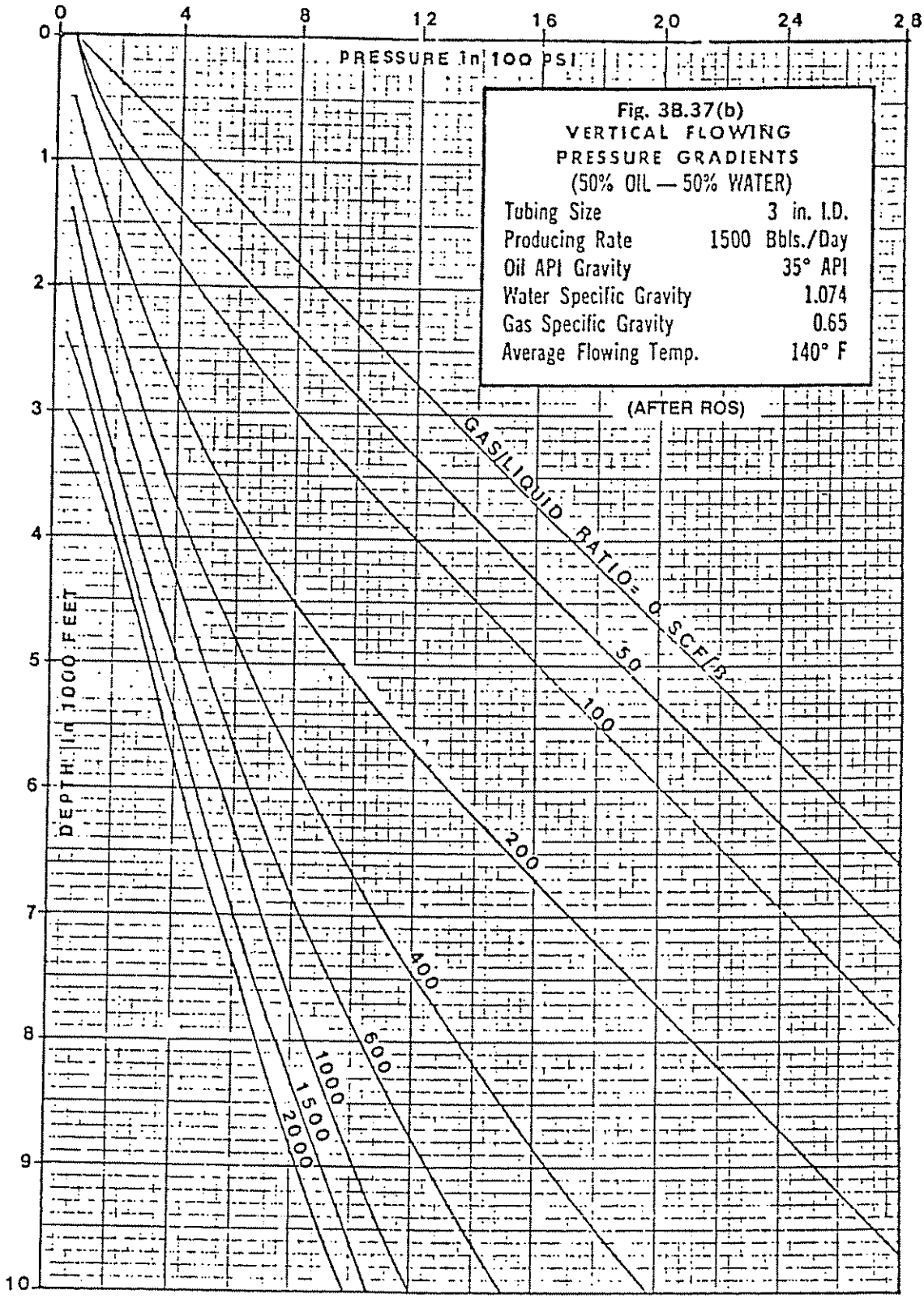

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.34
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL OIL)
Tubing Size 3 in. I.D.
Producing Rate 3000 Bbls./Day
Oil API Gravity 35° API
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000

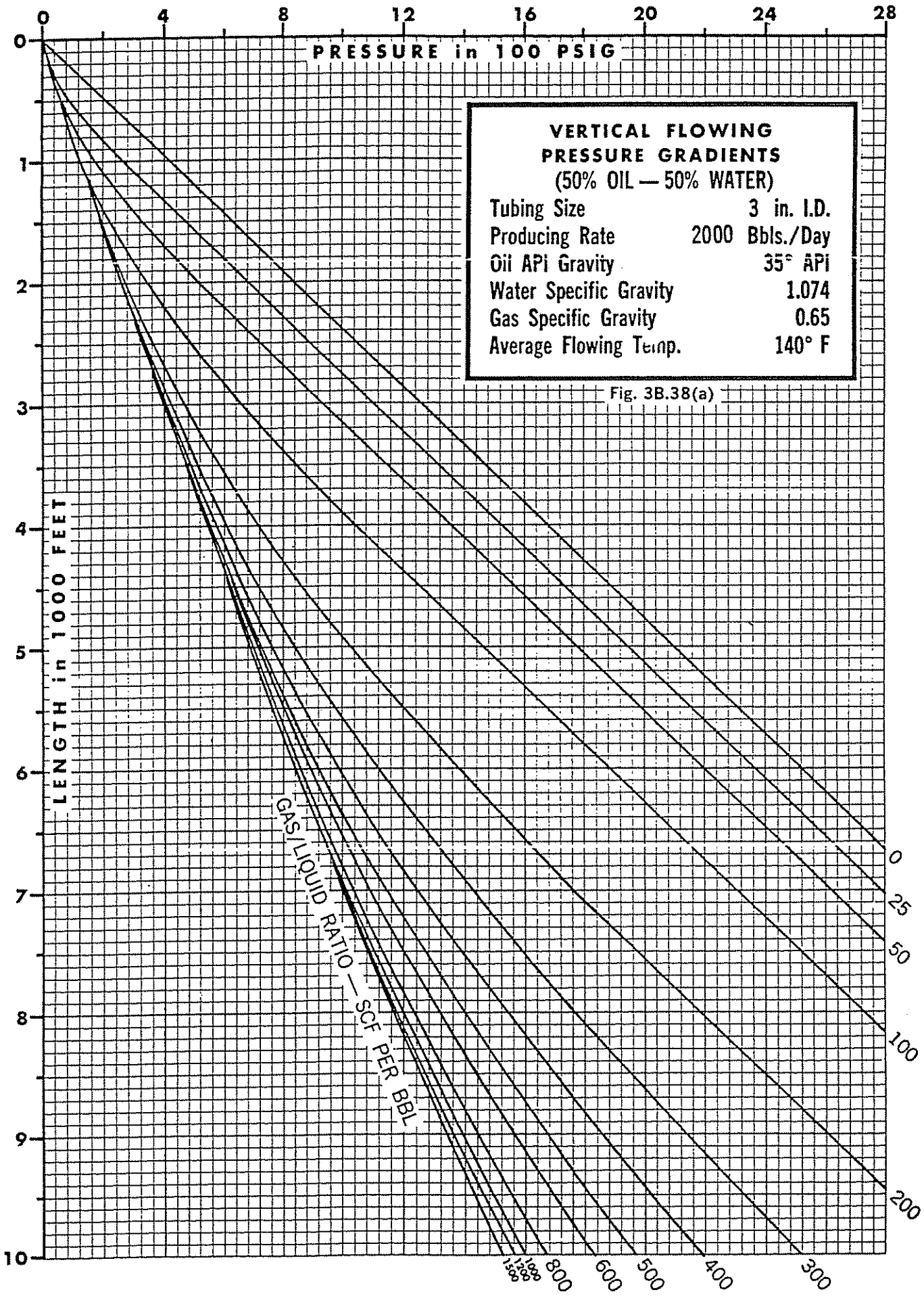

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.35
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL OIL)
Tubing Size 3 in. I.D.
Producing Rate 4000 Bbls./Day
Oil API Gravity 35° API
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000

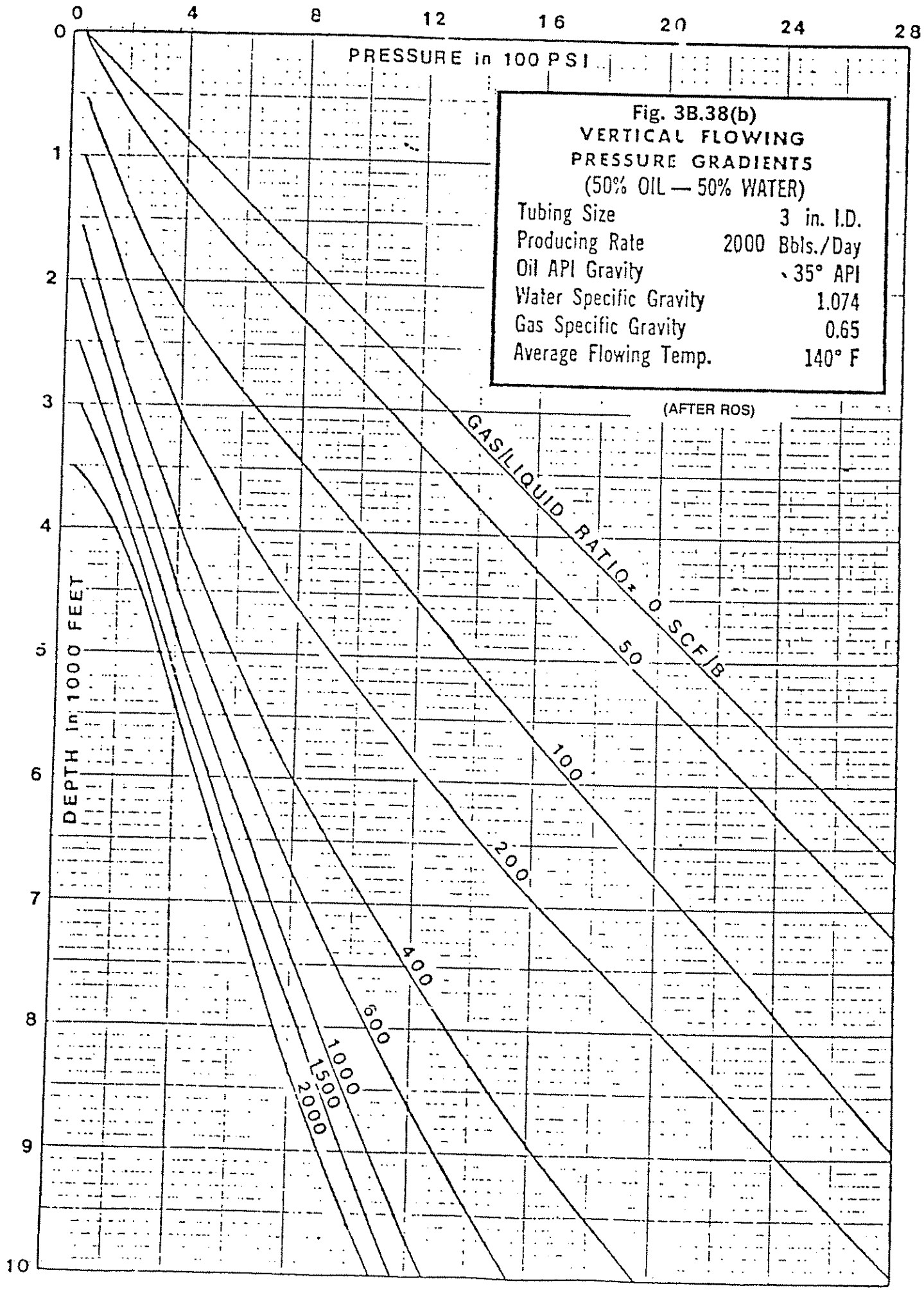

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 3 in. I.D.
Producing Rate 1000 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
Fig. 3B.36(a)
LENGTH in 1000 FEET
0
1
2
3
4
5
6
7
8
9
10
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800

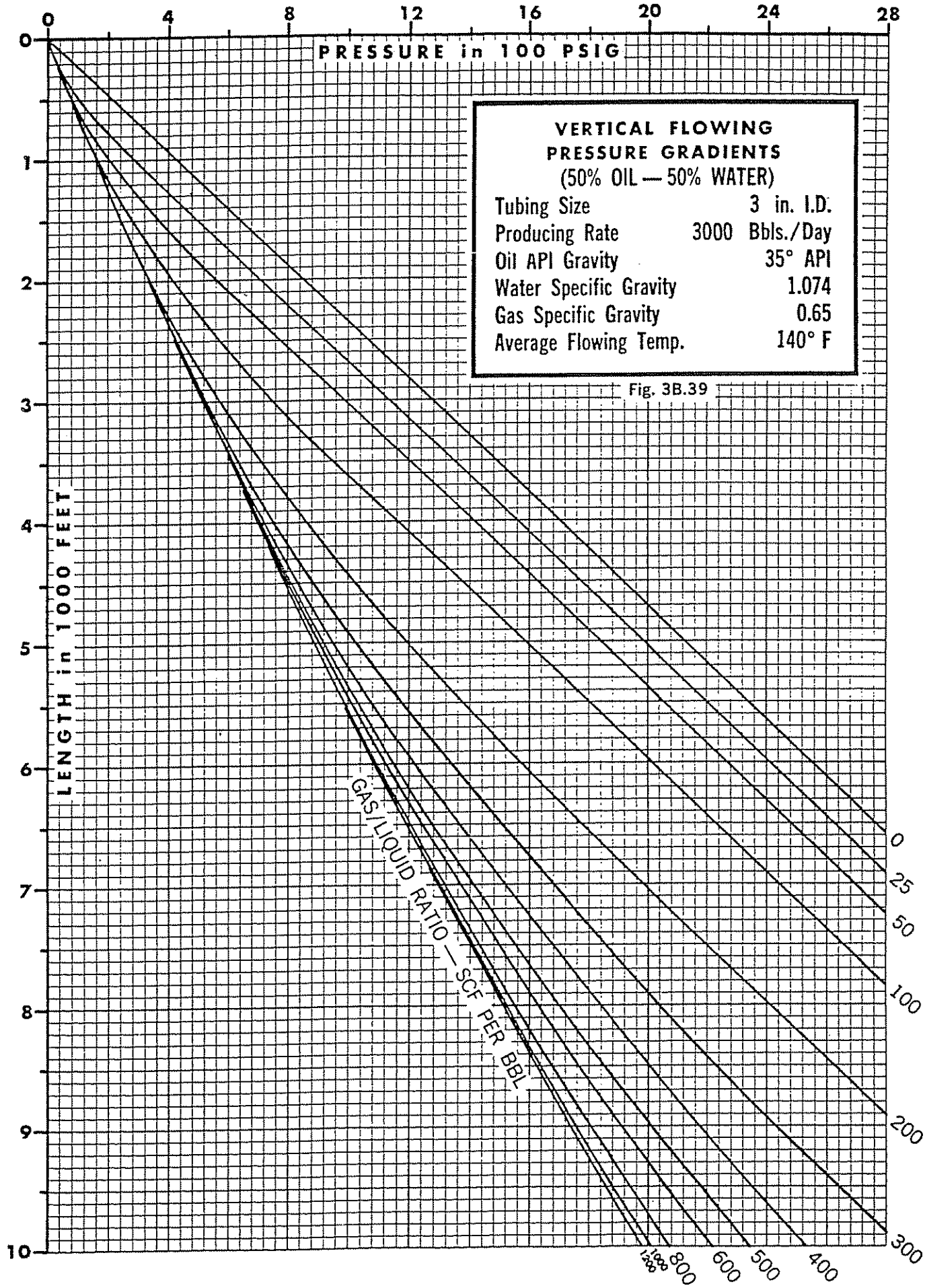

PRESSURE In 100 PSI
0
4
8
12
16
20
24
28
DEPTH In 1000 FEET
0
1
2
3
4
5
6
7
8
9
10
Fig. 3B.36(b)
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 3 in. I.D.
Producing Rate 1000 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
(AFTER ROS)
GAS/LIQUID RATIO - 0 SCF/B
50
100
200
400
600
1000
1500
2000

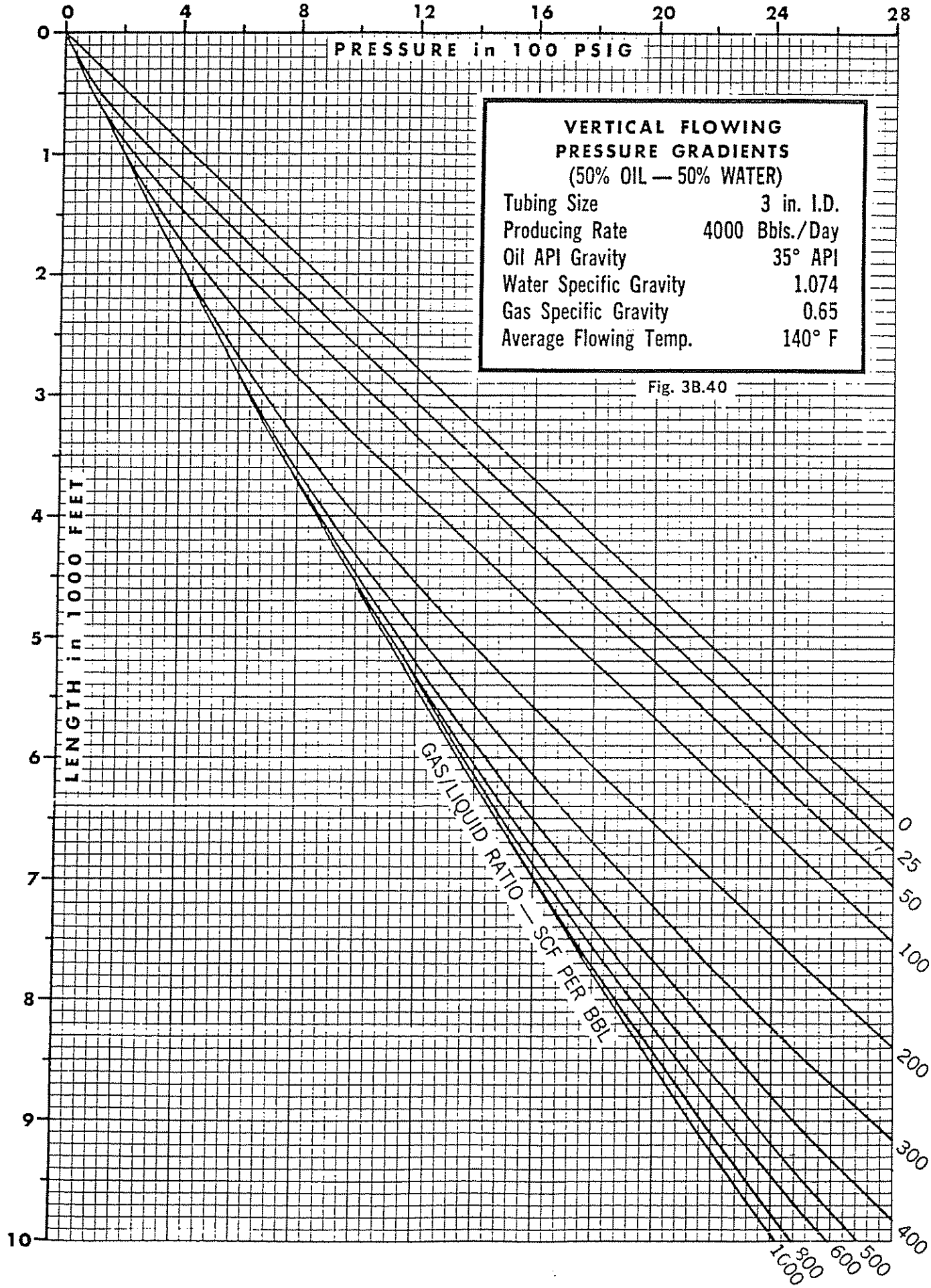

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 3 in. I.D.
Producing Rate 1500 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
Fig. 3B.37(a)
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800

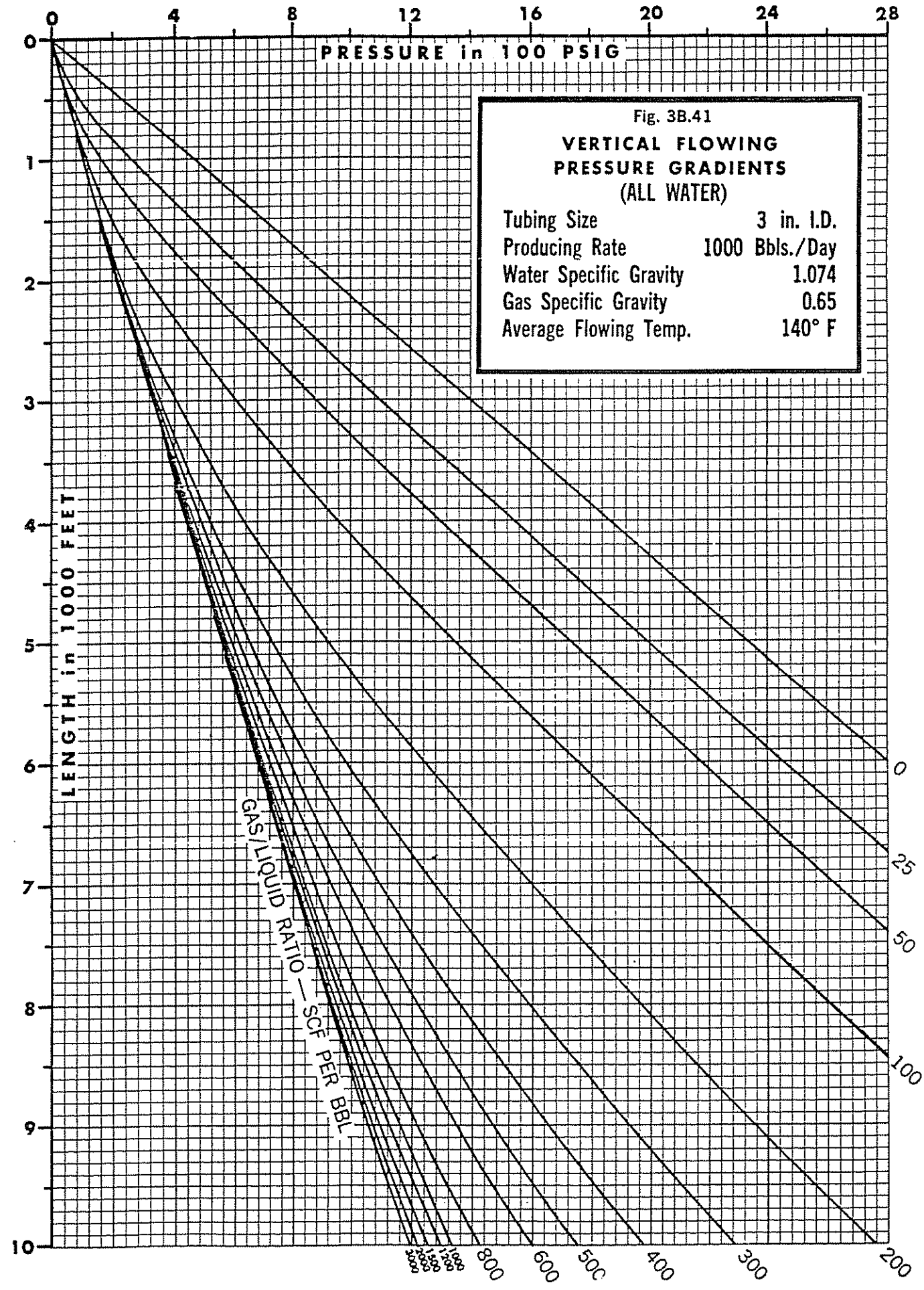

0
4
8
12
16
20
24
28
PRESSURE In 100 PSI
DEPTH In 1000 FEET
0
1
2
3
4
5
6
7
8
9
10
Fig. 3B.37(b)
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 3 in. I.D.
Producing Rate 1500 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
(AFTER ROS)
GAS/LIQUID RATIO = 0 SCF/B
50
100
200
400
600
1000
1500
2000

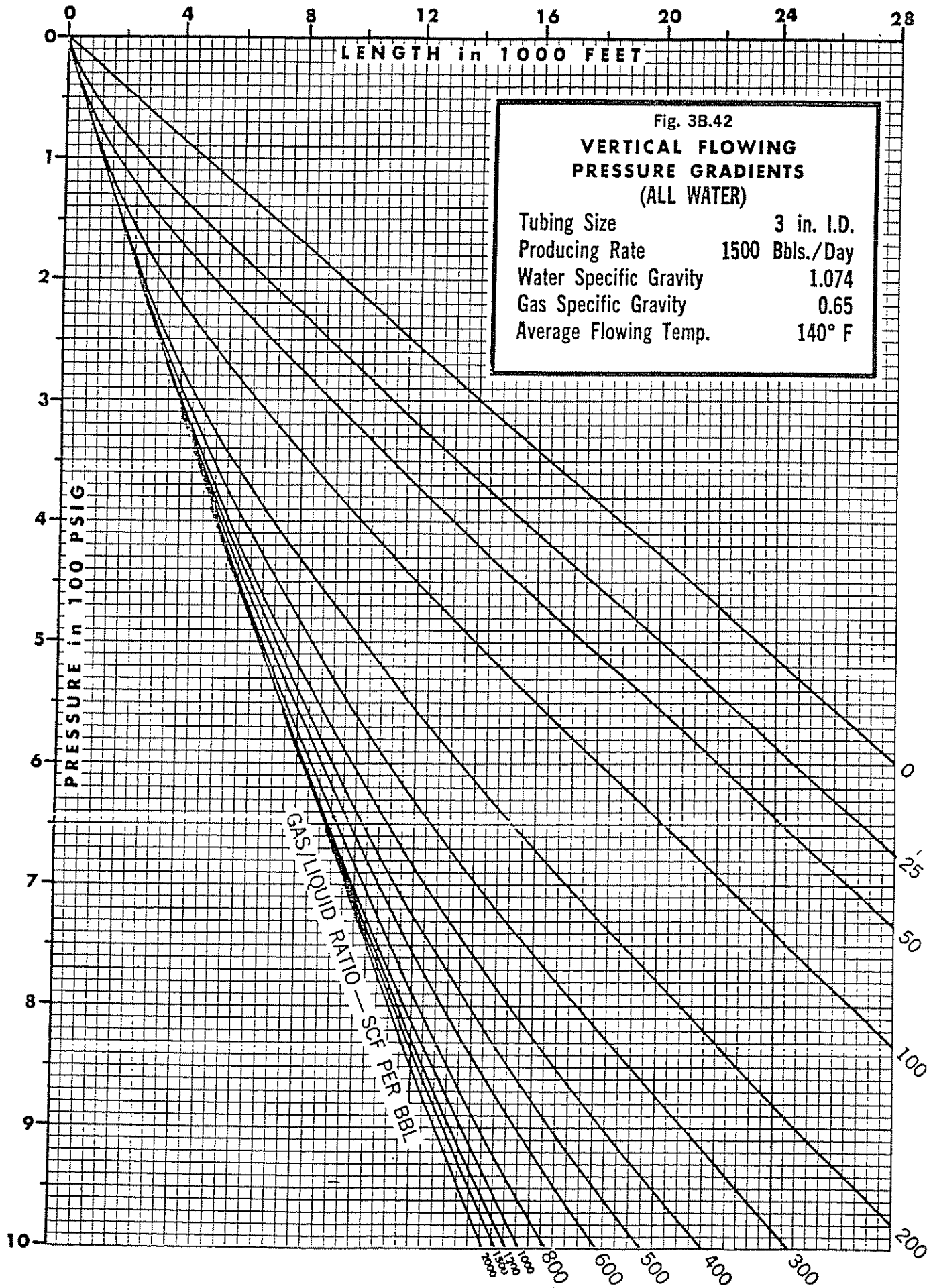

PRESSURE in 100 PSIG
0
4
8
12
16
20
24
28
LENGTH in 1000 FEET
0
1
2
3
4
5
6
7
8
9
10
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 3 in. I.D.
Producing Rate 2000 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
Fig. 3B.38(a)
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000
1200
1500

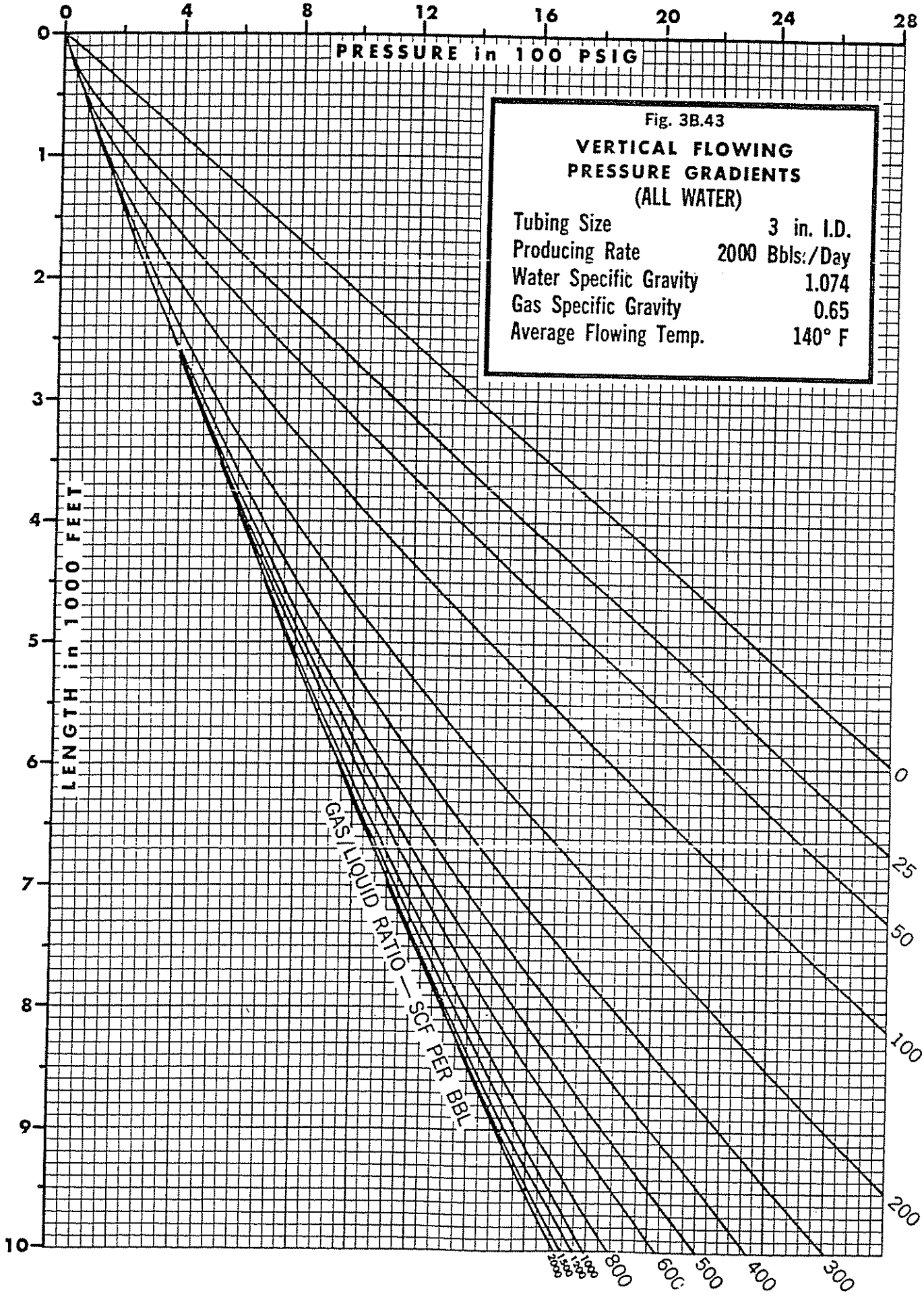

0
4
8
12
16
20
24
28
PRESSURE in 100 PSI
0
1
2
3
4
5
6
7
8
9
10
DEPTH in 1000 FEET
Fig. 3B.38(b)
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 3 in. I.D.
Producing Rate 2000 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
(AFTER ROS)
GAS/LIQUID RATIO = 0 SCF/B
50
100
200
400
600
1000
1500
2000

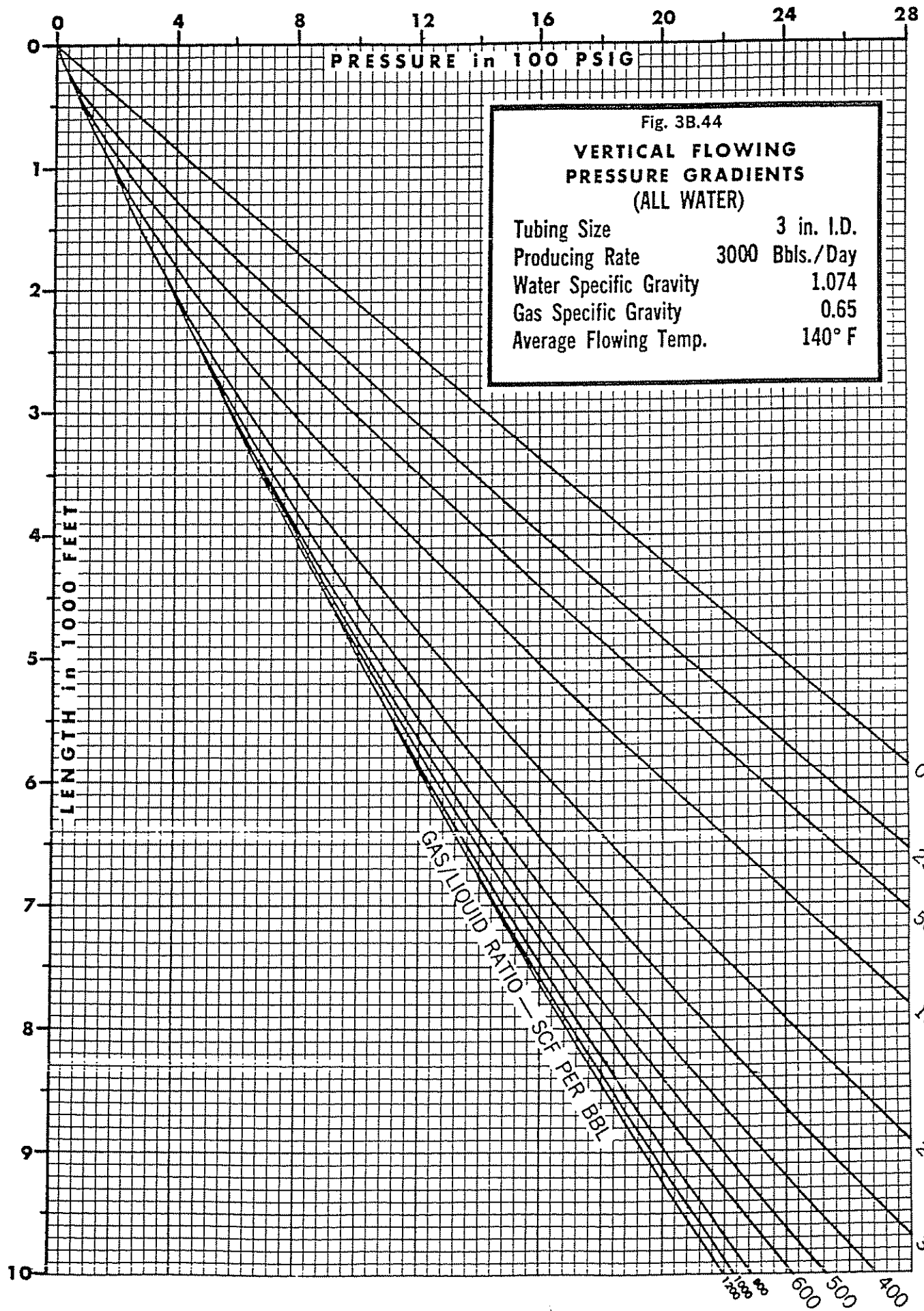

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 3 in. I.D.
Producing Rate 3000 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
Fig. 3B.39
LENGTH in 1000 FEET
0
1
2
3
4
5
6
7
8
9
10
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000
1200

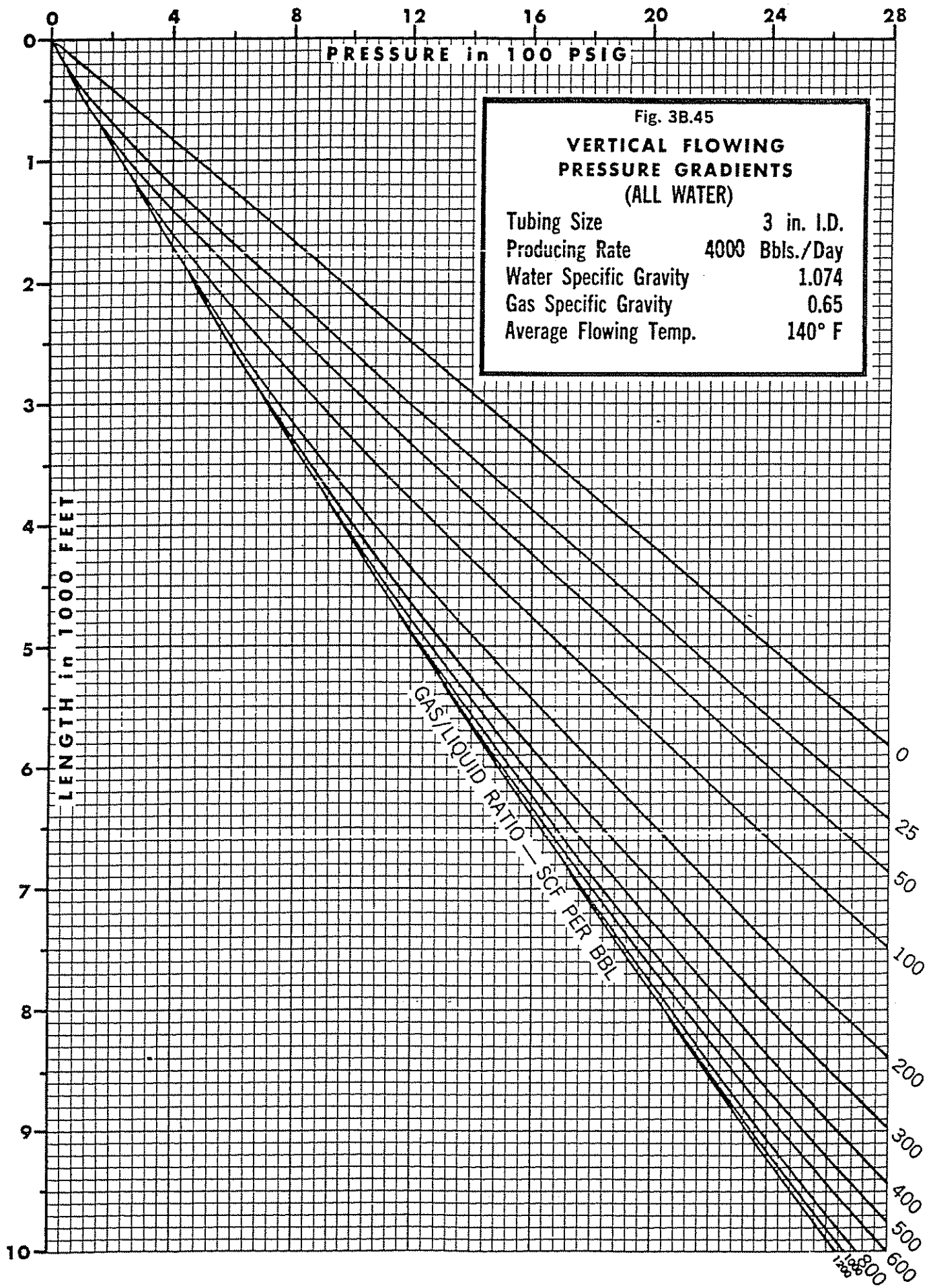

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 3 in. I.D.
Producing Rate 4000 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
Fig. 3B.40
LENGTH in 1000 FEET
0
1
2
3
4
5
6
7
8
9
10
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000

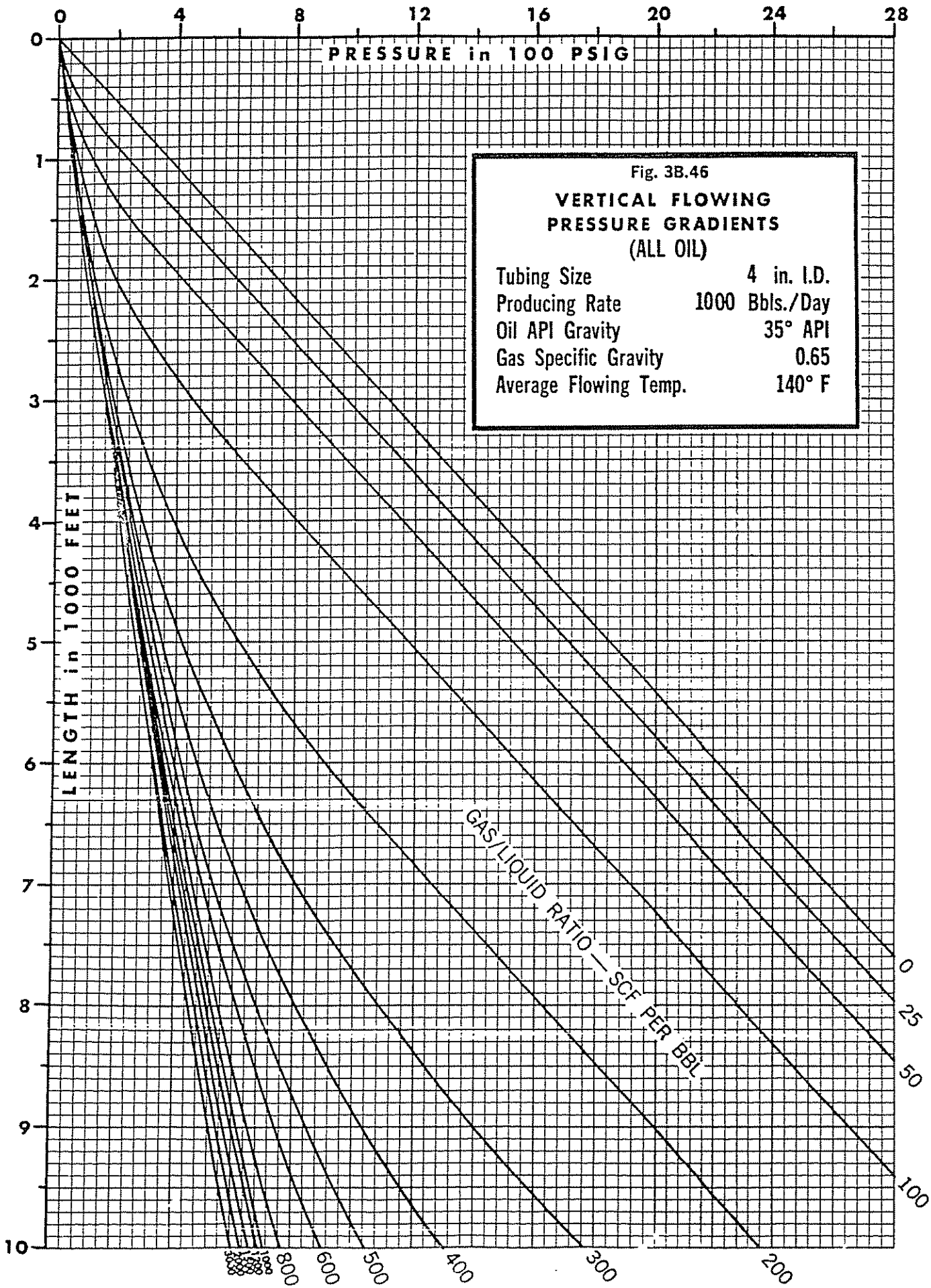

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.41
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Tubing Size 3 in. I.D.
Producing Rate 1000 Bbls./Day
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800

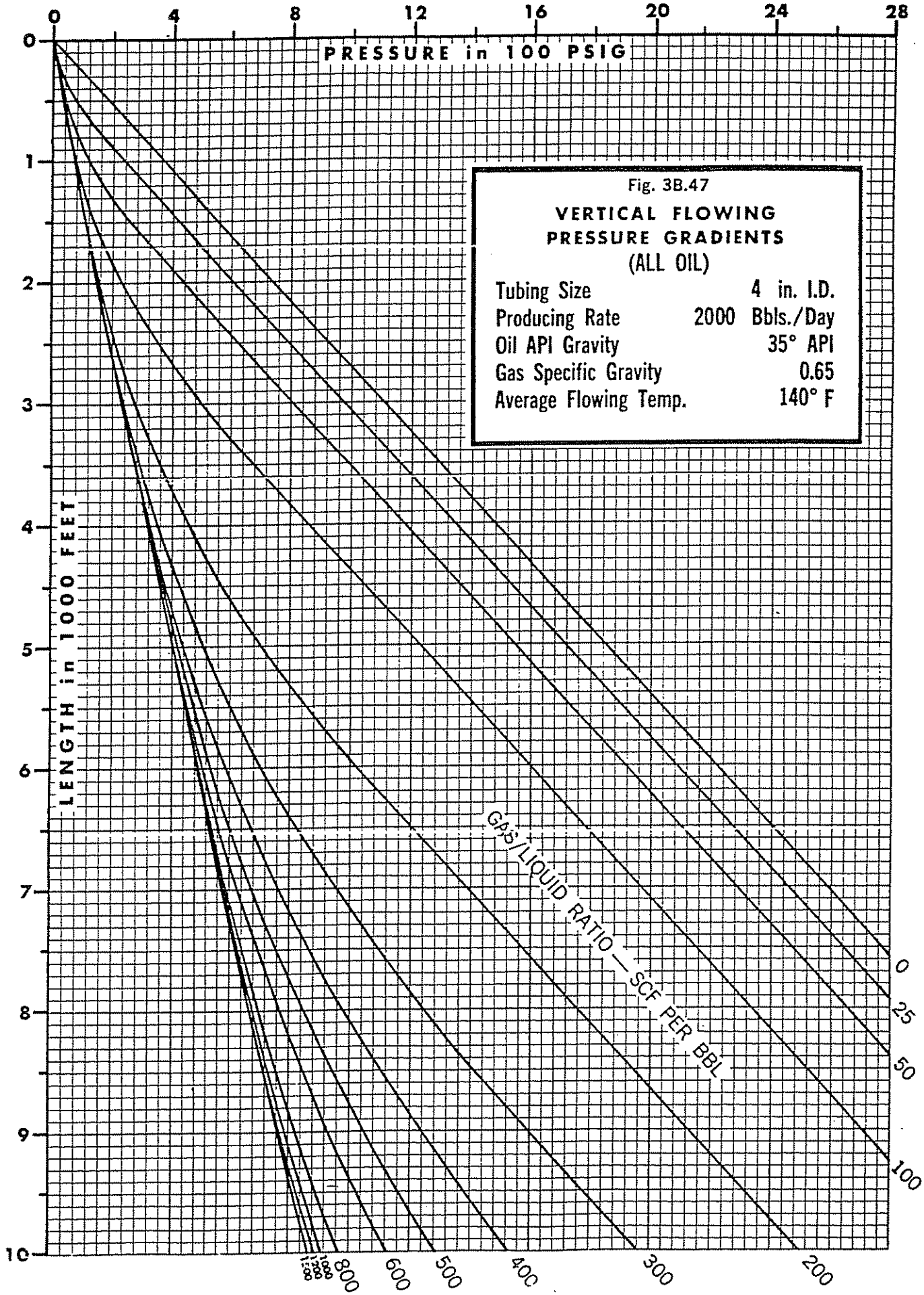

0
4
8
12
16
20
24
28
LENGTH in 1000 FEET
0
1
2
3
4
5
6
7
8
9
10
PRESSURE in 100 PSIG
Fig. 3B.42
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Tubing Size 3 in. I.D.
Producing Rate 1500 Bbls./Day
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800

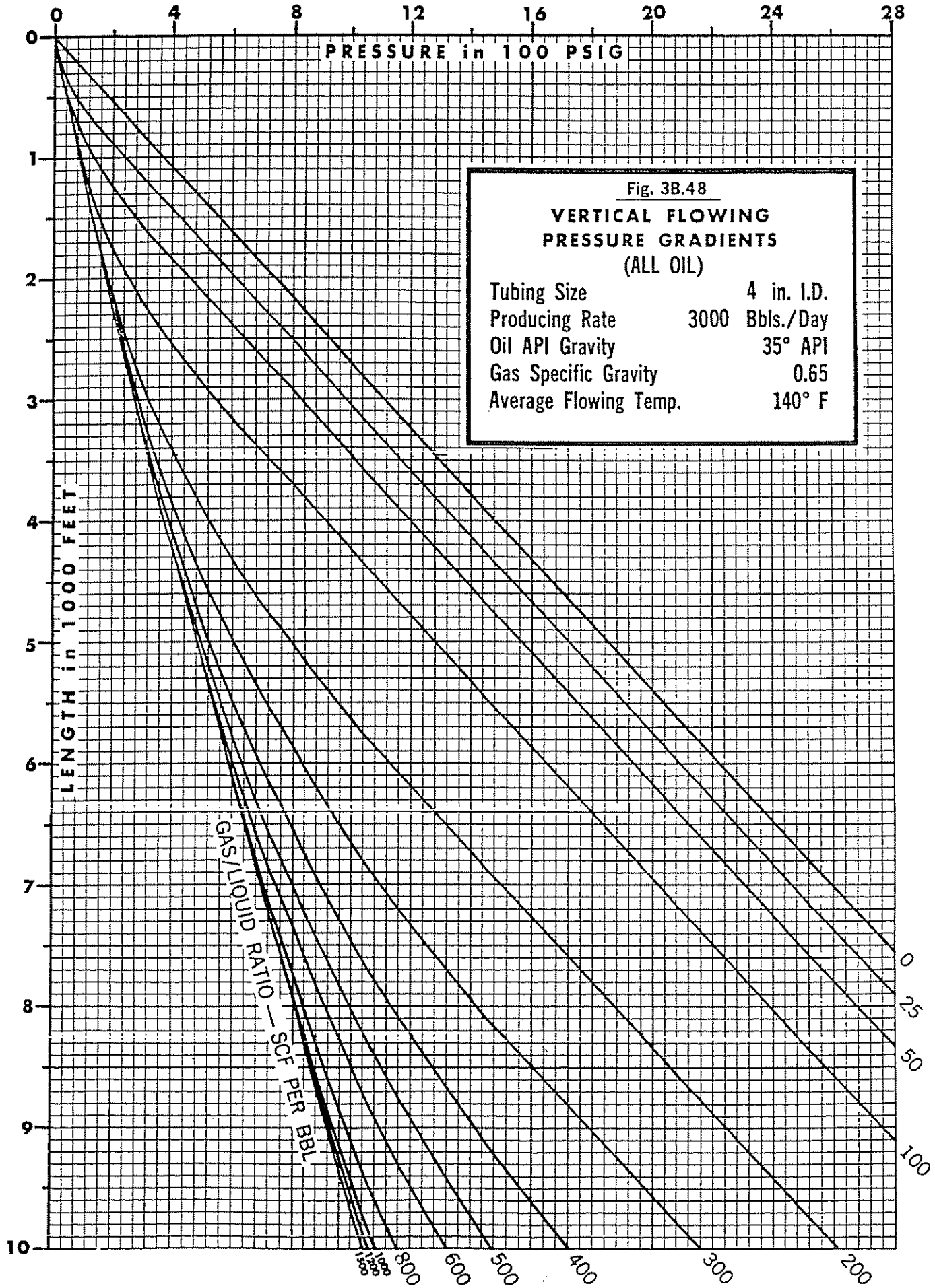

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.43
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Tubing Size 3 in. I.D.
Producing Rate 2000 Bbls./Day
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800


0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.44
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Tubing Size 3 in. I.D.
Producing Rate 3000 Bbls./Day
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1,000
1,200


0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.45
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Tubing Size 3 in. I.D.
Producing Rate 4000 Bbls./Day
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000
1200


0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.46
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL OIL)
Tubing Size 4 in. I.D.
Producing Rate 1000 Bbls./Day
Oil API Gravity 35° API
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800


0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.47
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL OIL)
Tubing Size 4 in. I.D.
Producing Rate 2000 Bbls./Day
Oil API Gravity 35° API
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800


0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.48
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL OIL)
Tubing Size 4 in. I.D.
Producing Rate 3000 Bbls./Day
Oil API Gravity 35° API
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000
1500
2000

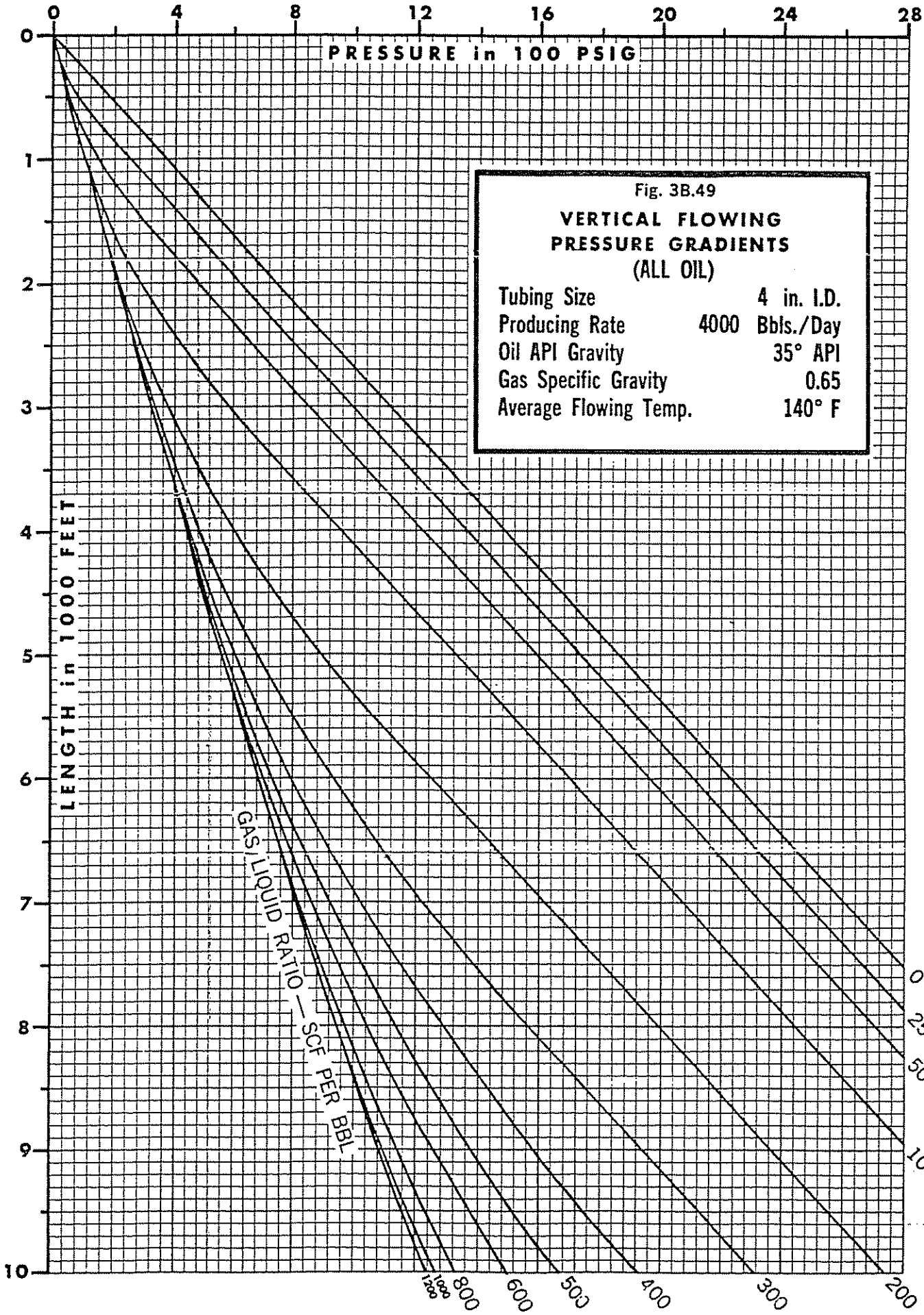


Fig. 3B.49

VERTICAL FLOWING PRESSURE GRADIENTS
(ALL OIL)

Tubing Size	4 in. I.D.
Producing Rate	4000 Bbls./Day
Oil API Gravity	35° API
Gas Specific Gravity	0.65
Average Flowing Temp.	140° F

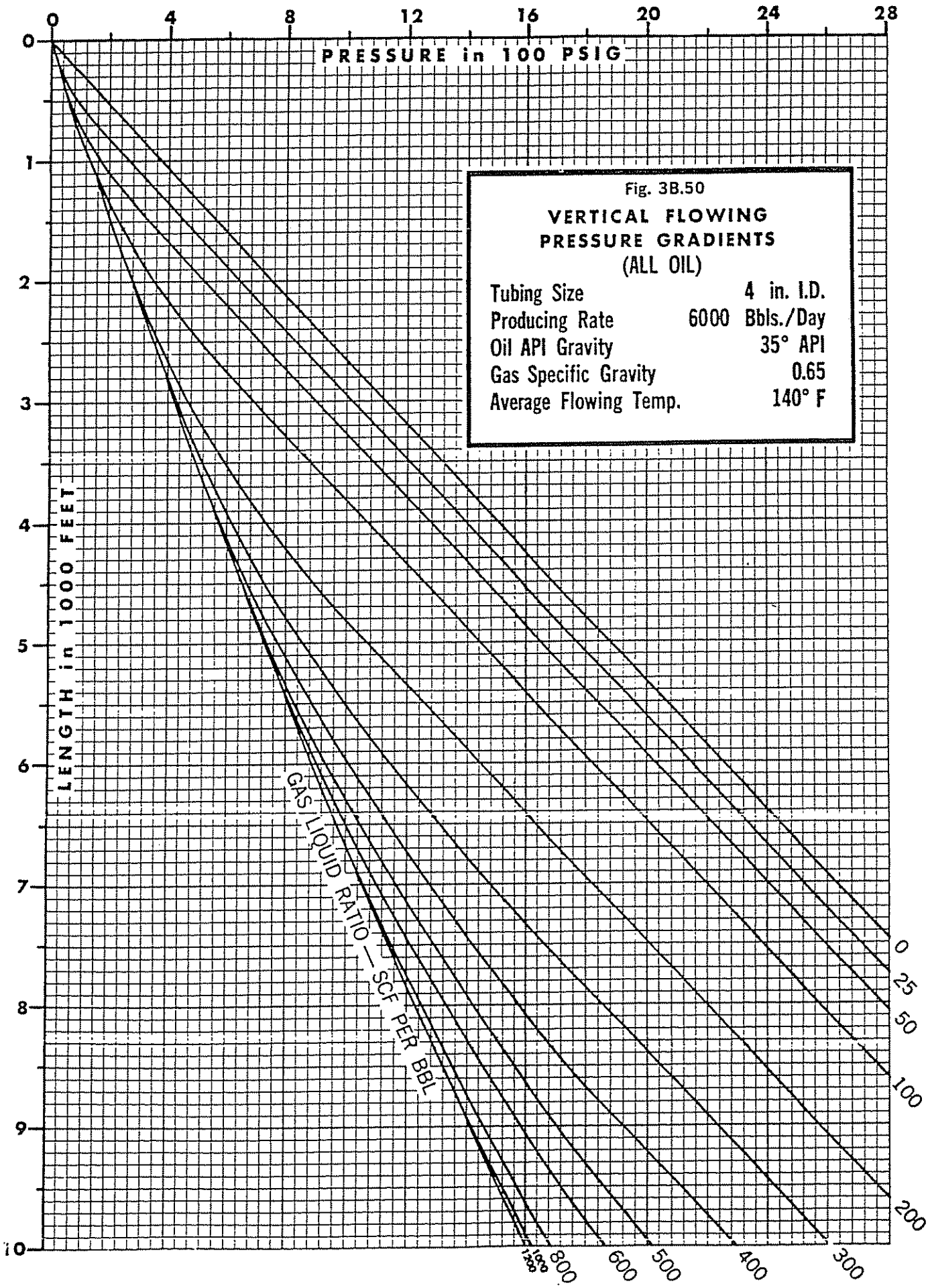


Fig. 3B.50

VERTICAL FLOWING PRESSURE GRADIENTS

(ALL OIL)

Tubing Size	4 in. I.D.
Producing Rate	6000 Bbls./Day
Oil API Gravity	35° API
Gas Specific Gravity	0.65
Average Flowing Temp.	140° F

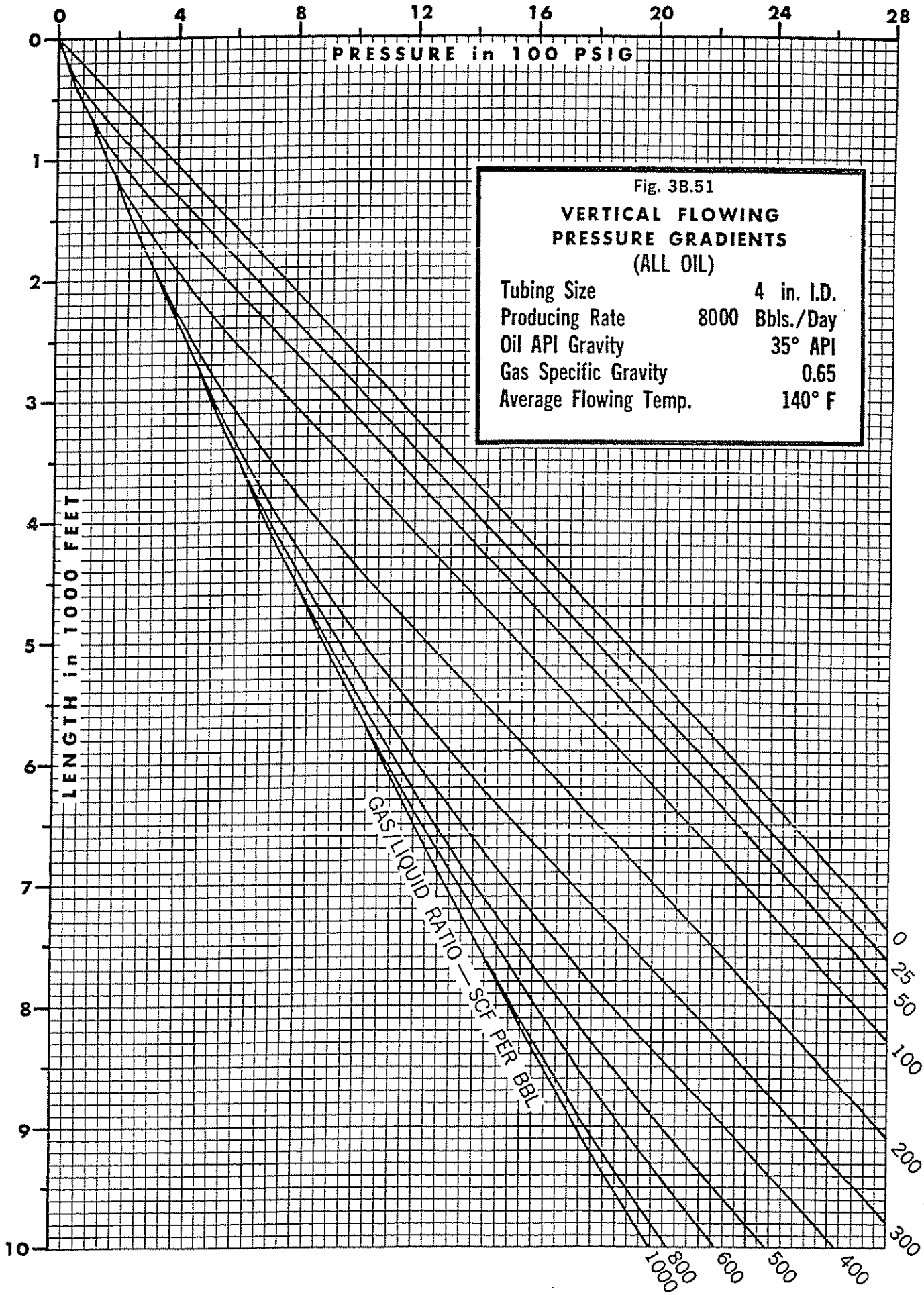
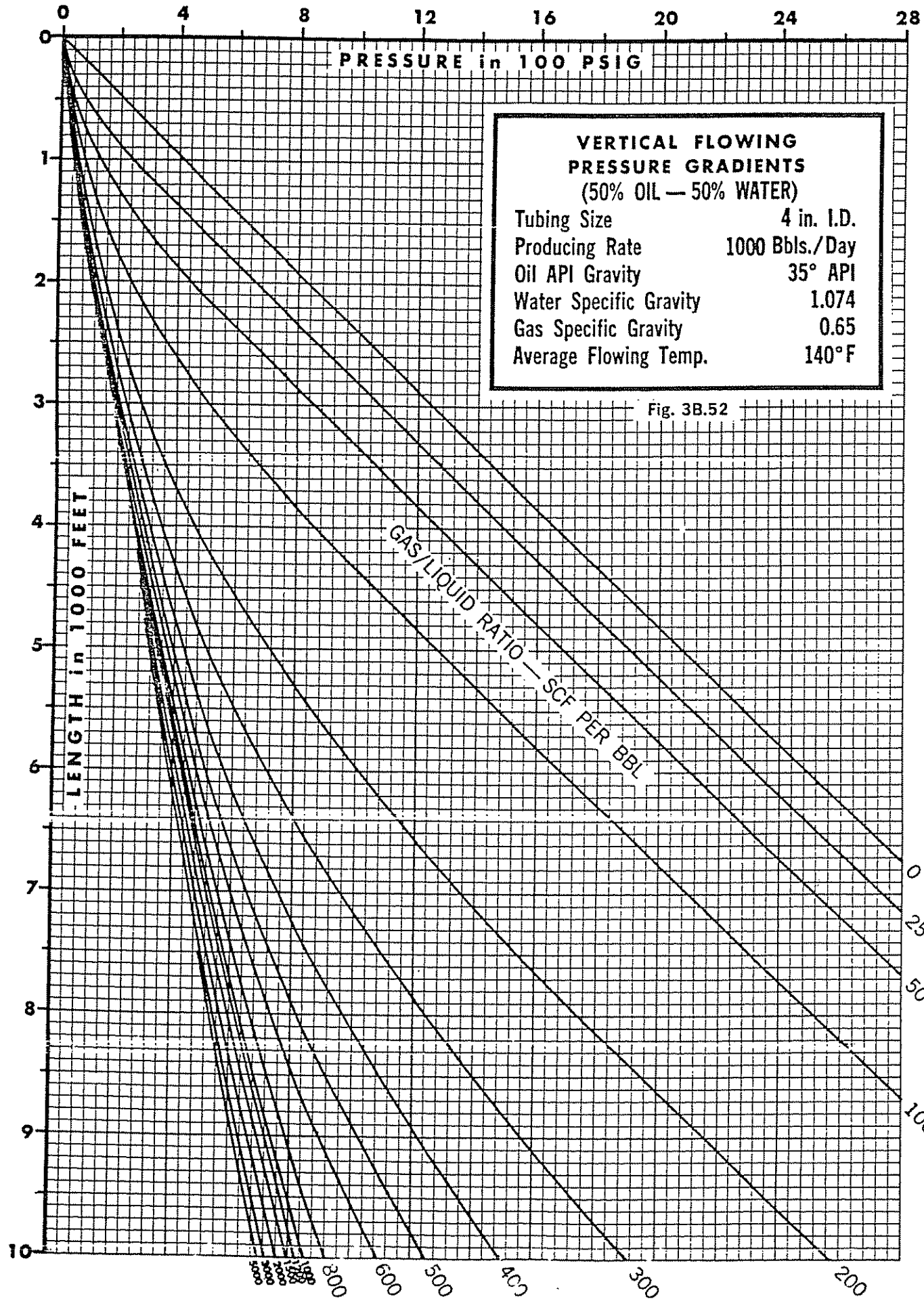


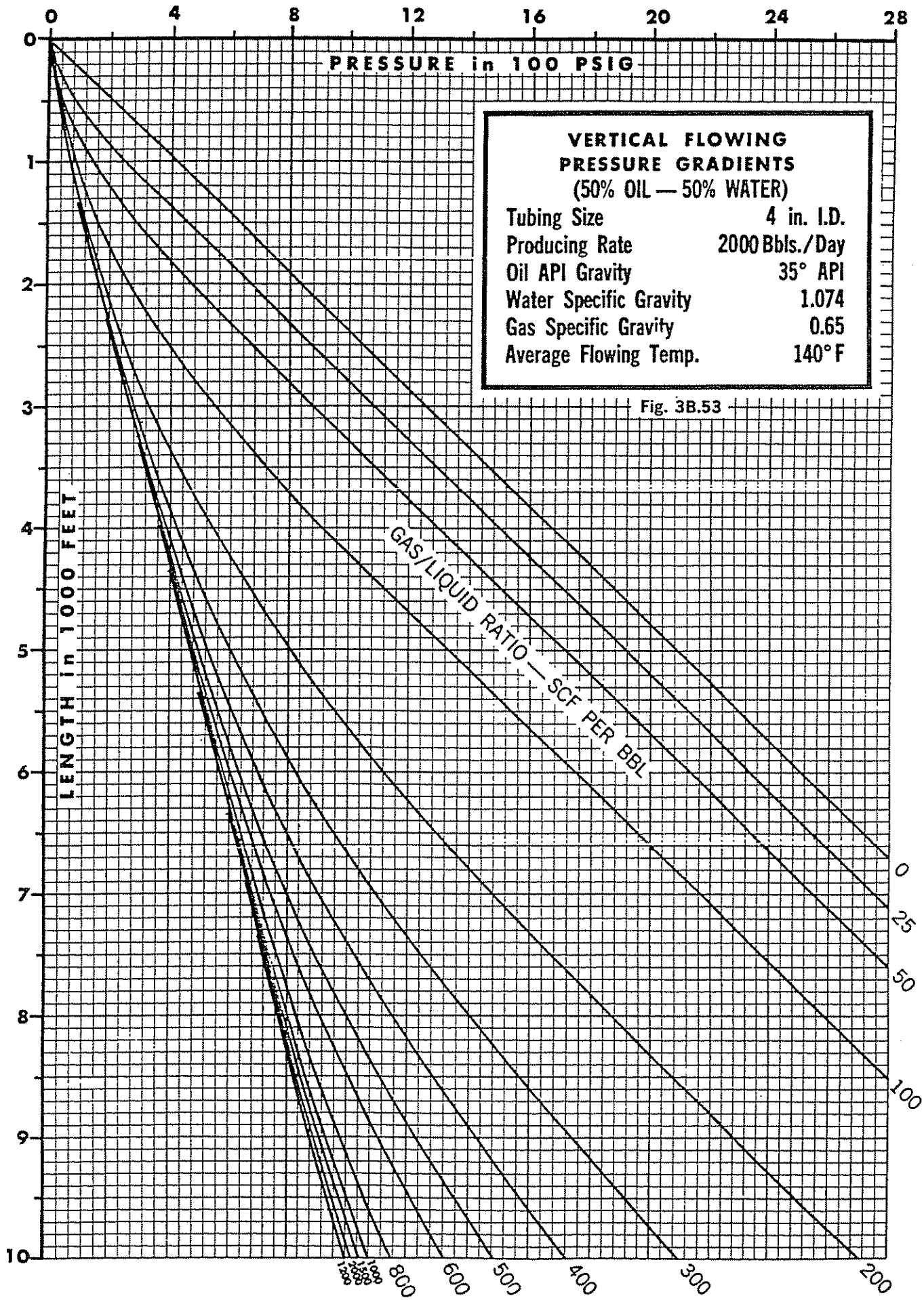
Fig. 3B.51

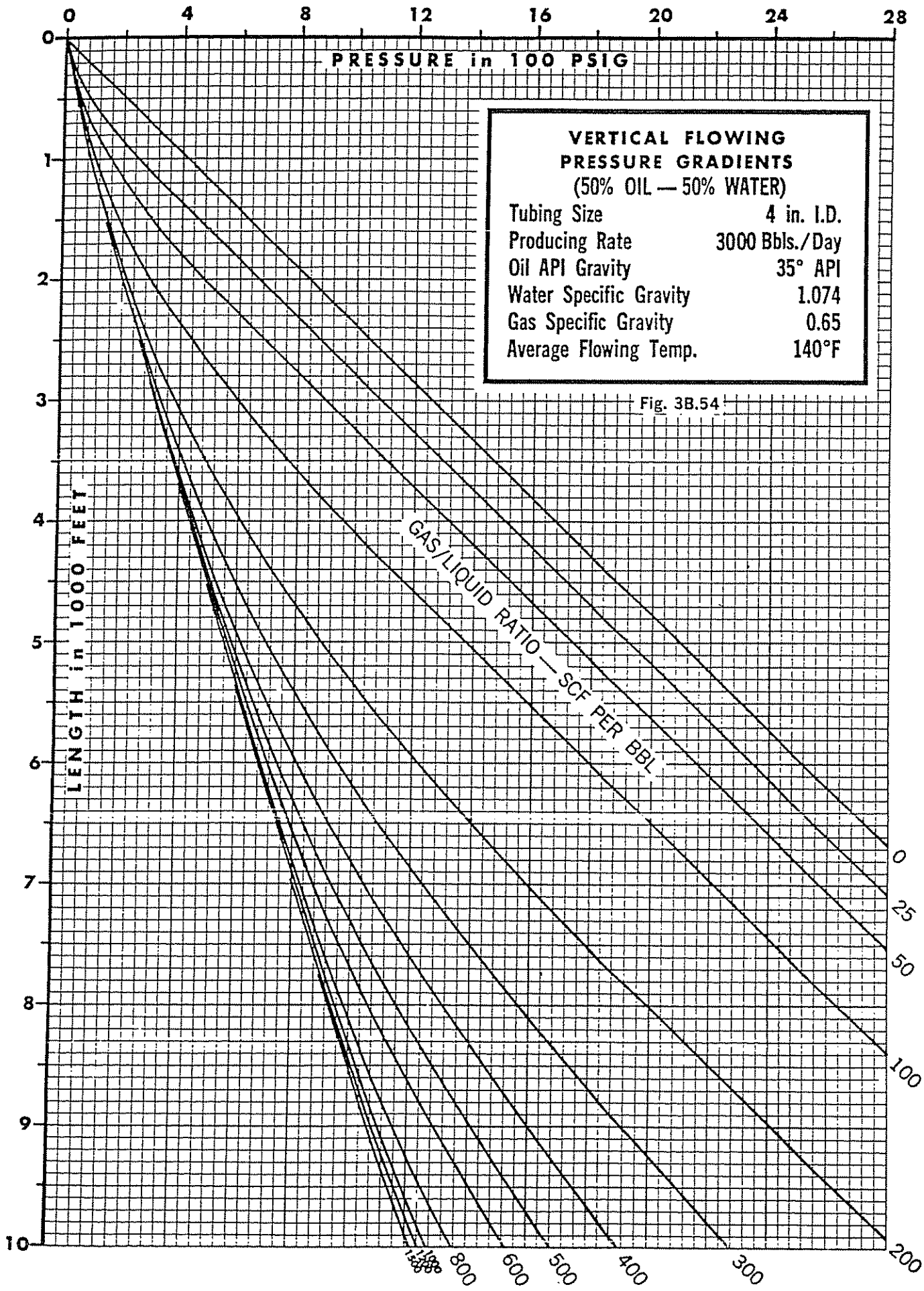
VERTICAL FLOWING PRESSURE GRADIENTS

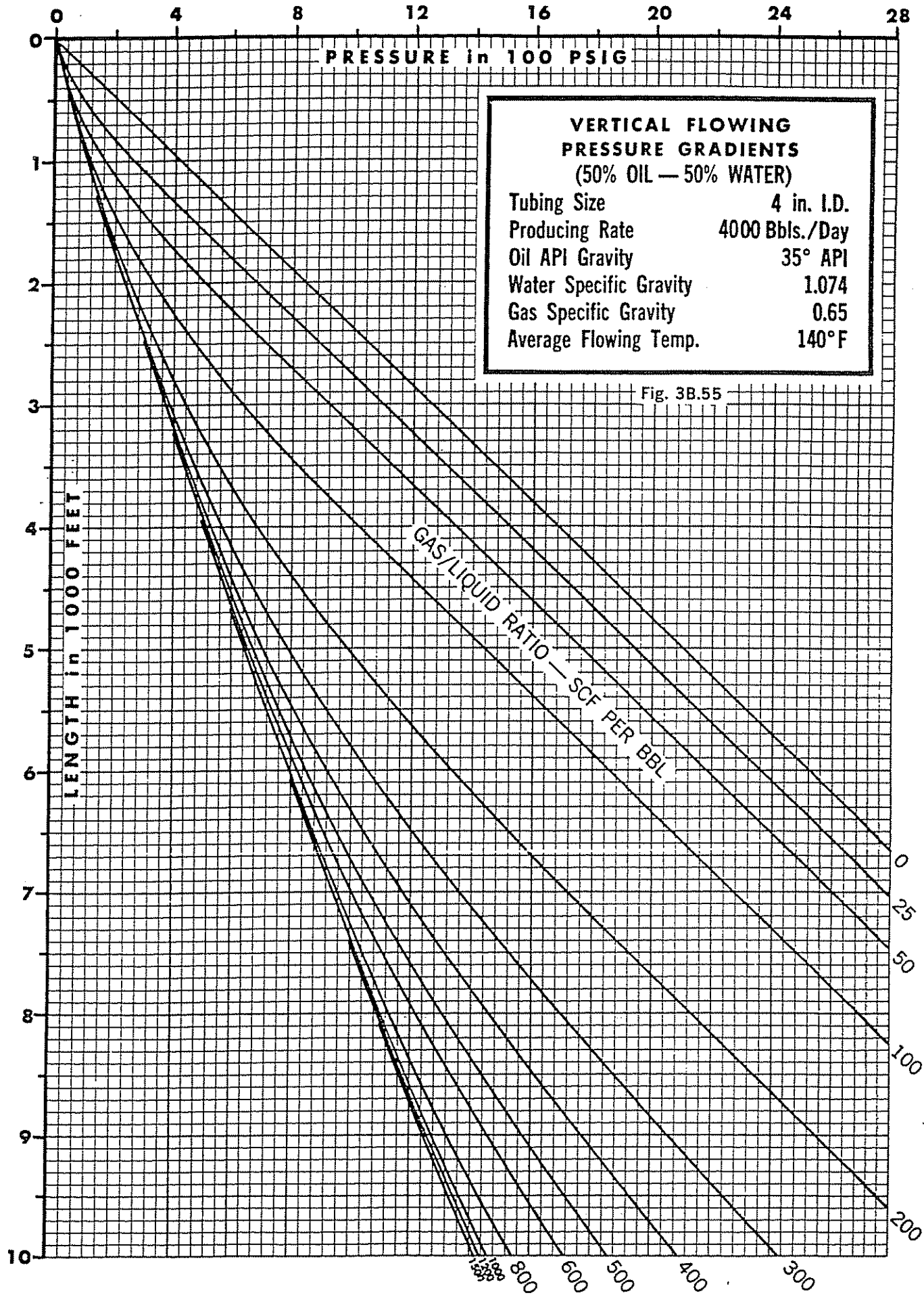
(ALL OIL)

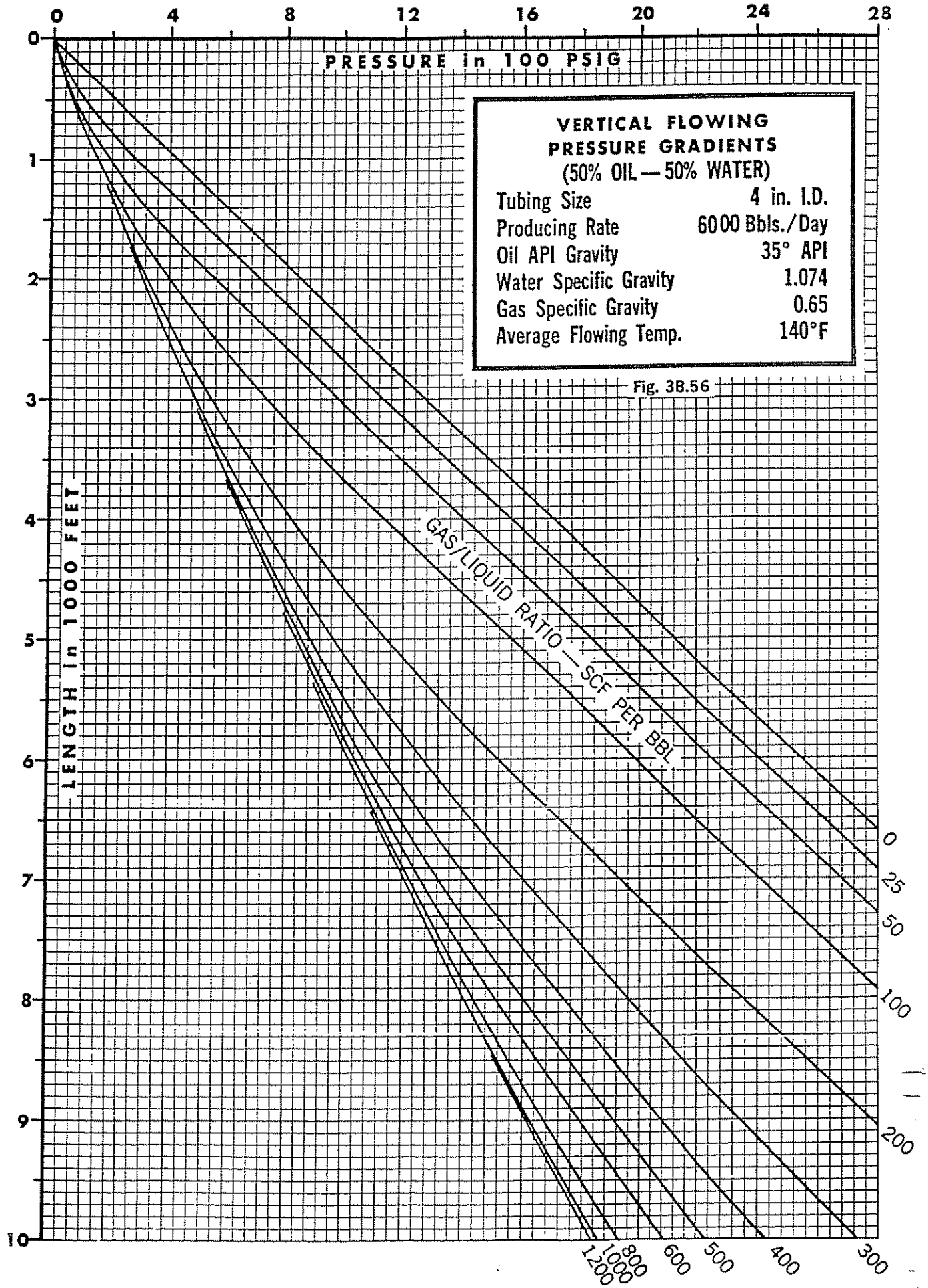
Tubing Size	4 in. I.D.
Producing Rate	8000 Bbls./Day
Oil API Gravity	35° API
Gas Specific Gravity	0.65
Average Flowing Temp.	140° F

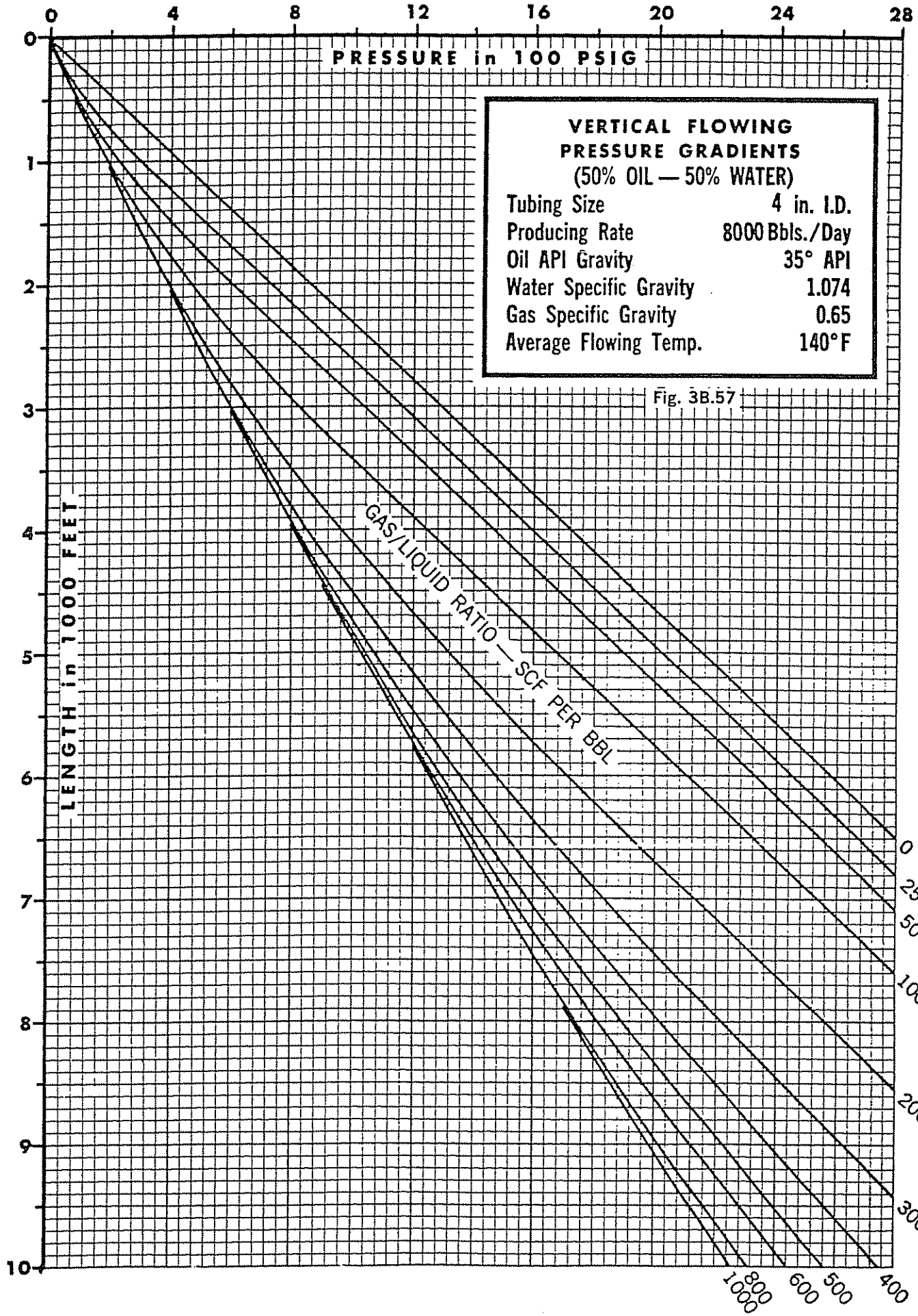

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 4 in. I.D.
Producing Rate 1000 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F
Fig. 3B.52
LENGTH in 1000 FEET
GAS/LIQUID RATIO — SCF PER BBL
0
1
2
3
4
5
6
7
8
9
10
0
25
50
100
200
300
400
500
600
800

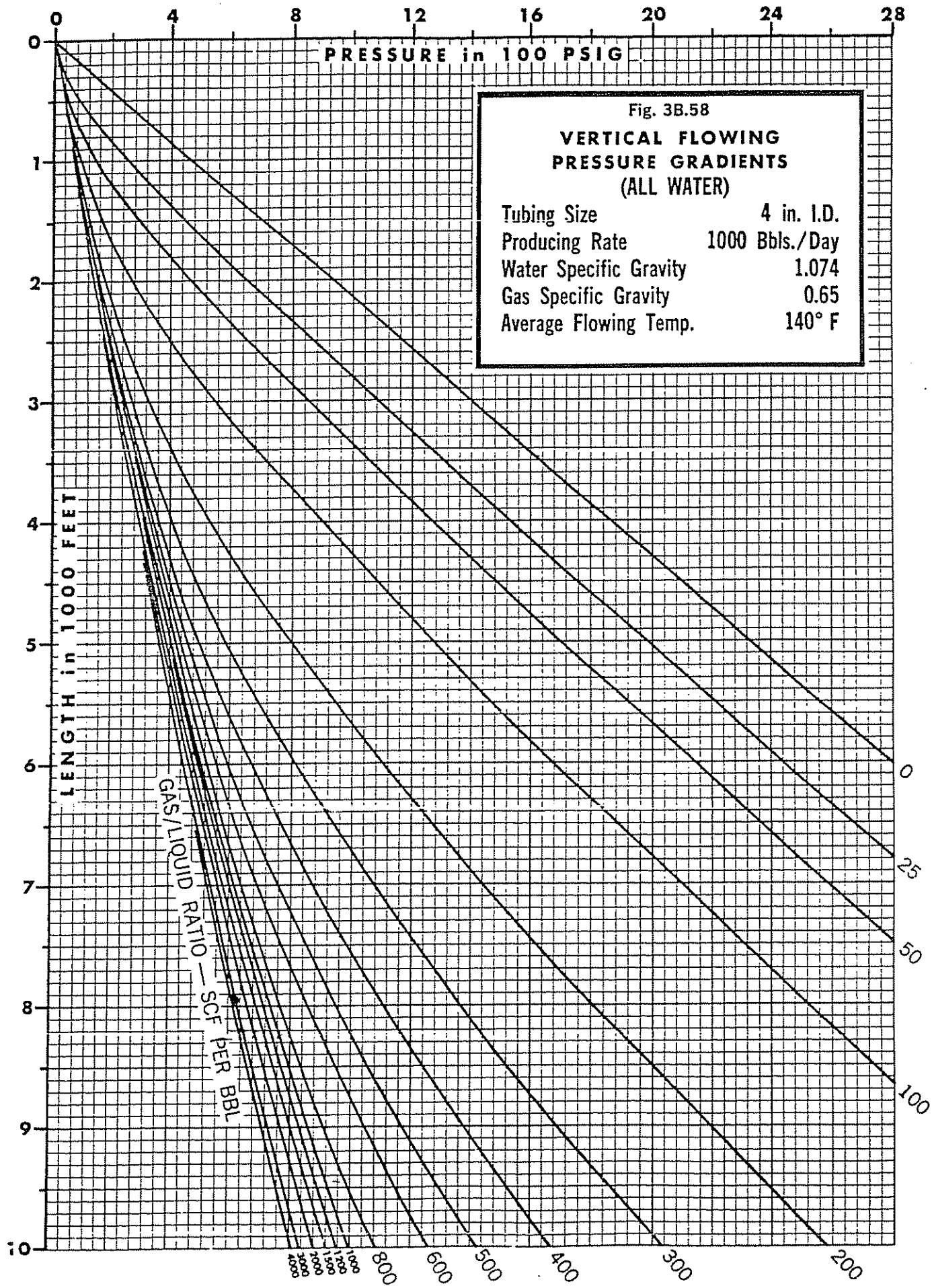

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 4 in. I.D.
Producing Rate 2000 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
Fig. 3B.53
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800

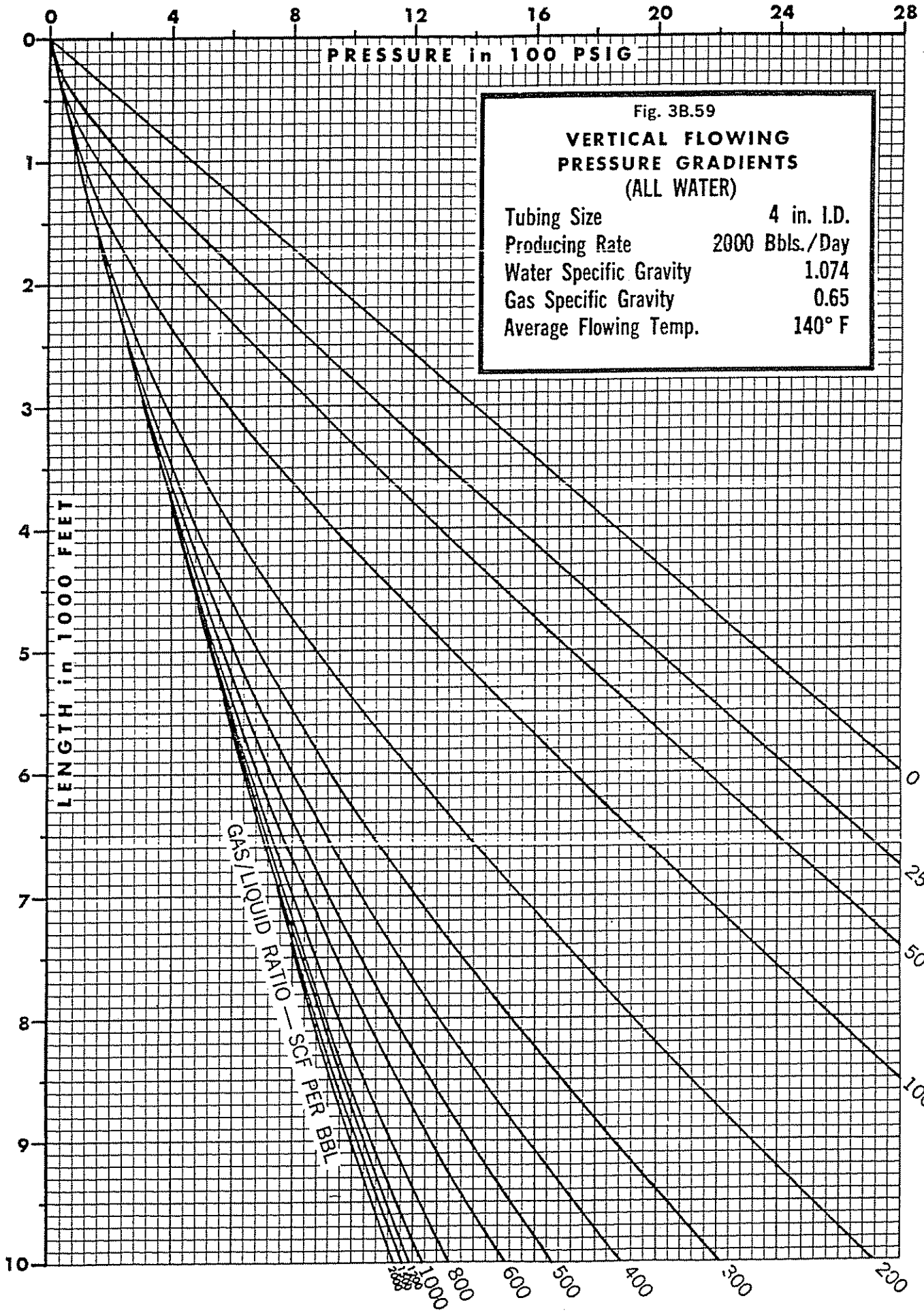

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 4 in. I.D.
Producing Rate 3000 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F
Fig. 3B.54
LENGTH in 1000 FEET
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000
1200

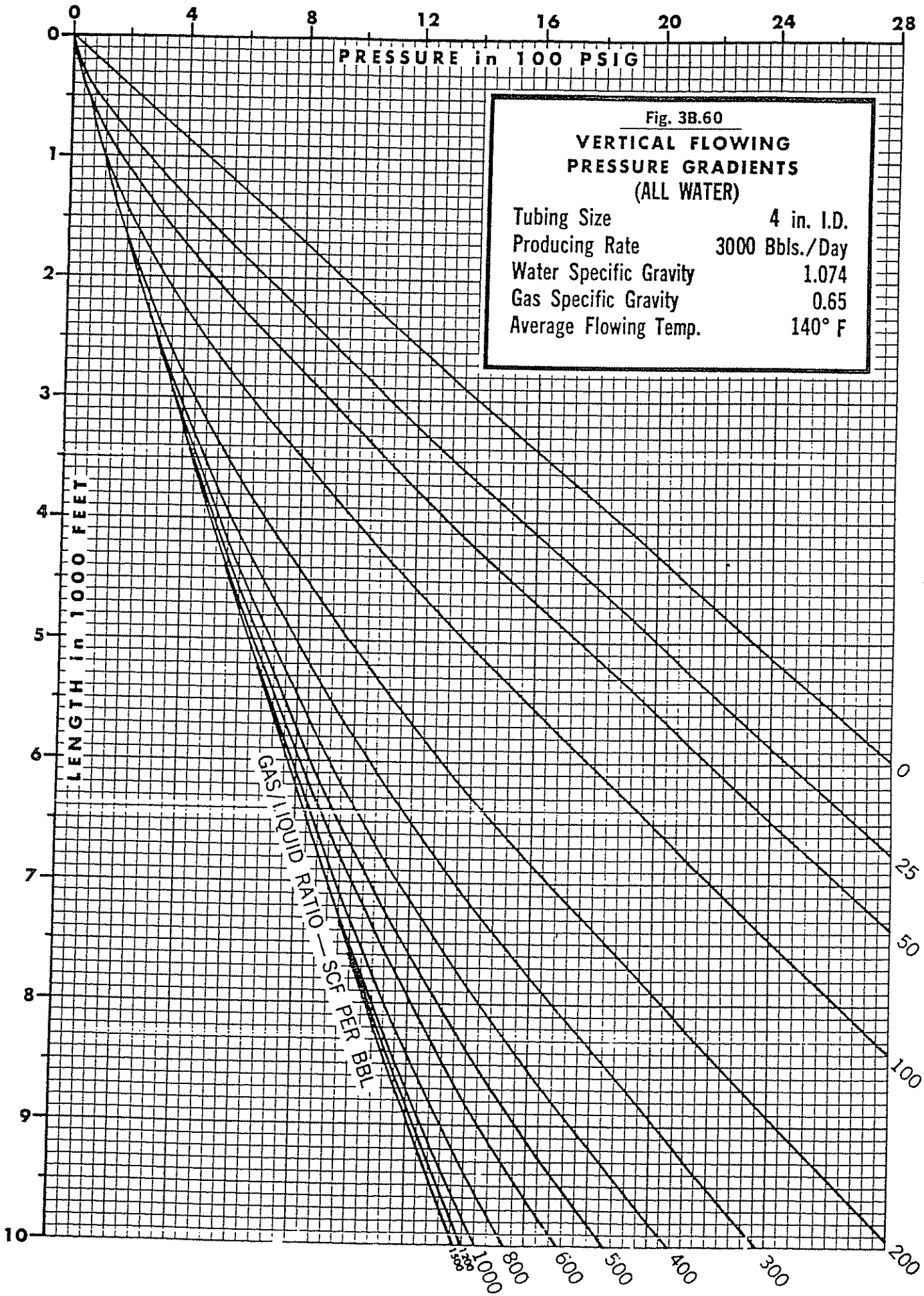

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 4 in. I.D.
Producing Rate 4000 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F
Fig. 3B.55
LENGTH in 1000 FEET
GAS/LIQUID RATIO — SCF PER BBL
0
1
2
3
4
5
6
7
8
9
10
0
25
50
100
200
300
400
500
600
800

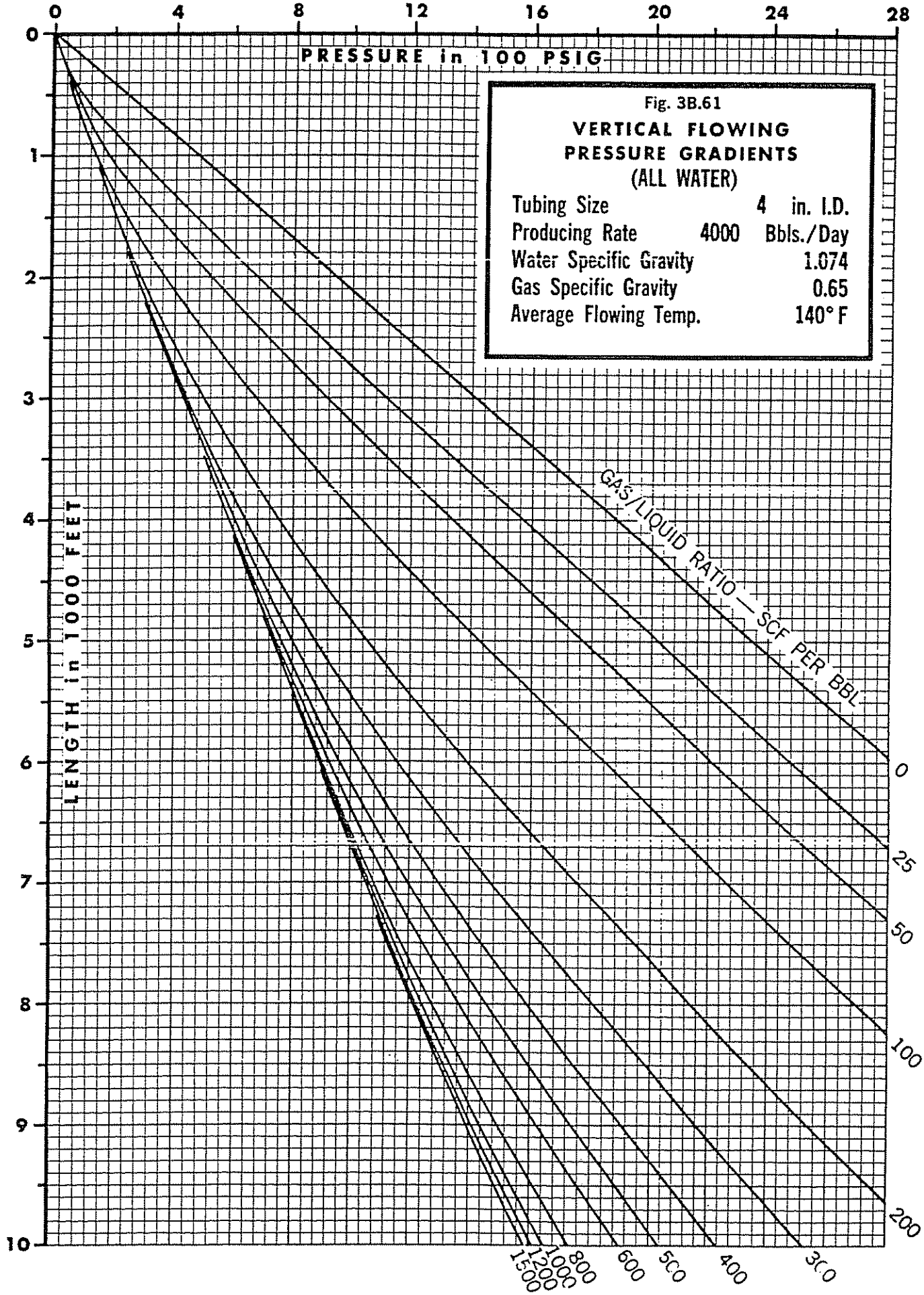

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 4 in. I.D.
Producing Rate 6000 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F
Fig. 3B.56
LENGTH in 1000 FEET
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000
1200

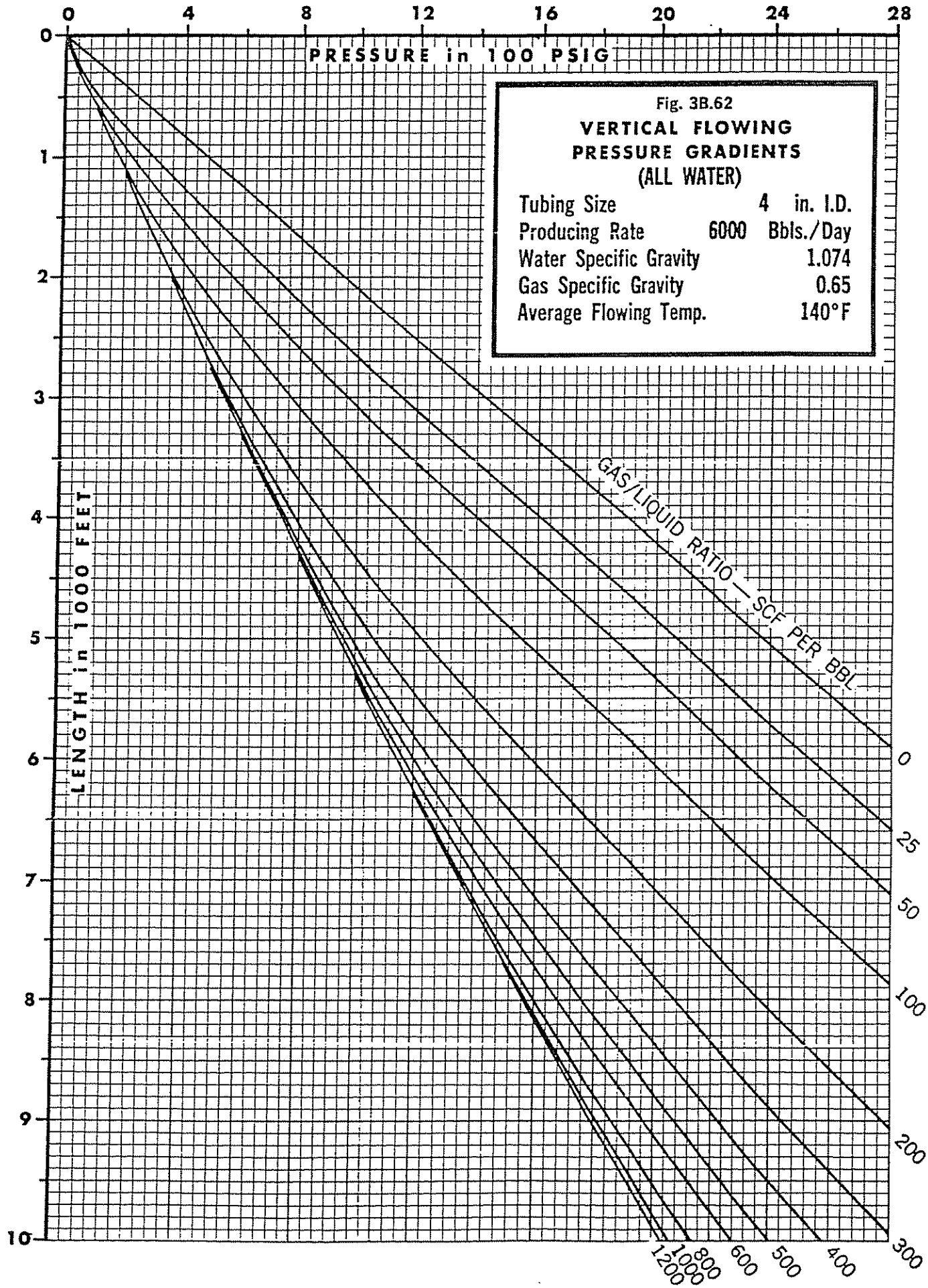

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
VERTICAL FLOWING
PRESSURE GRADIENTS
(50% OIL — 50% WATER)
Tubing Size 4 in. I.D.
Producing Rate 8000 Bbls./Day
Oil API Gravity 35° API
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
Fig. 3B.57
LENGTH in 1000 FEET
GAS/LIQUID RATIO — SCF PER BBL
0
1
2
3
4
5
6
7
8
9
10
0
25
50
100
200
300
400
500
600
800
1000

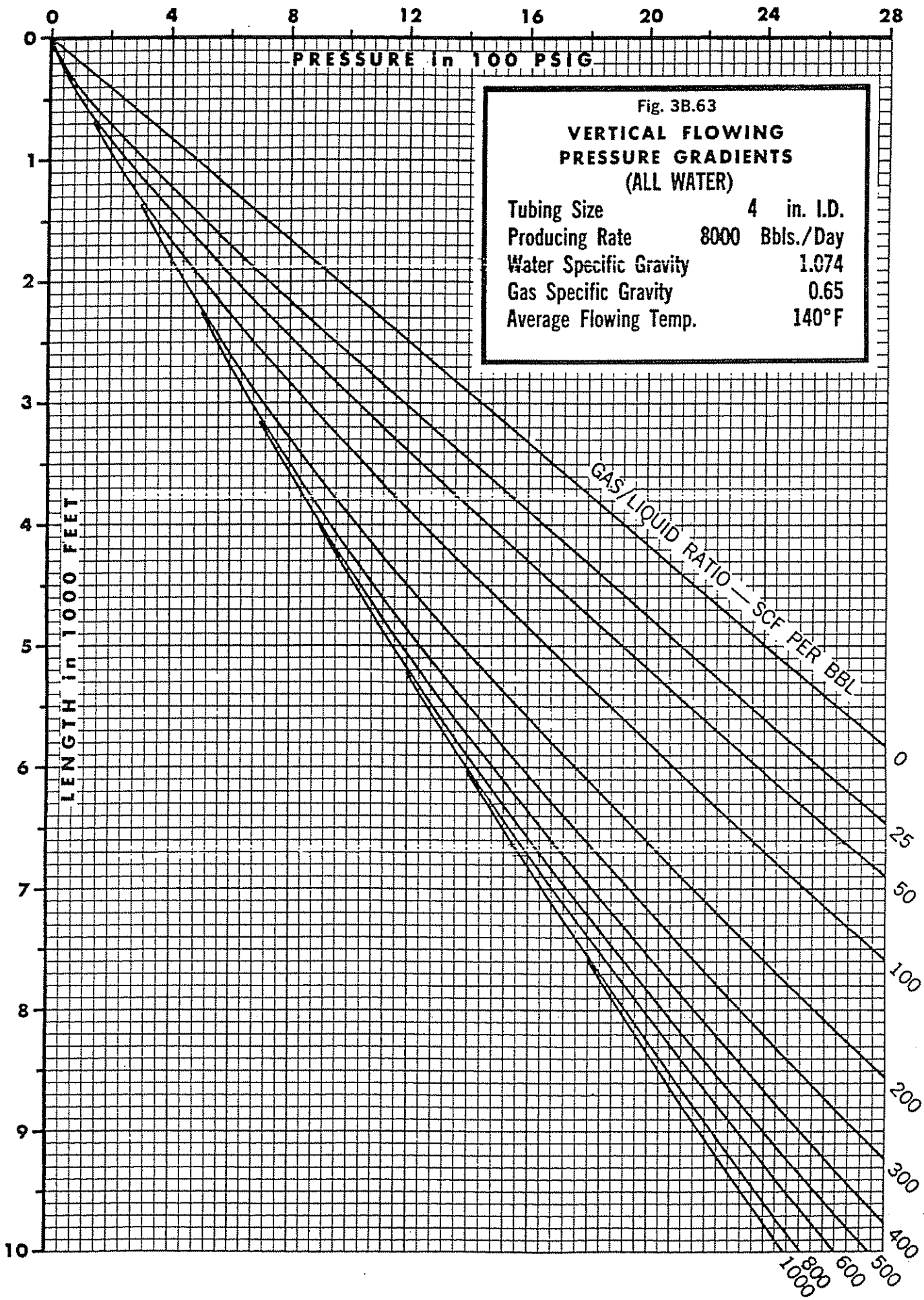

Fig. 3B.58
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Tubing Size 4 in. I.D.
Producing Rate 1000 Bbls./Day
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
PRESSURE in 100 PSIG
0
4
8
12
16
20
24
28
LENGTH in 1000 FEET
0
1
2
3
4
5
6
7
8
9
10
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800

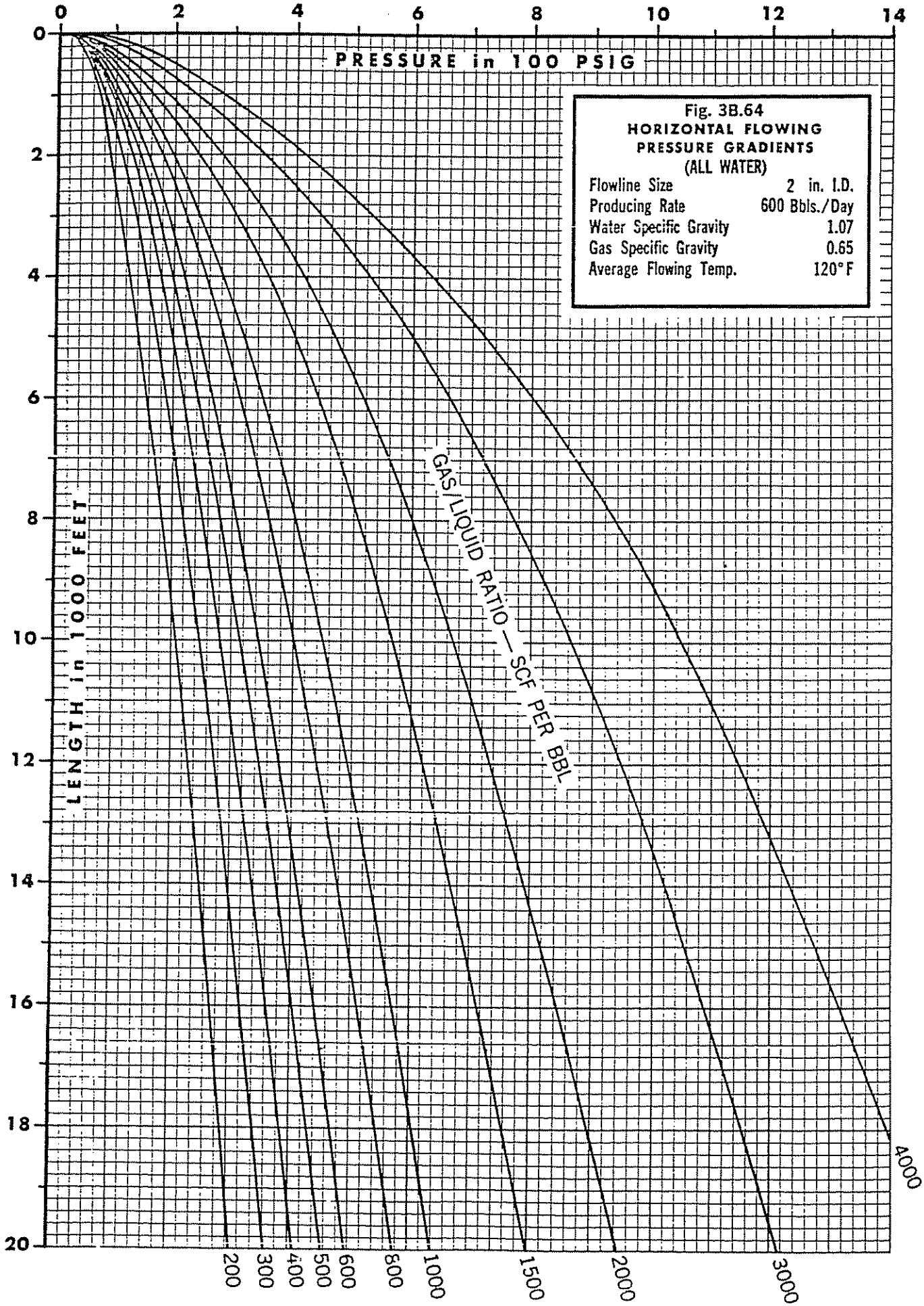

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.59
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Tubing Size 4 in. I.D.
Producing Rate 2000 Bbls./Day
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000

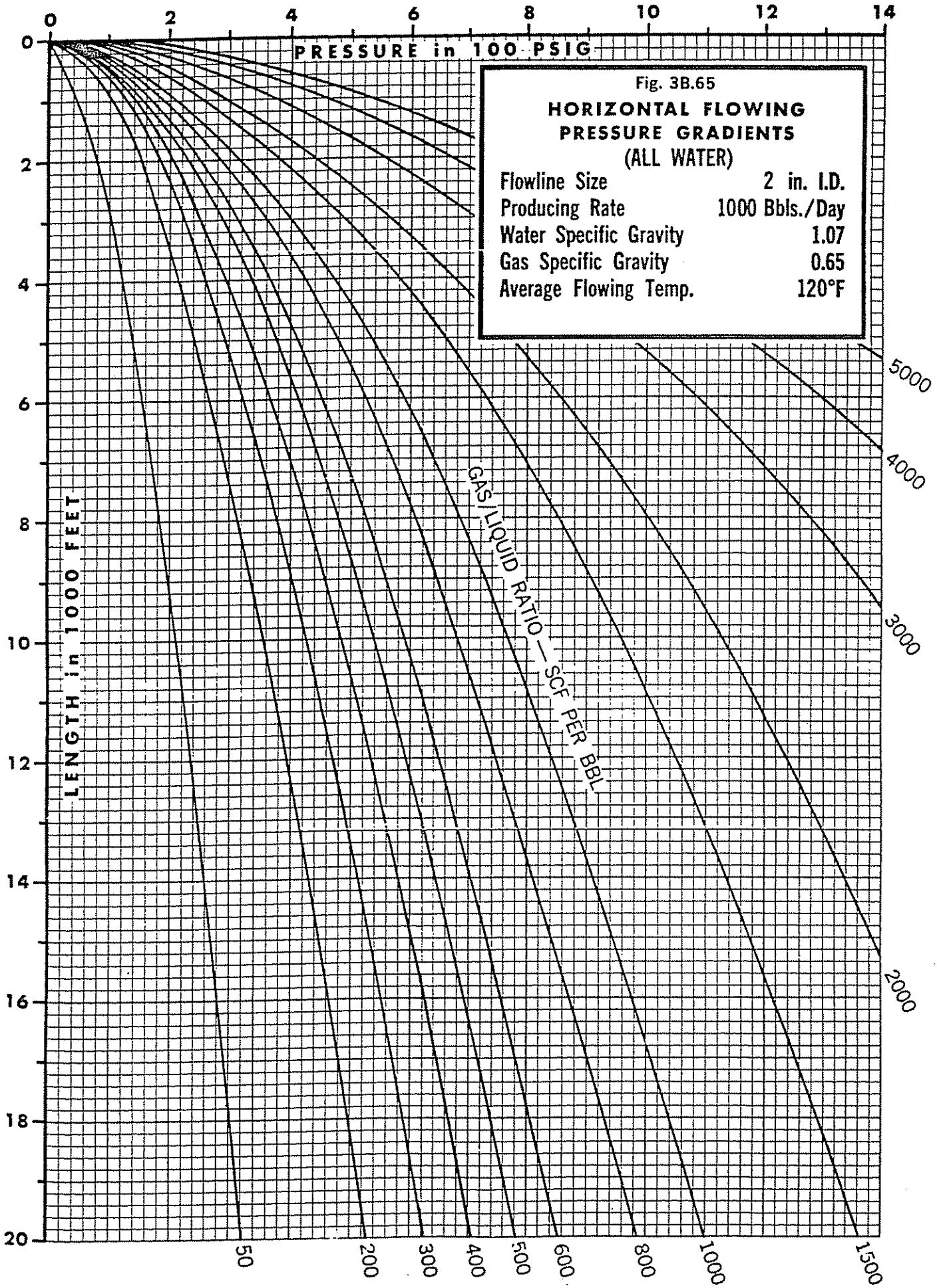

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.60
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Tubing Size 4 in. I.D.
Producing Rate 3000 Bbls./Day
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS / LIQUID RATIO — SCF PER BBL.
0
25
50
100
200
300
400
500
600
800
1000
1333

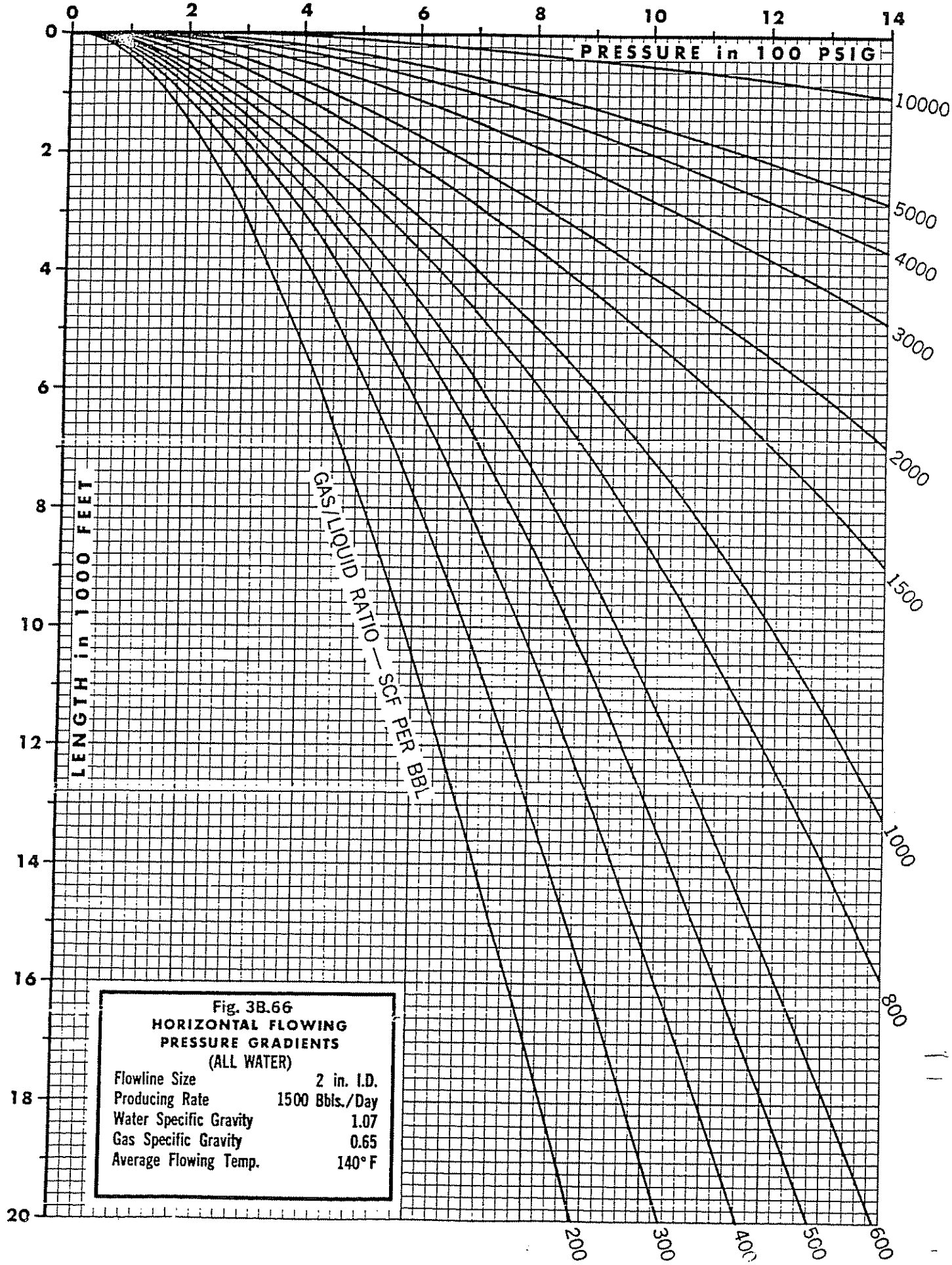

0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.61
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Tubing Size 4 in. I.D.
Producing Rate 4000 Bbls./Day
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000
1200
1500


0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.62
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Tubing Size 4 in. I.D.
Producing Rate 6000 Bbls./Day
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000
1200


0
4
8
12
16
20
24
28
PRESSURE in 100 PSIG
0
1
2
3
4
5
6
7
8
9
10
LENGTH in 1000 FEET
Fig. 3B.63
VERTICAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Tubing Size 4 in. I.D.
Producing Rate 8000 Bbls./Day
Water Specific Gravity 1.074
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F
GAS/LIQUID RATIO — SCF PER BBL
0
25
50
100
200
300
400
500
600
800
1000


0
2
4
6
8
10
12
14
PRESSURE in 100 PSIG
LENGTH in 1000 FEET
GAS/LIQUID RATIO — SCF PER BBL
200
300
400
500
600
800
1000
1500
2000
3000
4000
Fig. 3B.64
HORIZONTAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Flowline Size 2 in. I.D.
Producing Rate 600 Bbls./Day
Water Specific Gravity 1.07
Gas Specific Gravity 0.65
Average Flowing Temp. 120°F


0
2
4
6
8
10
12
14
PRESSURE in 100 PSIG
LENGTH in 1000 FEET
0
2
4
6
8
10
12
14
16
18
20
Fig. 3B.65
HORIZONTAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Flowline Size 2 in. I.D.
Producing Rate 1000 Bbls./Day
Water Specific Gravity 1.07
Gas Specific Gravity 0.65
Average Flowing Temp. 120°F
GAS/LIQUID RATIO — SCF PER BBL
50
200
300
400
500
600
800
1000
1500
2000
3000
4000
5000


0
2
4
6
8
10
12
14
PRESSURE in 100 PSIG
LENGTH in 1000 FEET
GAS/LIQUID RATIO — SCF PER BBL
10000
5000
4000
3000
2000
1500
1000
800
600
500
400
300
200
Fig. 3B.66
HORIZONTAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Flowline Size 2 in. I.D.
Producing Rate 1500 Bbls./Day
Water Specific Gravity 1.07
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F

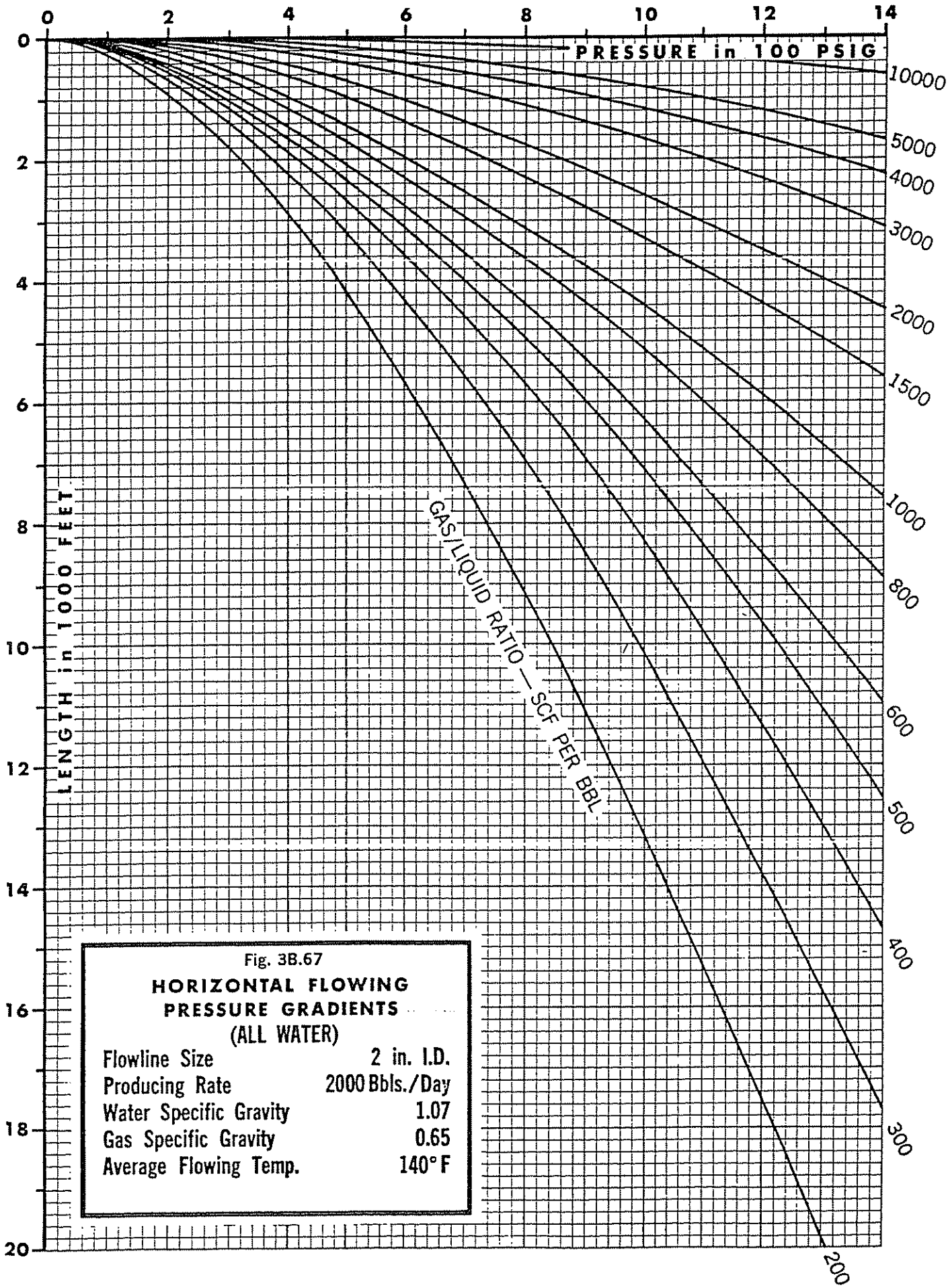
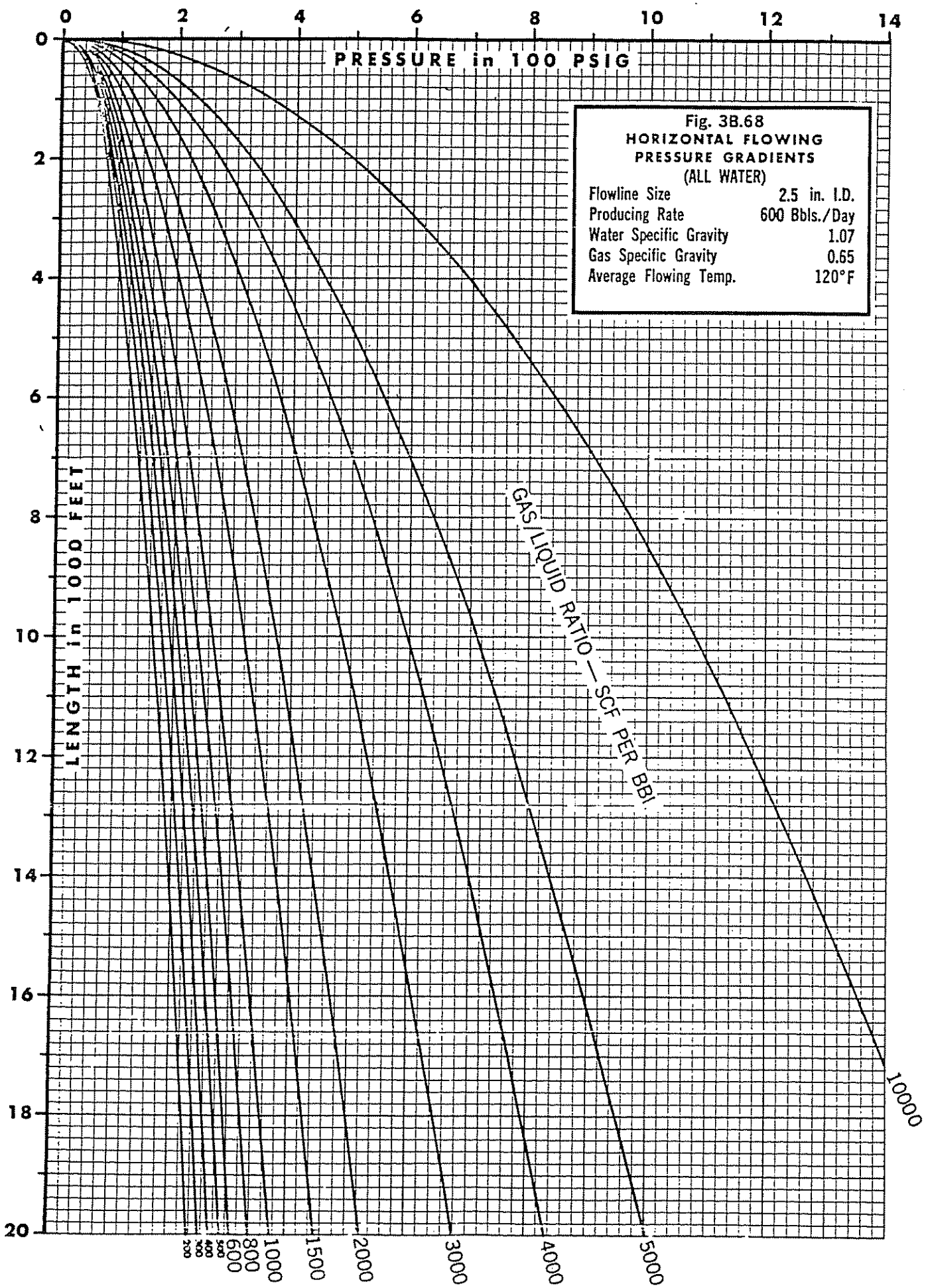


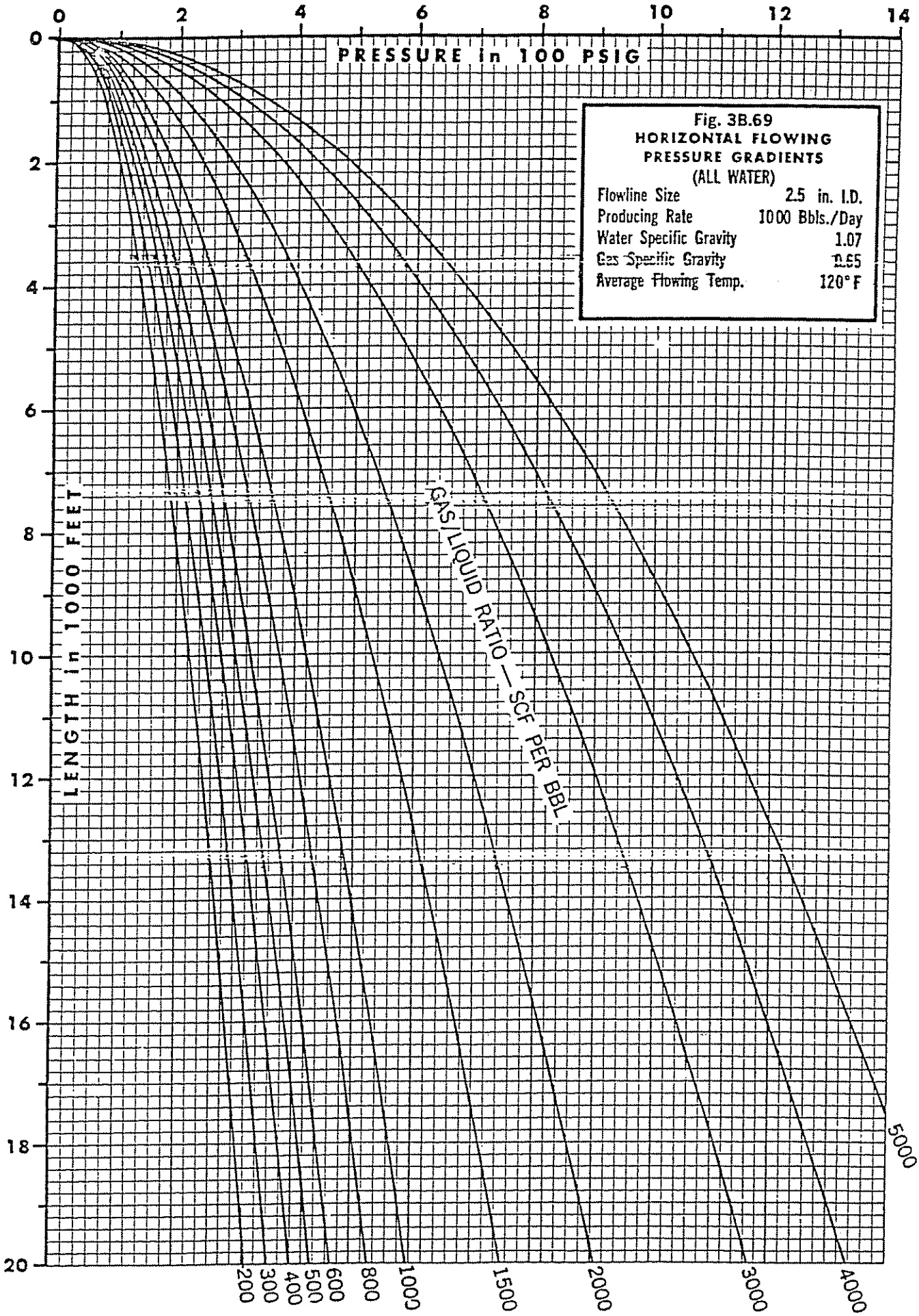
Fig. 3B.67

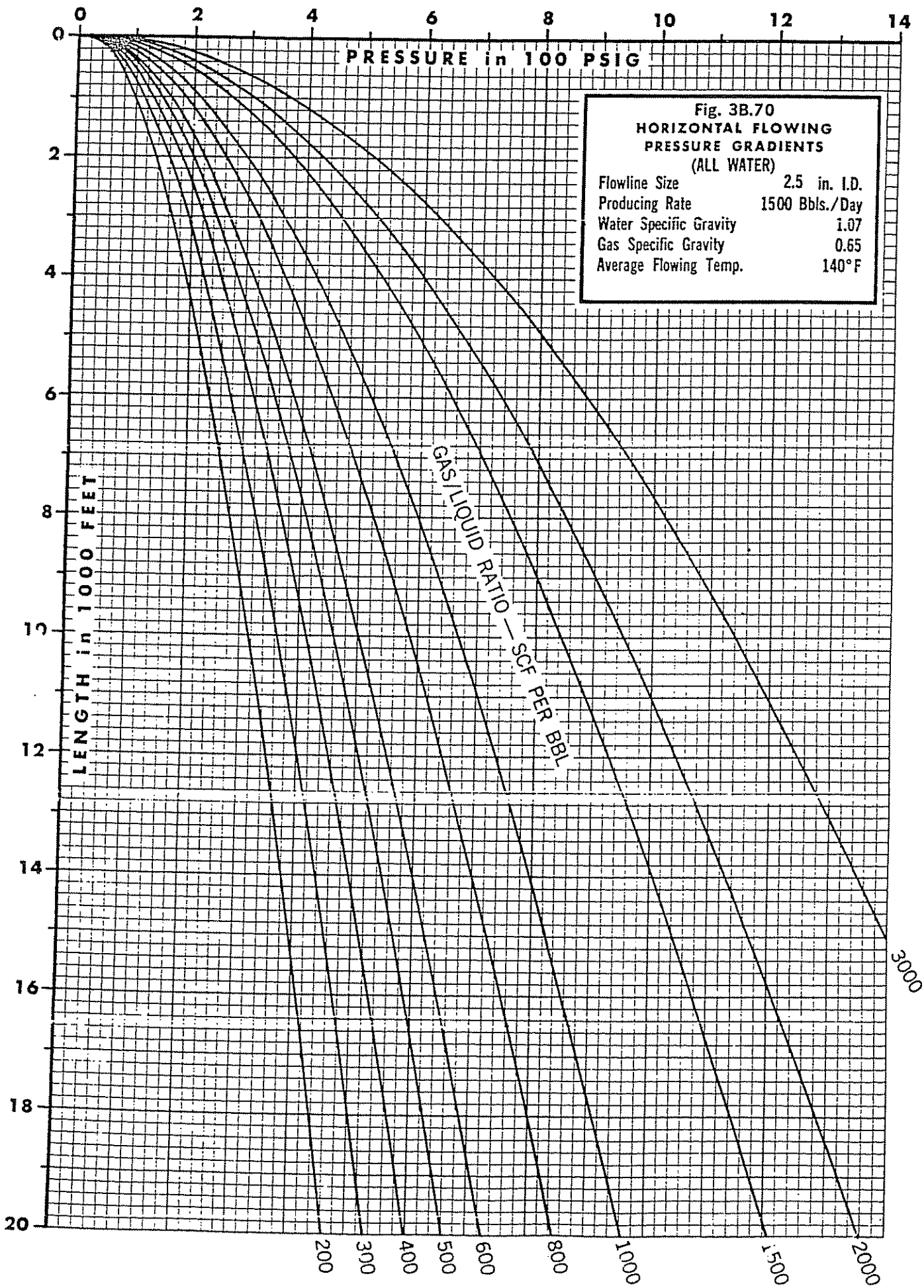
HORIZONTAL FLOWING PRESSURE GRADIENTS

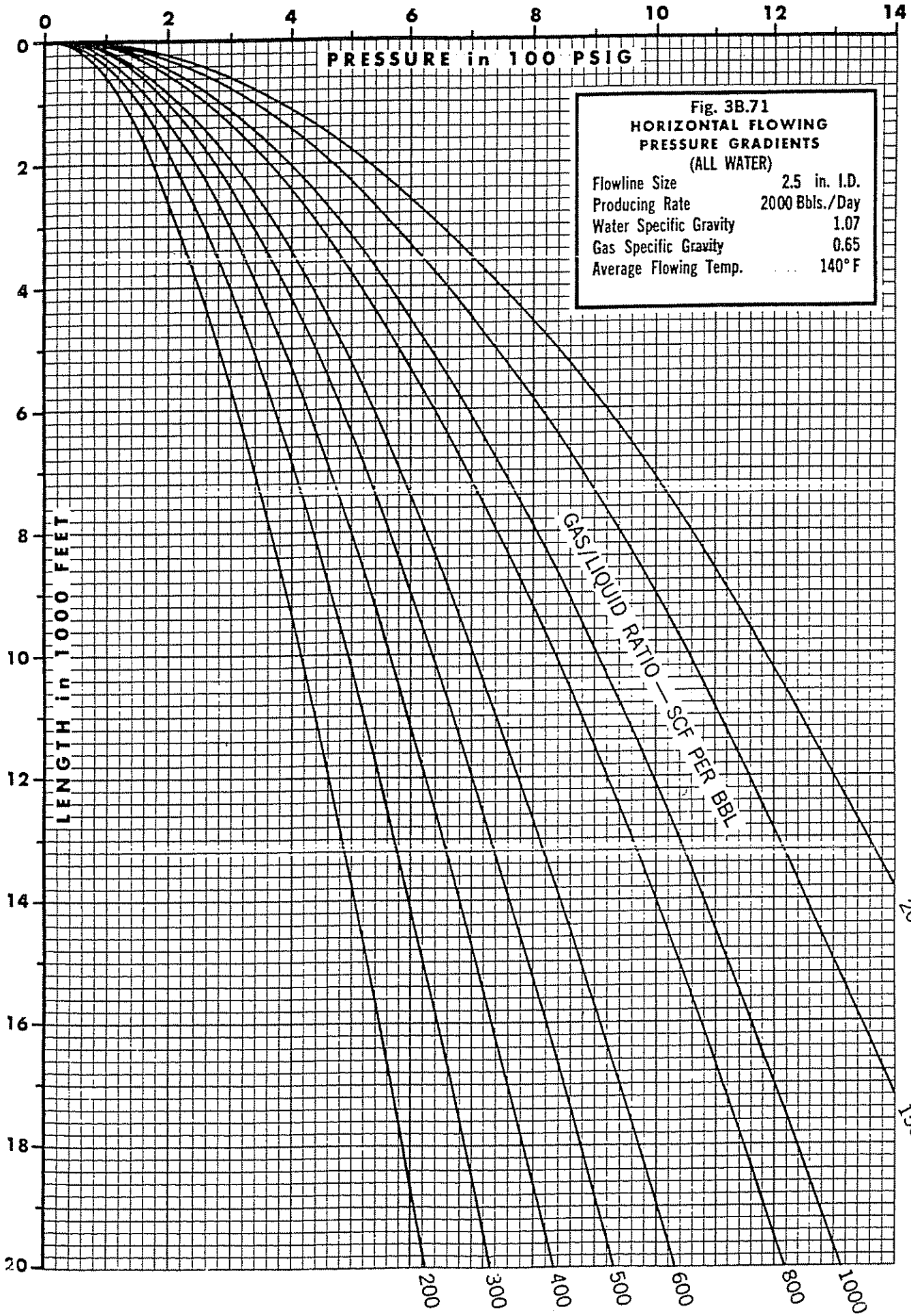
(ALL WATER)

Flowline Size	2 in. I.D.
Producing Rate	2000 Bbls./Day
Water Specific Gravity	1.07
Gas Specific Gravity	0.65
Average Flowing Temp.	140° F


0
2
4
6
8
10
12
14
PRESSURE in 100 PSIG
LENGTH in 1000 FEET
0
2
4
6
8
10
12
14
16
18
20
Fig. 3B.68
HORIZONTAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Flowline Size 2.5 in. I.D.
Producing Rate 600 Bbls./Day
Water Specific Gravity 1.07
Gas Specific Gravity 0.65
Average Flowing Temp. 120°F
GAS LIQUID RATIO — SCF PER BBL
200
300
400
500
600
800
1000
1500
2000
3000
4000
5000
10000


0
2
4
6
8
10
12
14
PRESSURE in 100 PSIG
Fig. 3B.69
HORIZONTAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Flowline Size 2.5 in. I.D.
Producing Rate 1000 Bbls./Day
Water Specific Gravity 1.07
Gas Specific Gravity 0.65
Average Flowing Temp. 120° F
LENGTH in 1000 FEET
GAS/LIQUID RATIO — SCF PER BBL
200
300
400
500
600
800
1000
1500
2000
3000
4000
5000


0
2
4
6
8
10
12
14
PRESSURE in 100 PSIG
Fig. 3B.70
HORIZONTAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Flowline Size 2.5 in. I.D.
Producing Rate 1500 Bbls./Day
Water Specific Gravity 1.07
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F
LENGTH in 1000 FEET
GAS/LIQUID RATIO — SCF PER BBL
200
300
400
500
600
800
1000
1500
2000
3000


0
2
4
6
8
10
12
14
PRESSURE in 100 PSIG
LENGTH in 1000 FEET
0
2
4
6
8
10
12
14
16
18
20
Fig. 3B.71
HORIZONTAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Flowline Size 2.5 in. I.D.
Producing Rate 2000 Bbls./Day
Water Specific Gravity 1.07
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
GAS/LIQUID RATIO — SCF PER BBL
200
300
400
500
600
800
1000
1500
2000

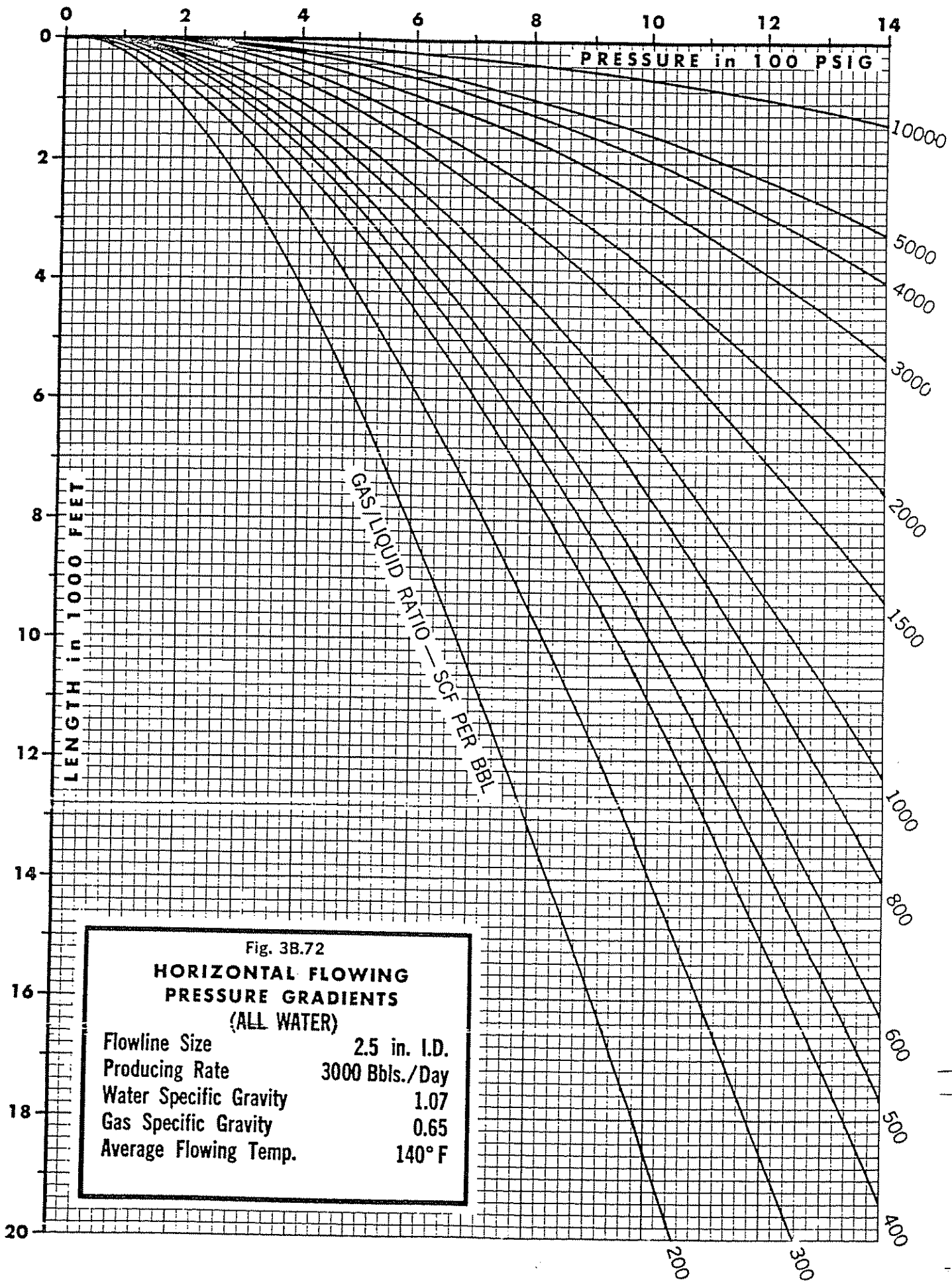
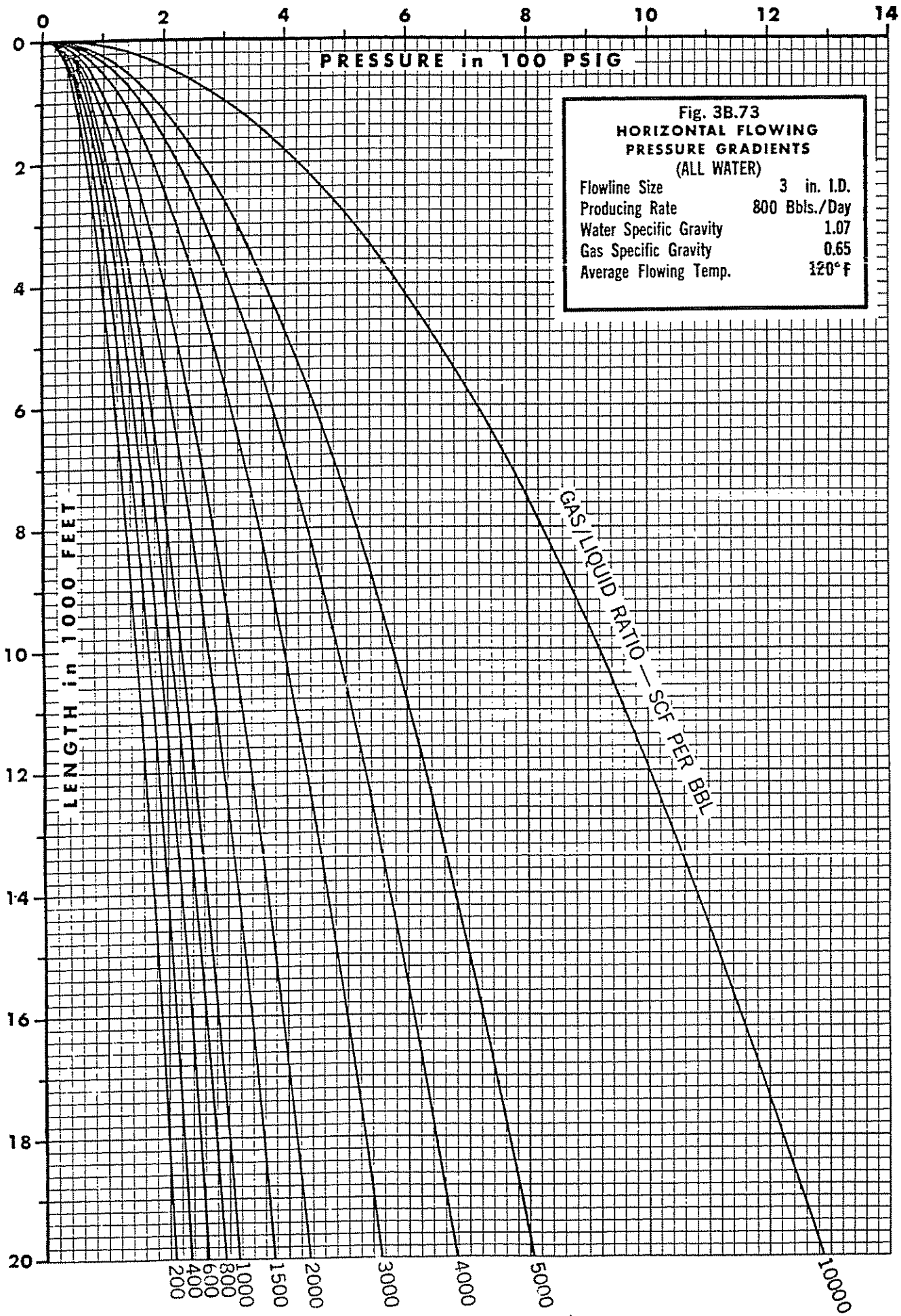
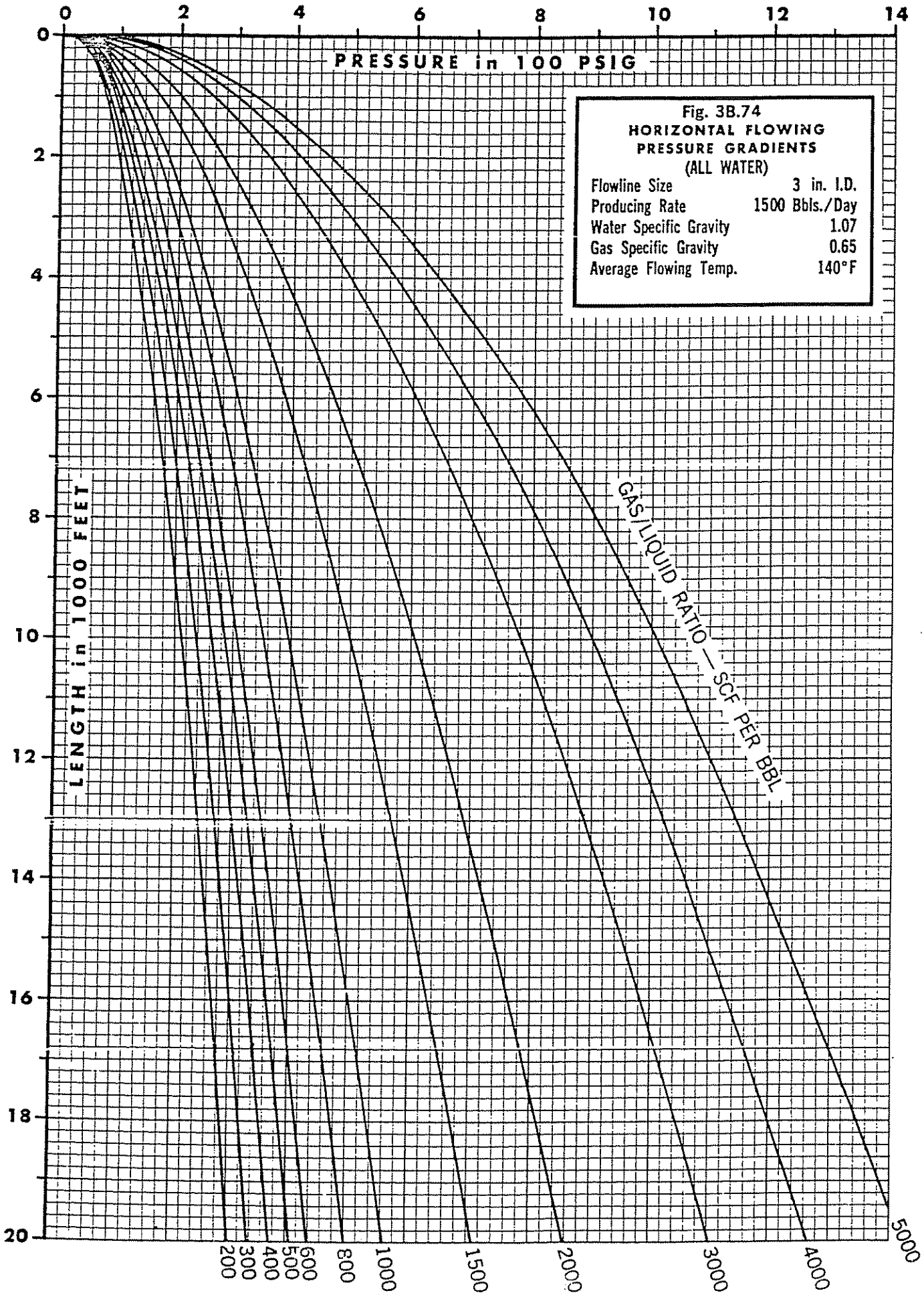


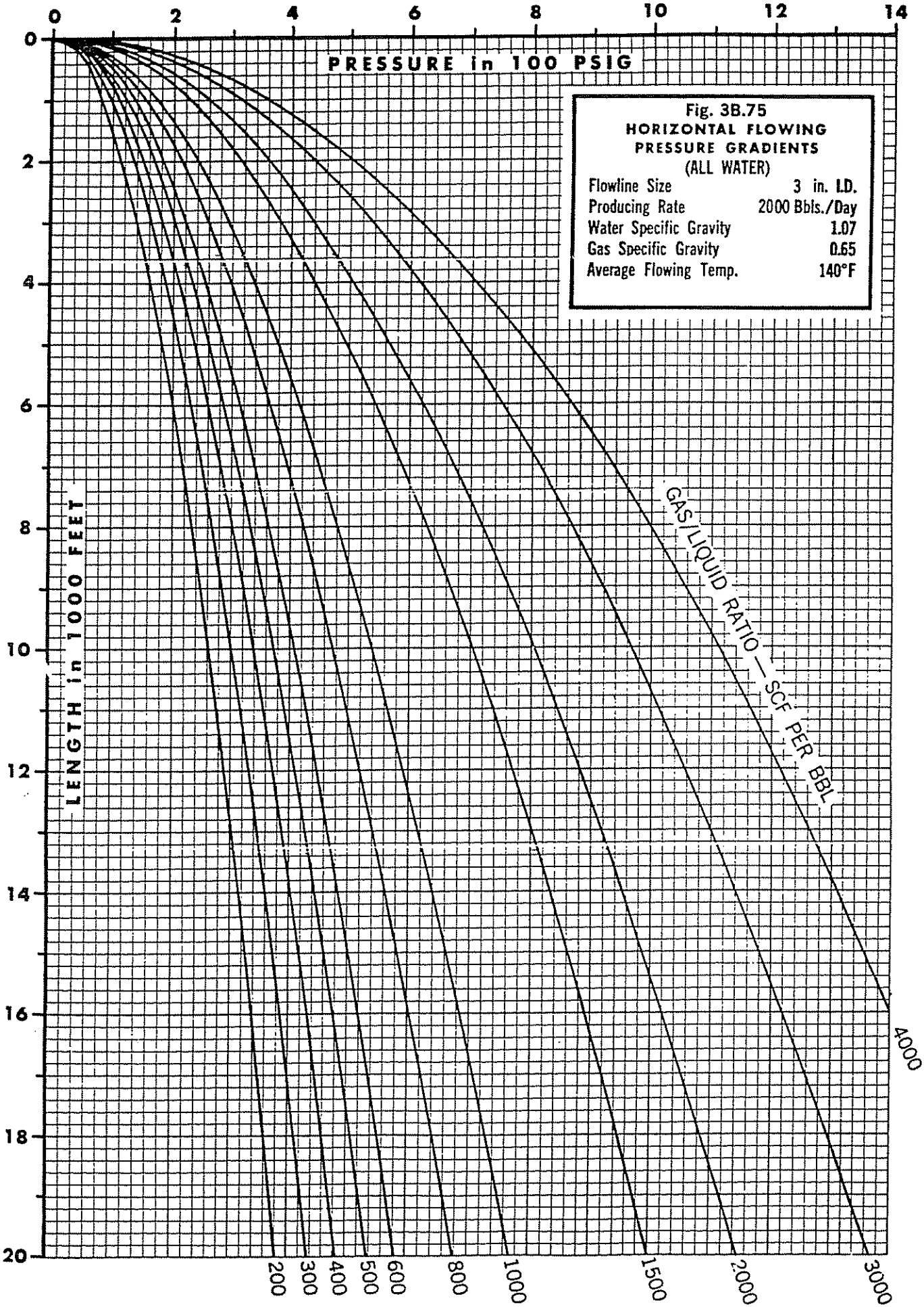
Fig. 3B.72

HORIZONTAL FLOWING PRESSURE GRADIENTS (ALL WATER)

Flowline Size	2.5 in. I.D.
Producing Rate	3000 Bbls./Day
Water Specific Gravity	1.07
Gas Specific Gravity	0.65
Average Flowing Temp.	140° F


0
2
4
6
8
10
12
14
PRESSURE in 100 PSIG
LENGTH in 1000 FEET
0
2
4
6
8
10
12
14
16
18
20
GAS/LIQUID RATIO — SCF PER BBL
200
400
600
800
1000
1500
2000
3000
4000
5000
10000
Fig. 3B.73
HORIZONTAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Flowline Size 3 in. I.D.
Producing Rate 800 Bbls./Day
Water Specific Gravity 1.07
Gas Specific Gravity 0.65
Average Flowing Temp. 120° F


PRESSURE in 100 PSIG
0
2
4
6
8
10
12
14
LENGTH in 1000 FEET
0
2
4
6
8
10
12
14
16
18
20
GAS/LIQUID RATIO — SCF PER BBL
200
300
400
500
600
800
1000
1500
2000
3000
4000
5000
Fig. 3B.74
HORIZONTAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Flowline Size 3 in. I.D.
Producing Rate 1500 Bbls./Day
Water Specific Gravity 1.07
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F


0
2
4
6
8
10
12
14
PRESSURE in 100 PSIG
LENGTH in 1000 FEET
0
2
4
6
8
10
12
14
16
18
20
GAS/LIQUID RATIO — SCF PER BBL
200
300
400
500
600
800
1000
1500
2000
3000
4000
Fig. 3B.75
HORIZONTAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Flowline Size 3 in. I.D.
Producing Rate 2000 Bbls./Day
Water Specific Gravity 1.07
Gas Specific Gravity 0.65
Average Flowing Temp. 140°F

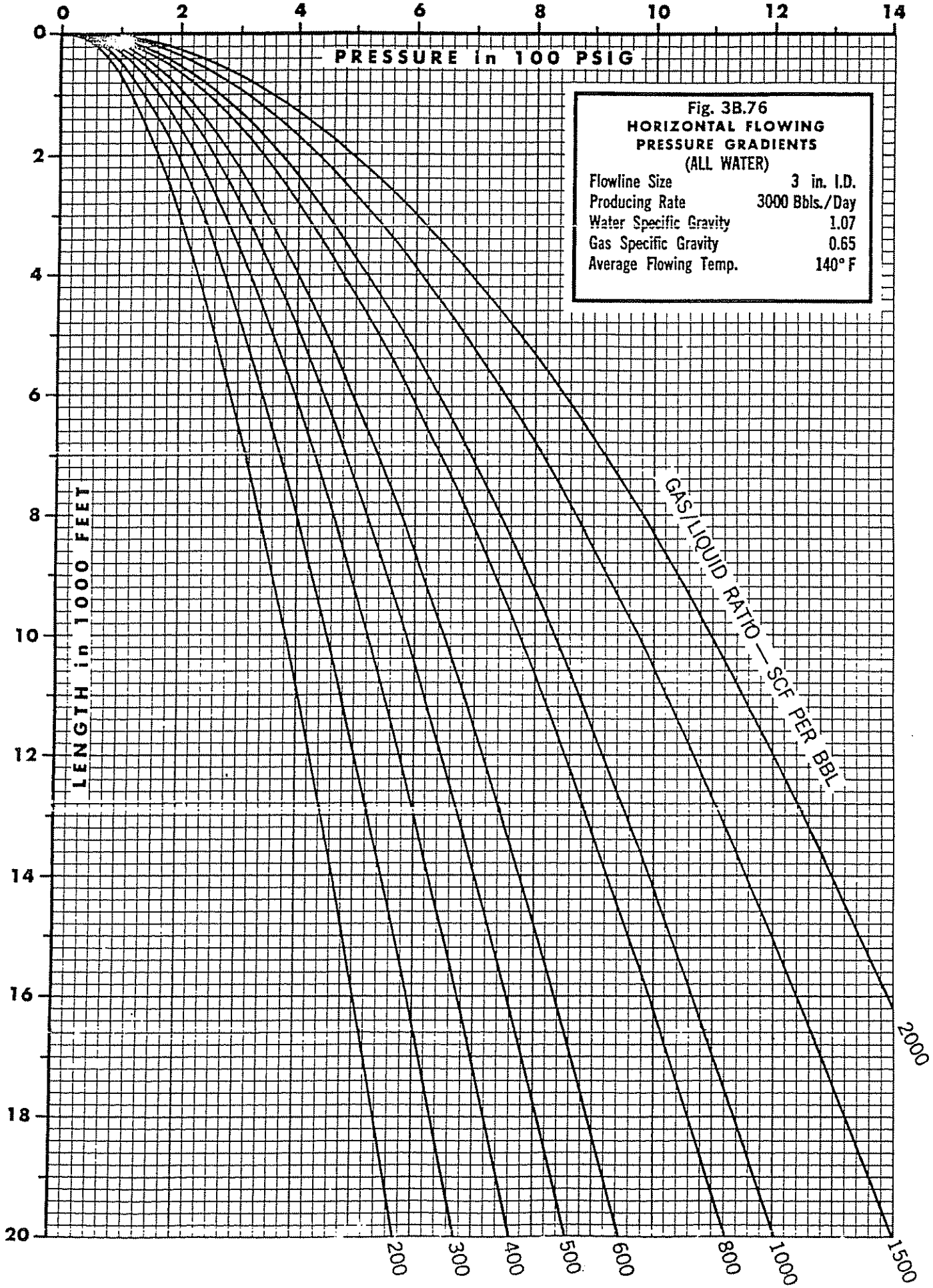


Fig. 3B.76
HORIZONTAL FLOWING PRESSURE GRADIENTS
(ALL WATER)

Flowline Size	3 in. I.D.
Producing Rate	3000 Bbls./Day
Water Specific Gravity	1.07
Gas Specific Gravity	0.65
Average Flowing Temp.	140° F

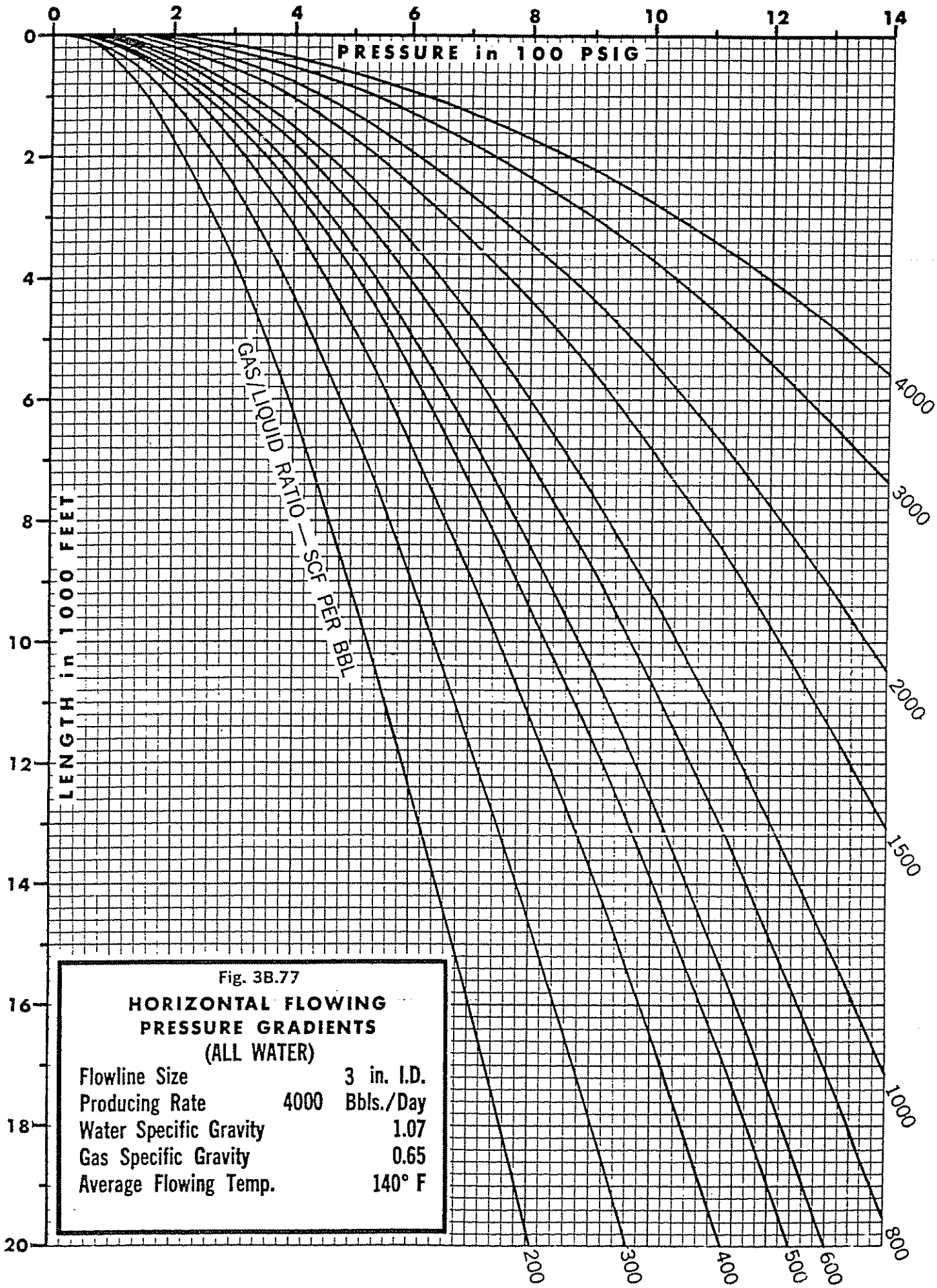


Fig. 3B.77

HORIZONTAL FLOWING PRESSURE GRADIENTS
(ALL WATER)

Flowline Size	3 in. I.D.
Producing Rate	4000 Bbls./Day
Water Specific Gravity	1.07
Gas Specific Gravity	0.65
Average Flowing Temp.	140° F

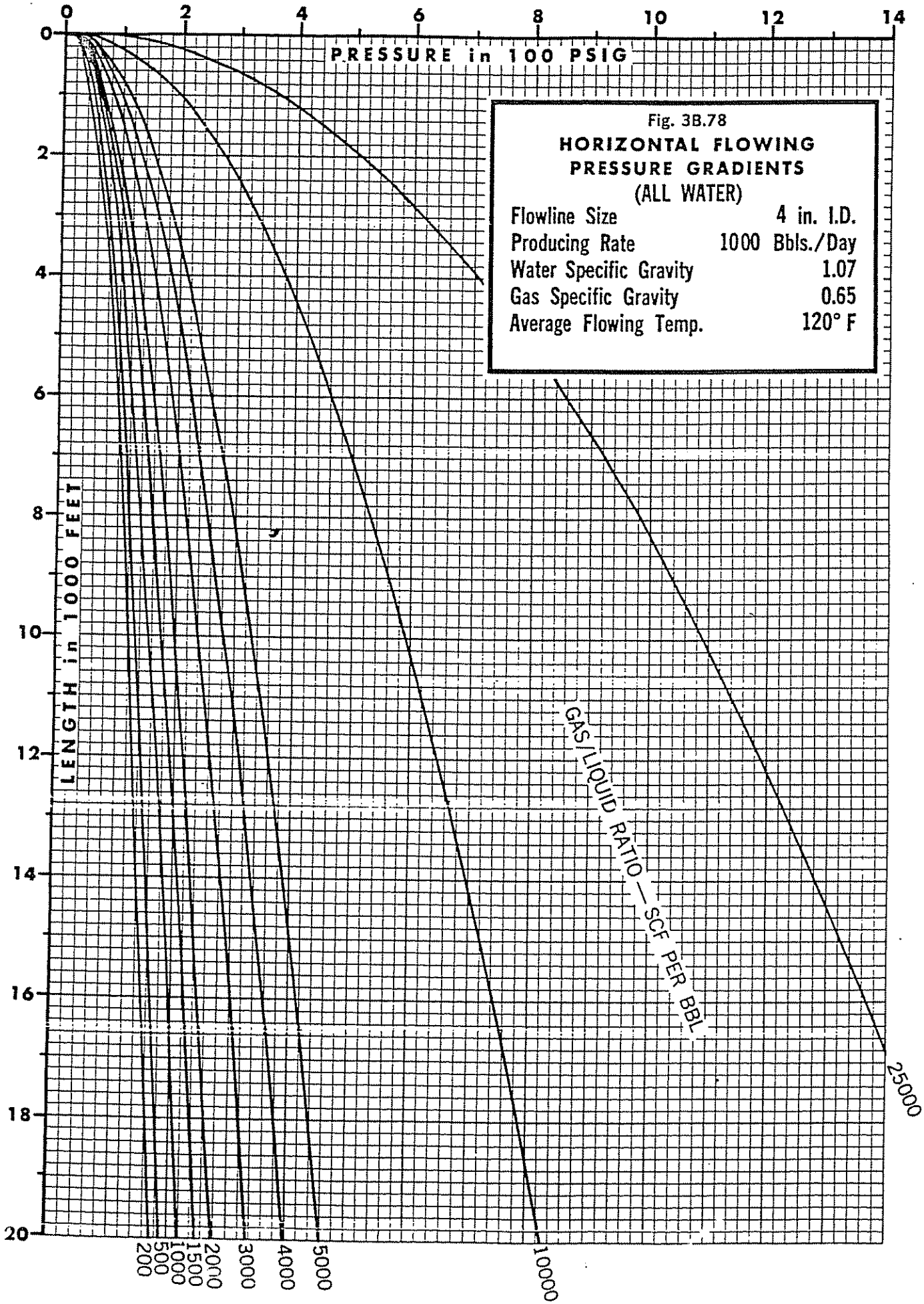
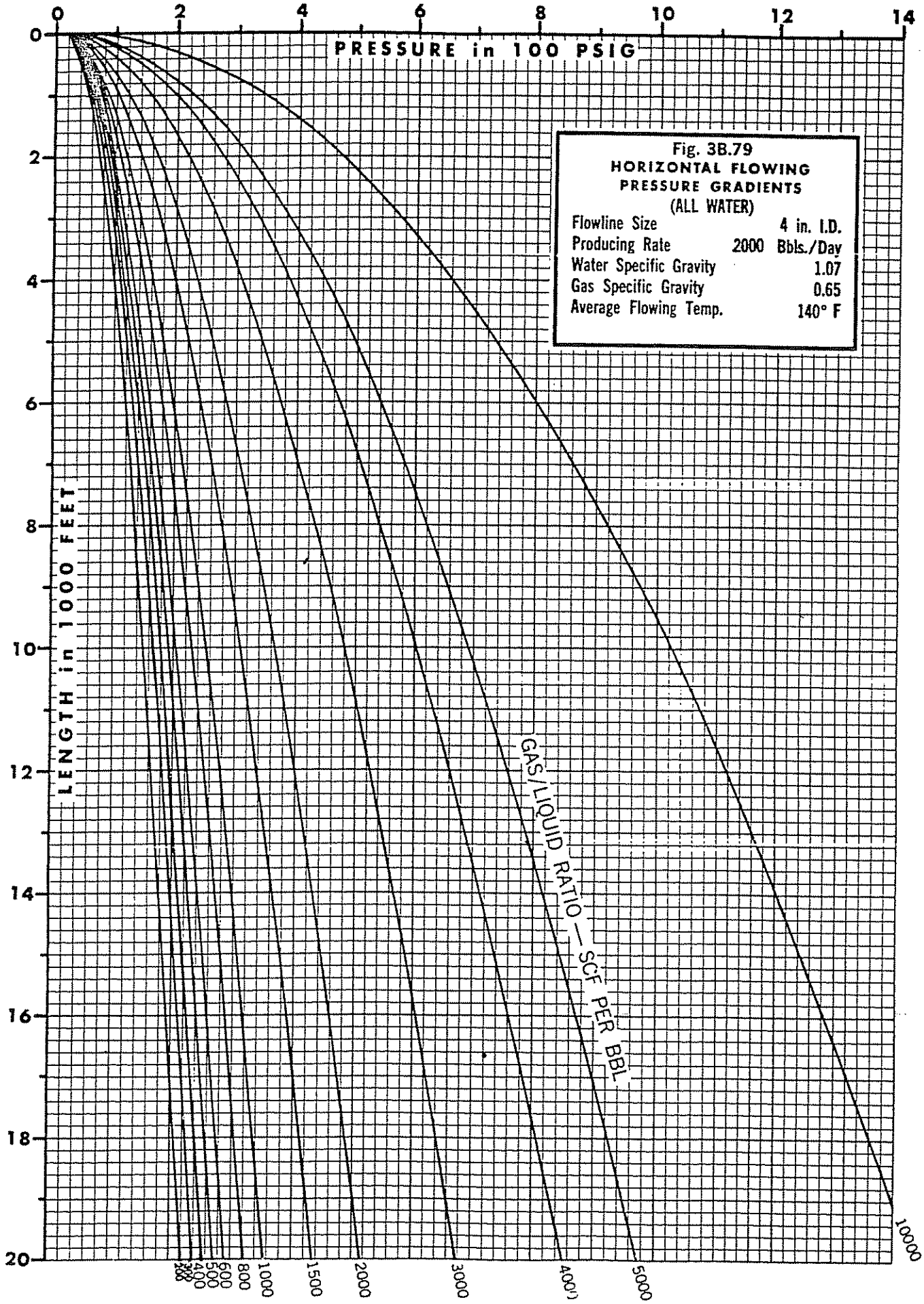


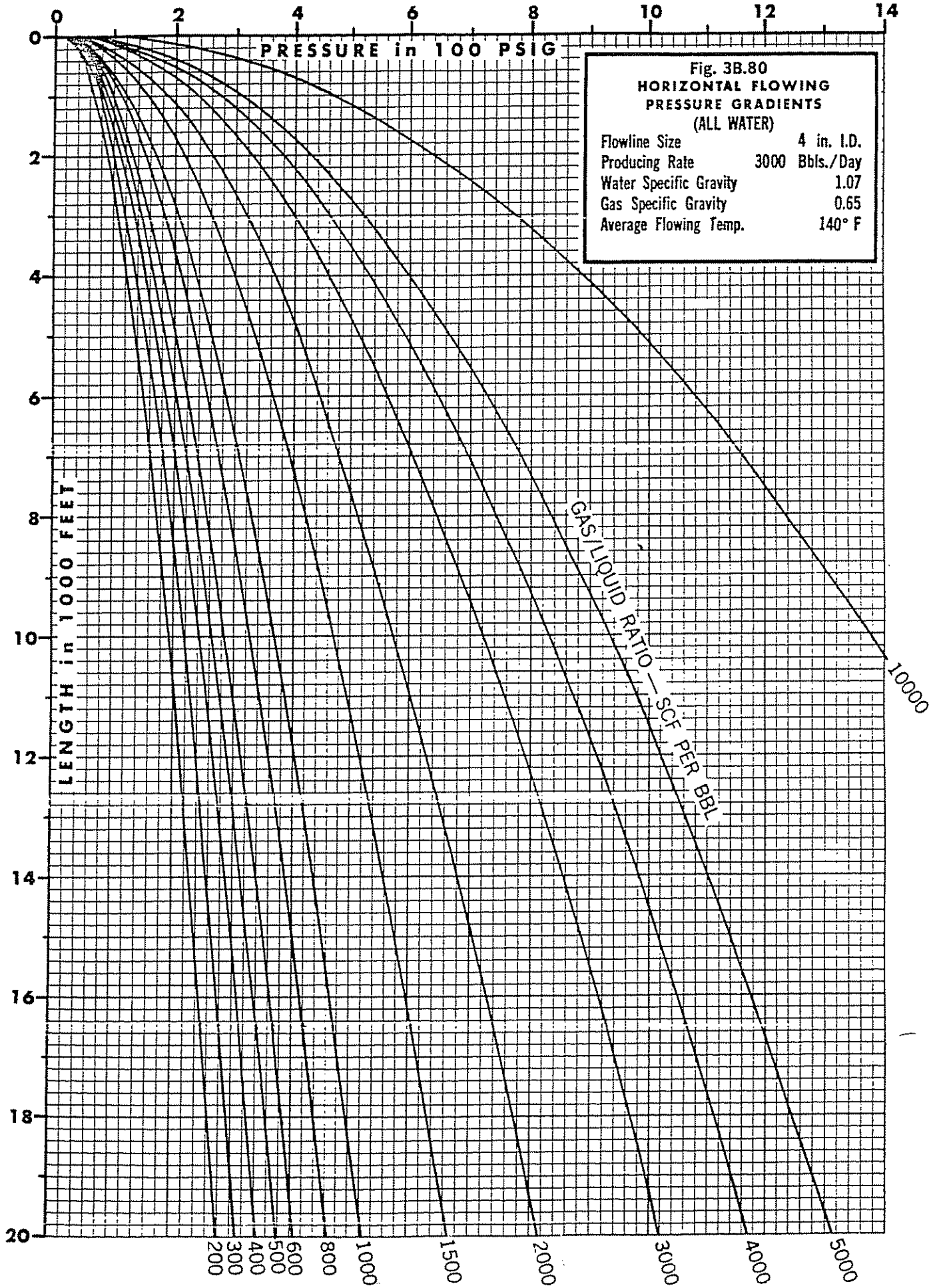
Fig. 3B.78

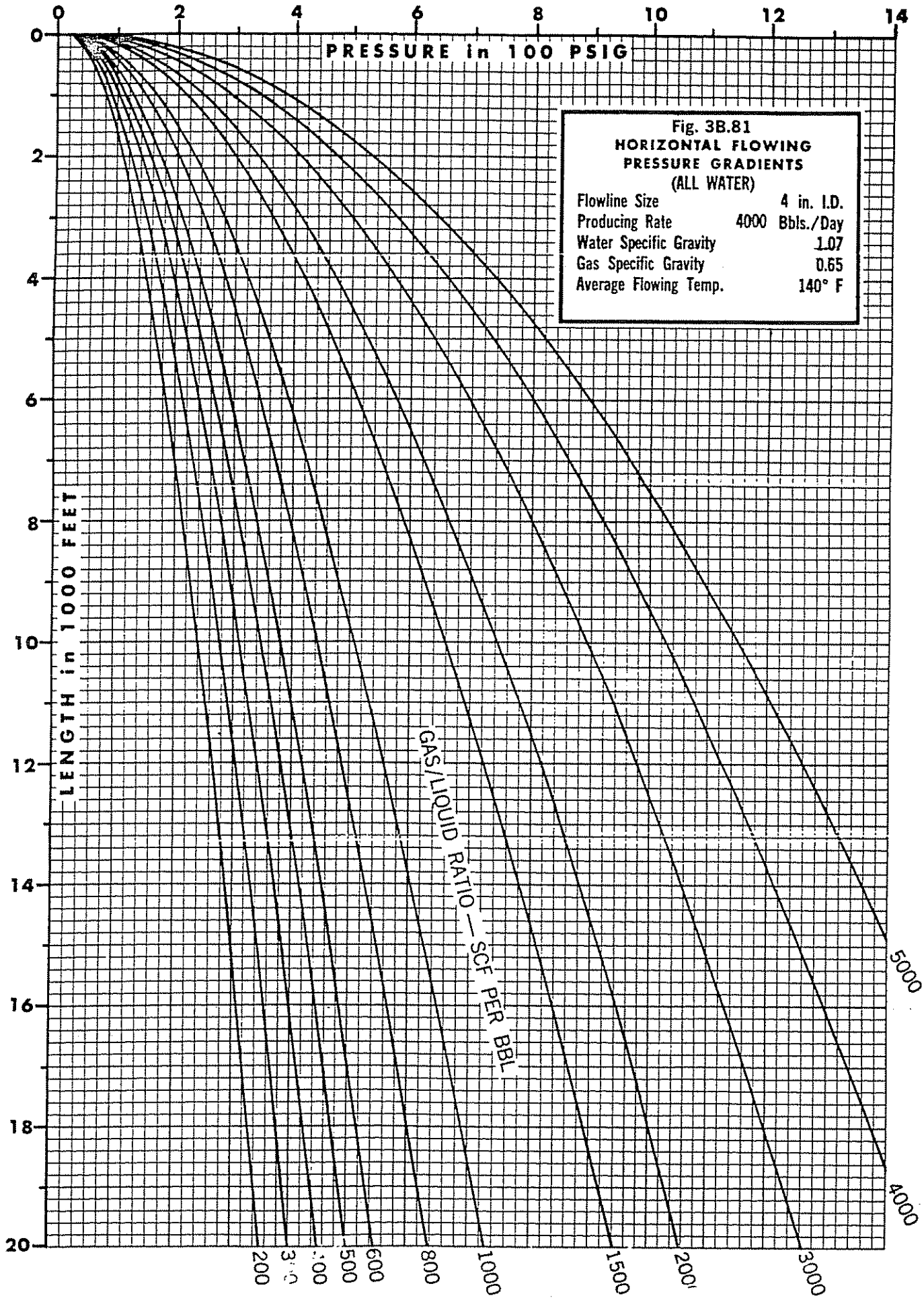
HORIZONTAL FLOWING PRESSURE GRADIENTS

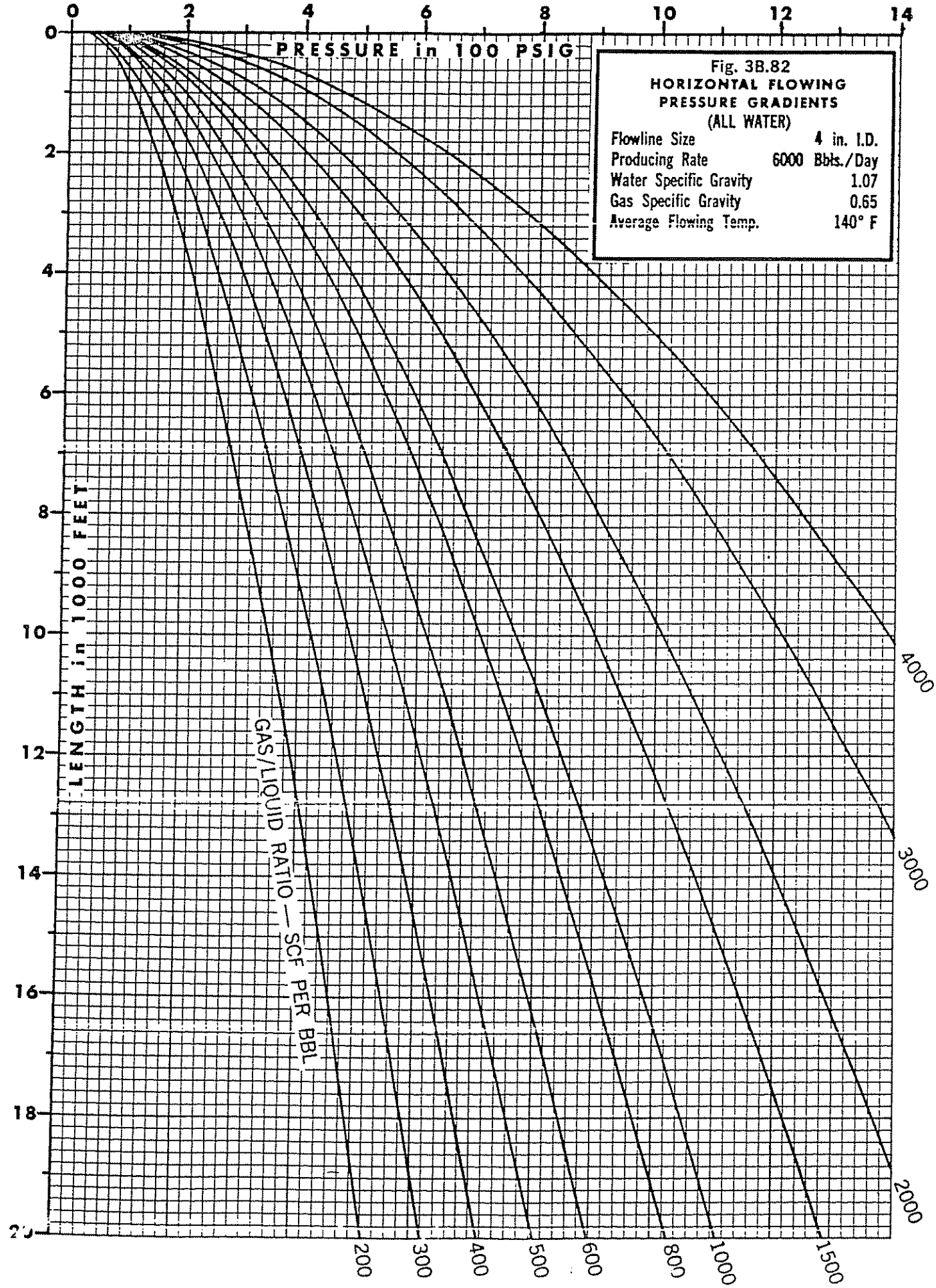
(ALL WATER)

Flowline Size	4 in. I.D.
Producing Rate	1000 Bbls./Day
Water Specific Gravity	1.07
Gas Specific Gravity	0.65
Average Flowing Temp.	120° F


PRESSURE in 100 PSIG
0
2
4
6
8
10
12
14
LENGTH in 1000 FEET
0
2
4
6
8
10
12
14
16
18
20
GAS/LIQUID RATIO — SCF PER BBL
200
300
400
500
600
800
1000
1500
2000
3000
4000
5000
10000
Fig. 3B.79
HORIZONTAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Flowline Size 4 in. I.D.
Producing Rate 2000 Bbls./Day
Water Specific Gravity 1.07
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F


Fig. 3B.80
HORIZONTAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Flowline Size 4 in. I.D.
Producing Rate 3000 Bbls./Day
Water Specific Gravity 1.07
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
PRESSURE in 100 PSIG
0
2
4
6
8
10
12
14
LENGTH in 1000 FEET
0
2
4
6
8
10
12
14
16
18
20
GAS/LIQUID RATIO — SCF PER BBL
200
300
400
500
600
800
1000
1500
2000
3000
4000
5000
10000


0
2
4
6
8
10
12
14
PRESSURE in 100 PSIG
Fig. 3B.81
HORIZONTAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Flowline Size 4 in. I.D.
Producing Rate 4000 Bbls./Day
Water Specific Gravity 1.07
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F
LENGTH in 1000 FEET
0
2
4
6
8
10
12
14
16
18
20
GAS/LIQUID RATIO — SCF PER BBL
200
300
400
500
600
800
1000
1500
2000
3000
4000
5000


0
2
4
6
8
10
12
14
PRESSURE in 100 PSIG
LENGTH in 1000 FEET
GAS/LIQUID RATIO — SCF PER BBL
200
300
400
500
600
800
1000
1500
2000
3000
4000
Fig. 3B.82
HORIZONTAL FLOWING
PRESSURE GRADIENTS
(ALL WATER)
Flowline Size 4 in. I.D.
Producing Rate 6000 Bbls./Day
Water Specific Gravity 1.07
Gas Specific Gravity 0.65
Average Flowing Temp. 140° F

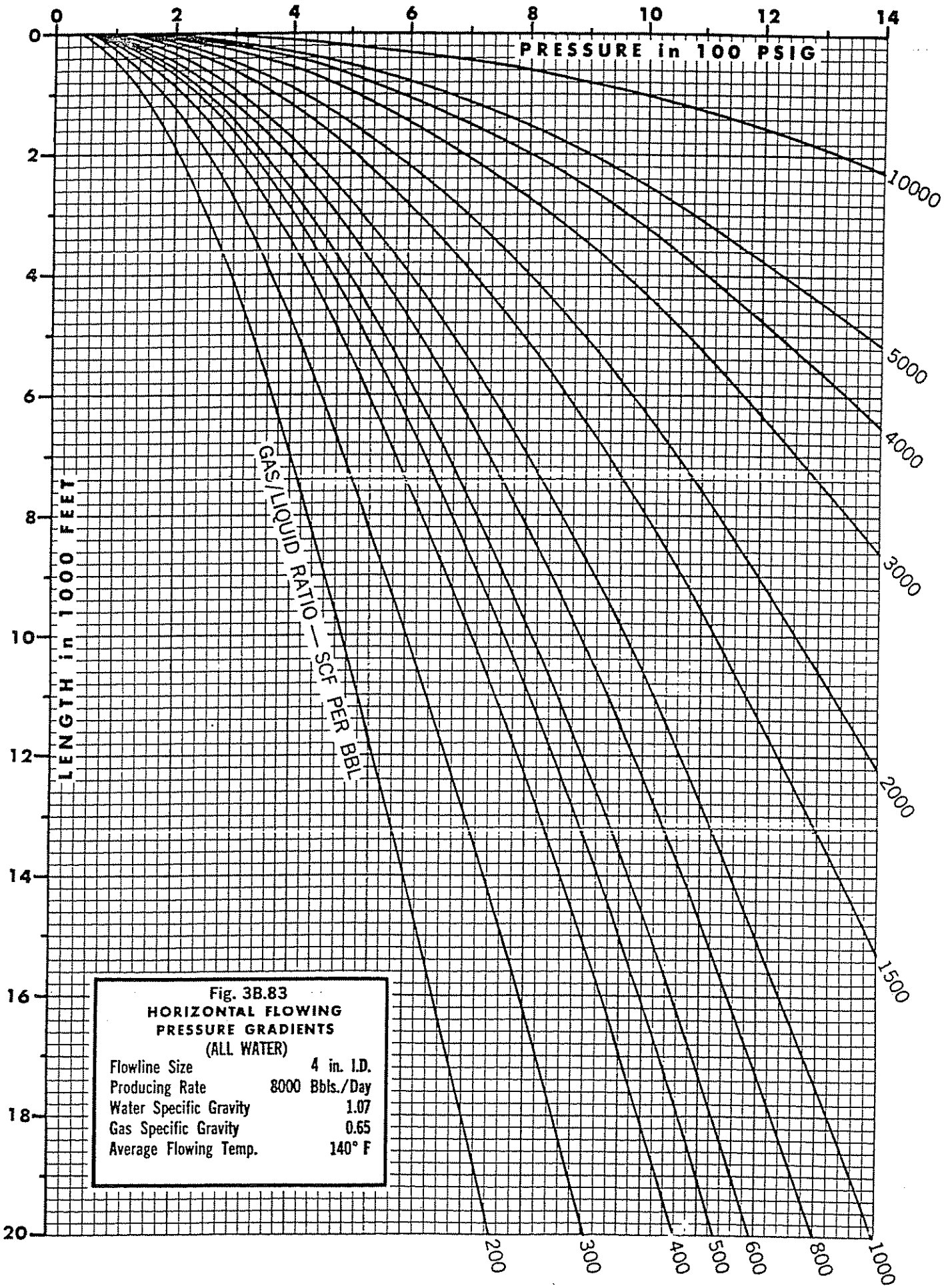


Fig. 3B.83
HORIZONTAL FLOWING PRESSURE GRADIENTS
(ALL WATER)

Flowline Size	4 in. I.D.
Producing Rate	8000 Bbls./Day
Water Specific Gravity	1.07
Gas Specific Gravity	0.65
Average Flowing Temp.	140° F

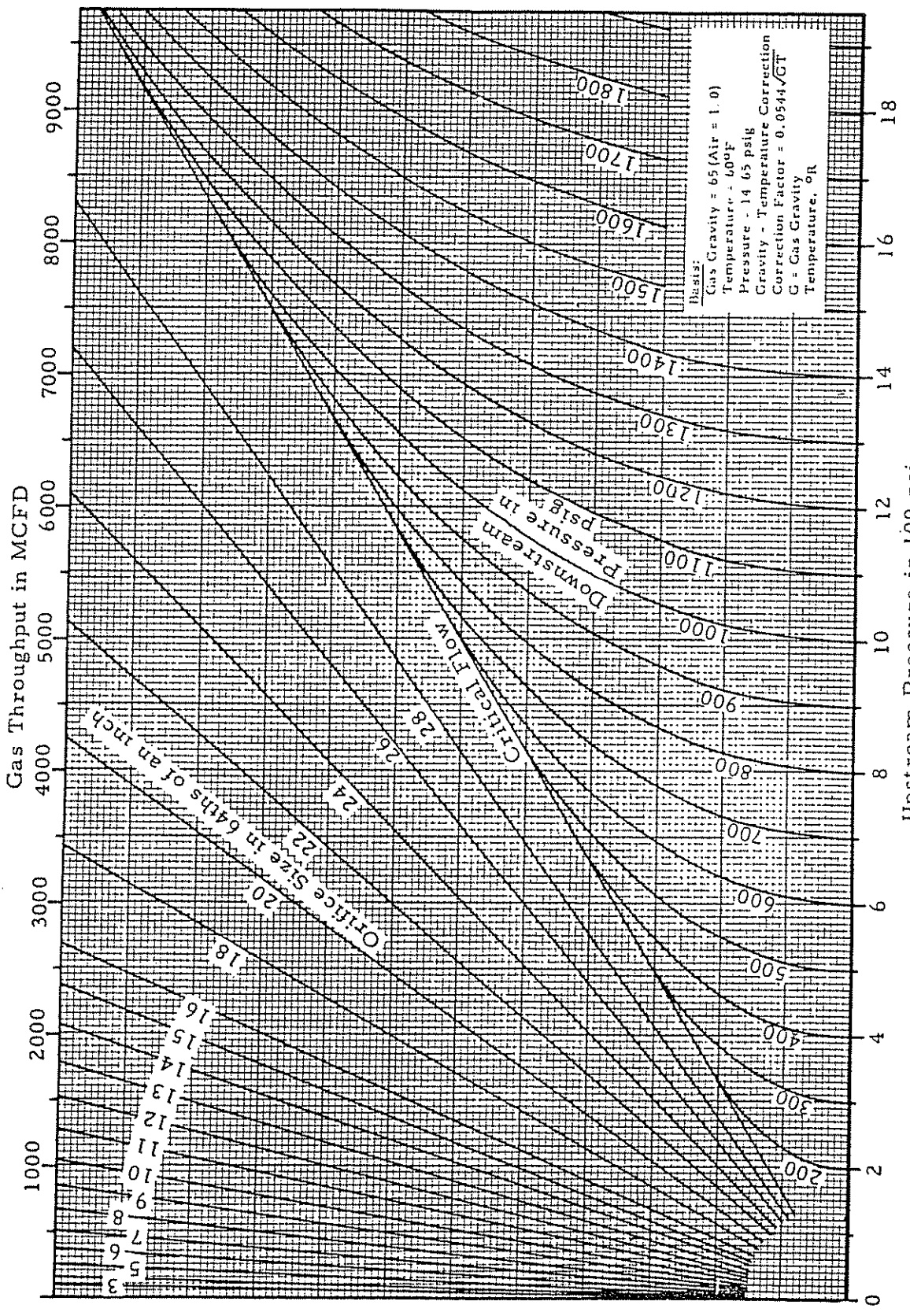


Fig. 3C-1 (a) Gas Passage Chart for Various Orifice Sizes

BASIS:

Correction Factor = $0.0544\sqrt{GT}$

Where: G = Gas Gravity (Air = 1.0)

T = Temperature, °R.

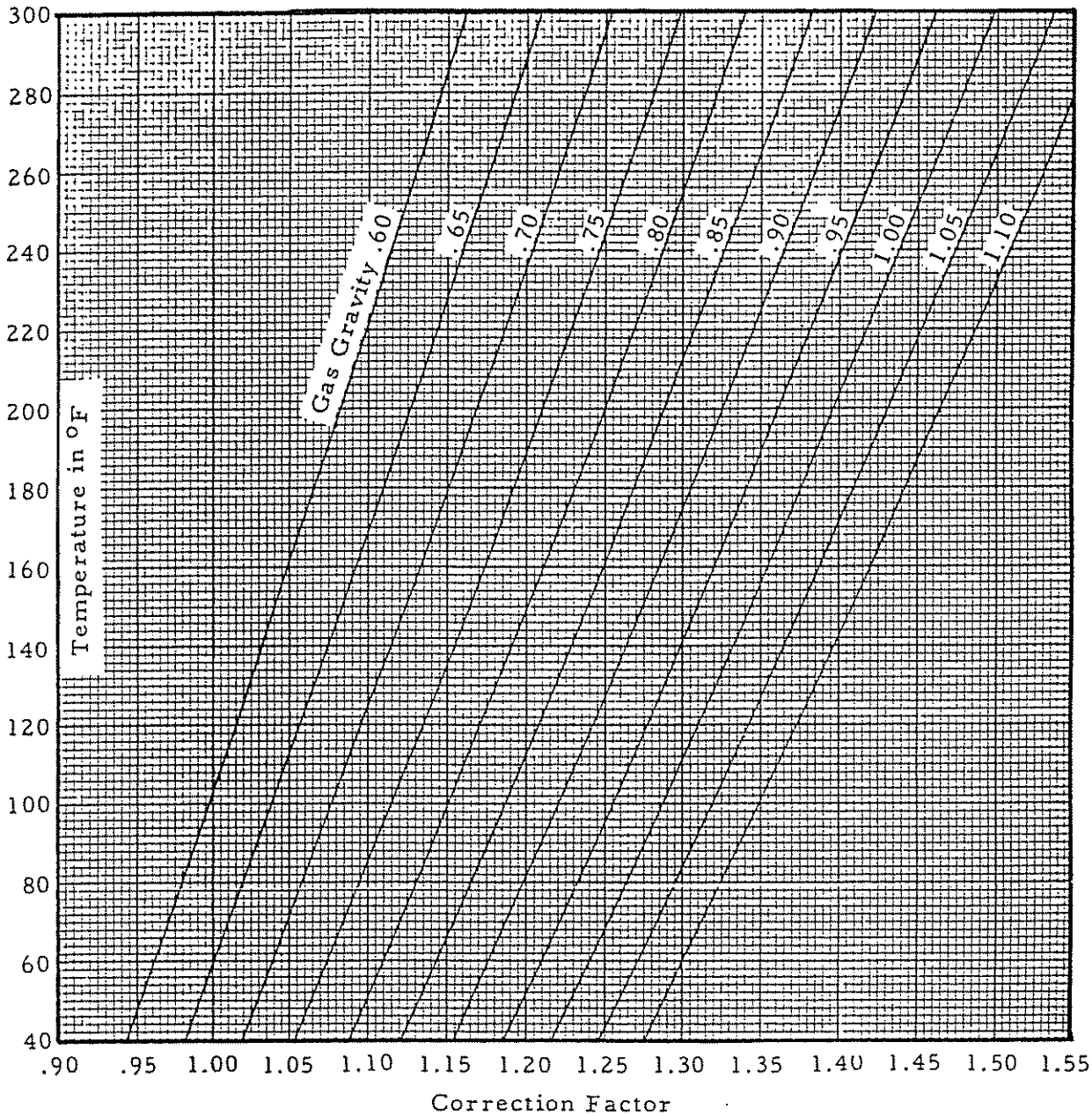
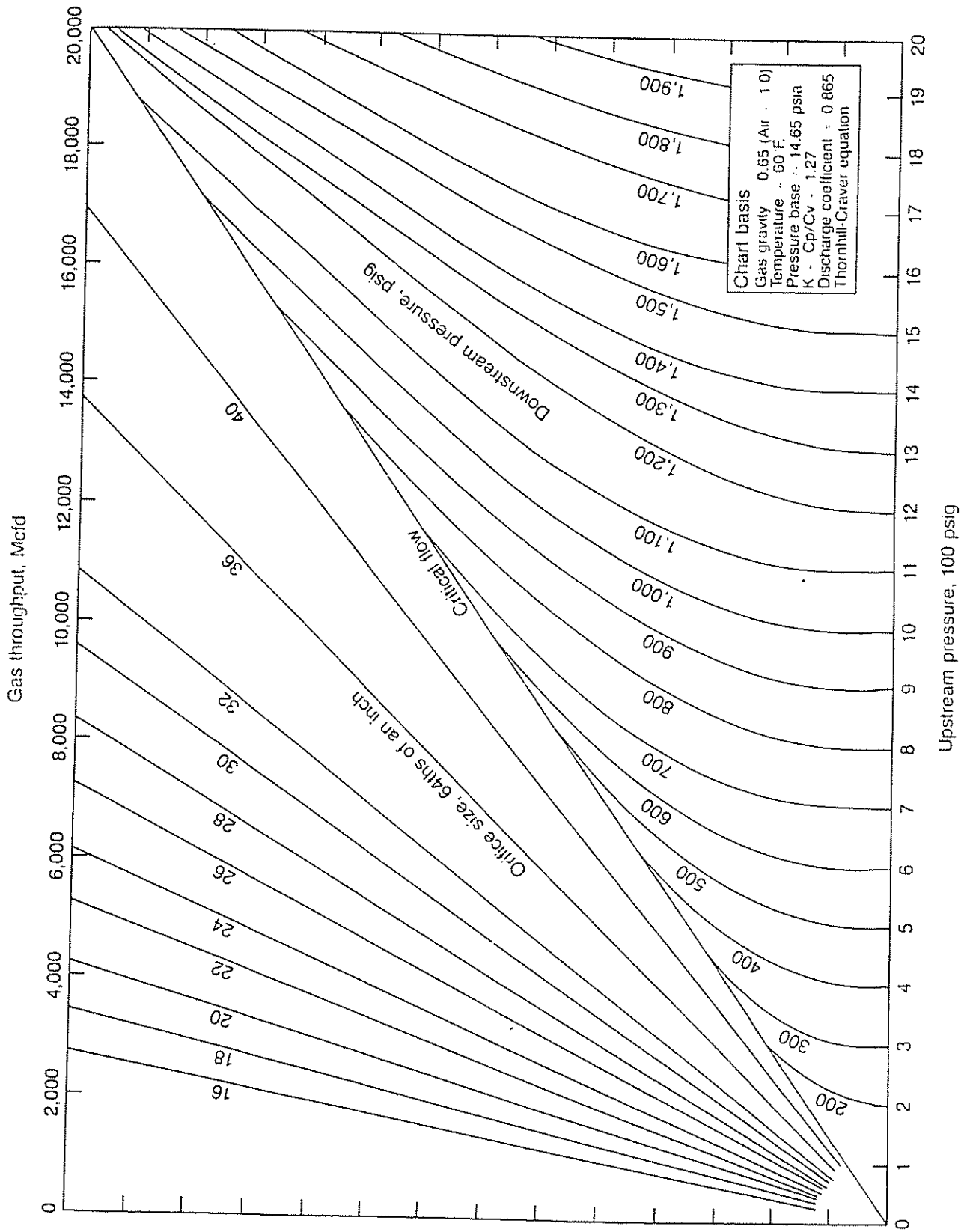


Fig. 3C-1 (b) Correction Factor Chart for Gas Passage Charts


Gas throughput, Mcfd
Upstream pressure, 100 psig
Downstream pressure, psig
Orifice size, 64ths of an inch
Critical flow
Chart basis
Gas gravity = 0.65 (Air = 1.0)
Temperature = 60°F.
Pressure base = 14.65 psia
K = Cp/Cv = 1.27
Discharge coefficient = 0.865
Thornhill-Craver equation
1,900
1,800
1,700
1,600
1,500
1,400
1,300
1,200
1,100
1,000
900
800
700
600
500
400
300
200
40
36
32
30
28
26
24
22
20
18
16
0
2,000
4,000
6,000
8,000
10,000
12,000
14,000
16,000
18,000
20,000
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20

EXPLANATION OF FIGURES ENTITLED CHOKE CAPACITY — SQUARE EDGE ORIFICE Figures 3C.2 through 3C.26

These charts are prepared to determine the choke size necessary to pass a required gas volume and may be used in selecting surface injection chokes as well as gas lift valve chokes.

These curves apply to a square edge orifice only, and the gas rate selected from these charts should be multiplied by a factor of 1.16 for a round edge orifice.

The following constants were utilized:

(1) Square edge orifice
(2) Temperature - 60°F.
(3) Gas specific gravity - .65
(4) Gas deviation accounted for
(5) Orifice length ¾" - charts should be accurate for lengths of ¼" - 2"
(6) Ratios of specific heats (C_p/C_v) = 1.256
(7) Gas passage is in mcf/day at 14.7 psia and 60°F.

Note: To correct for round edge orifice multiply values selected from these charts by a factor of 1.16.

TO CORRECT FOR OPERATING TEMPERATURE AND GAS GRAVITY

The correction for gas gravity and temperature is by the following equation:

$$\text{Correction factor} = \sqrt{\frac{\gamma_g(\text{actual})\,(\text{Temp actual})}{\gamma_g(\text{chart})\,(\text{Temp chart})}}$$

$$\text{Correction factor} = \sqrt{\frac{\gamma_g(\text{Temp, °R})}{(.65)(60+460)}}$$

$$= .0544\sqrt{\gamma_g T}$$

where T = °R

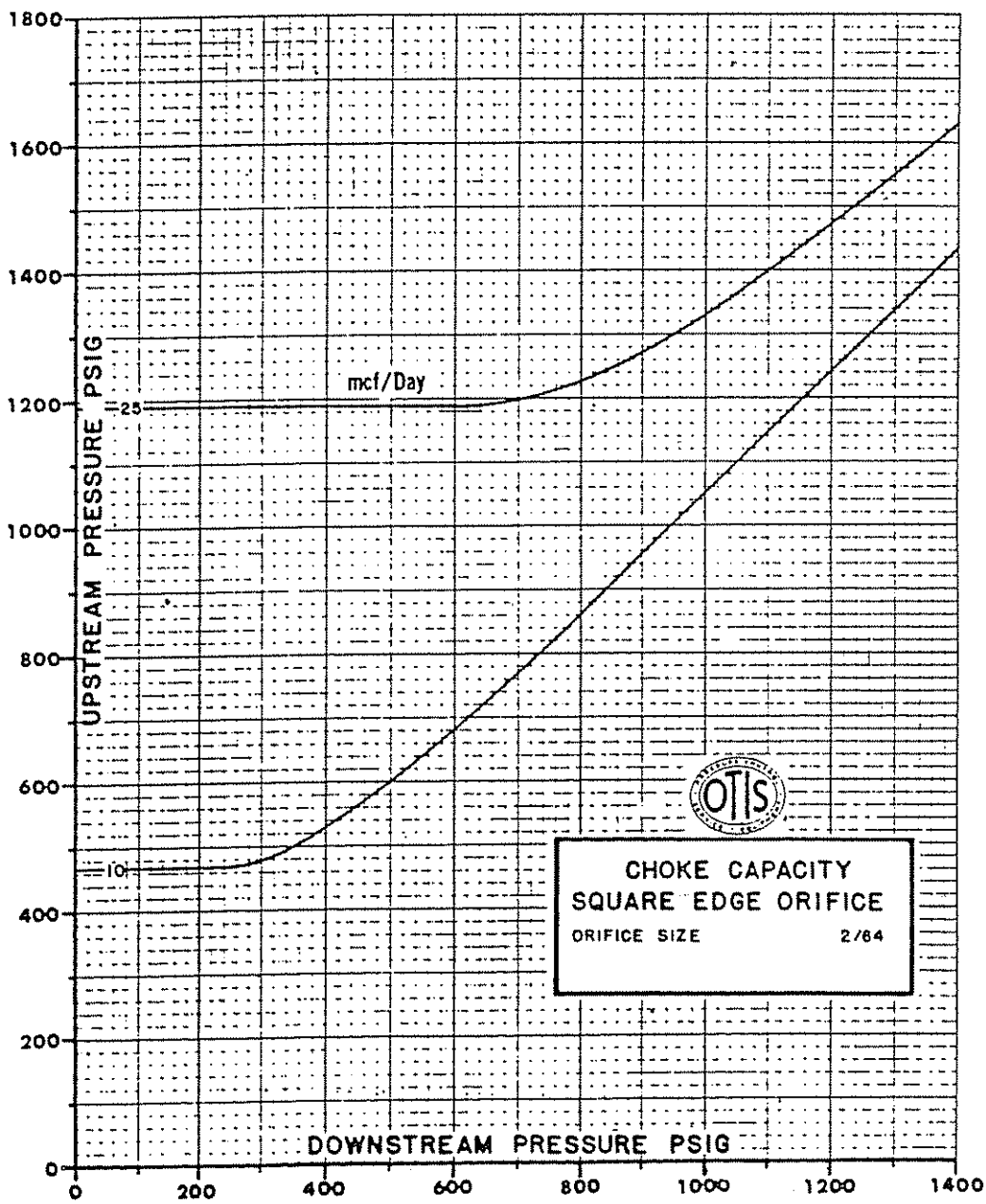


Fig. 3C-2 (Courtesy Otis Engineering Corp.)

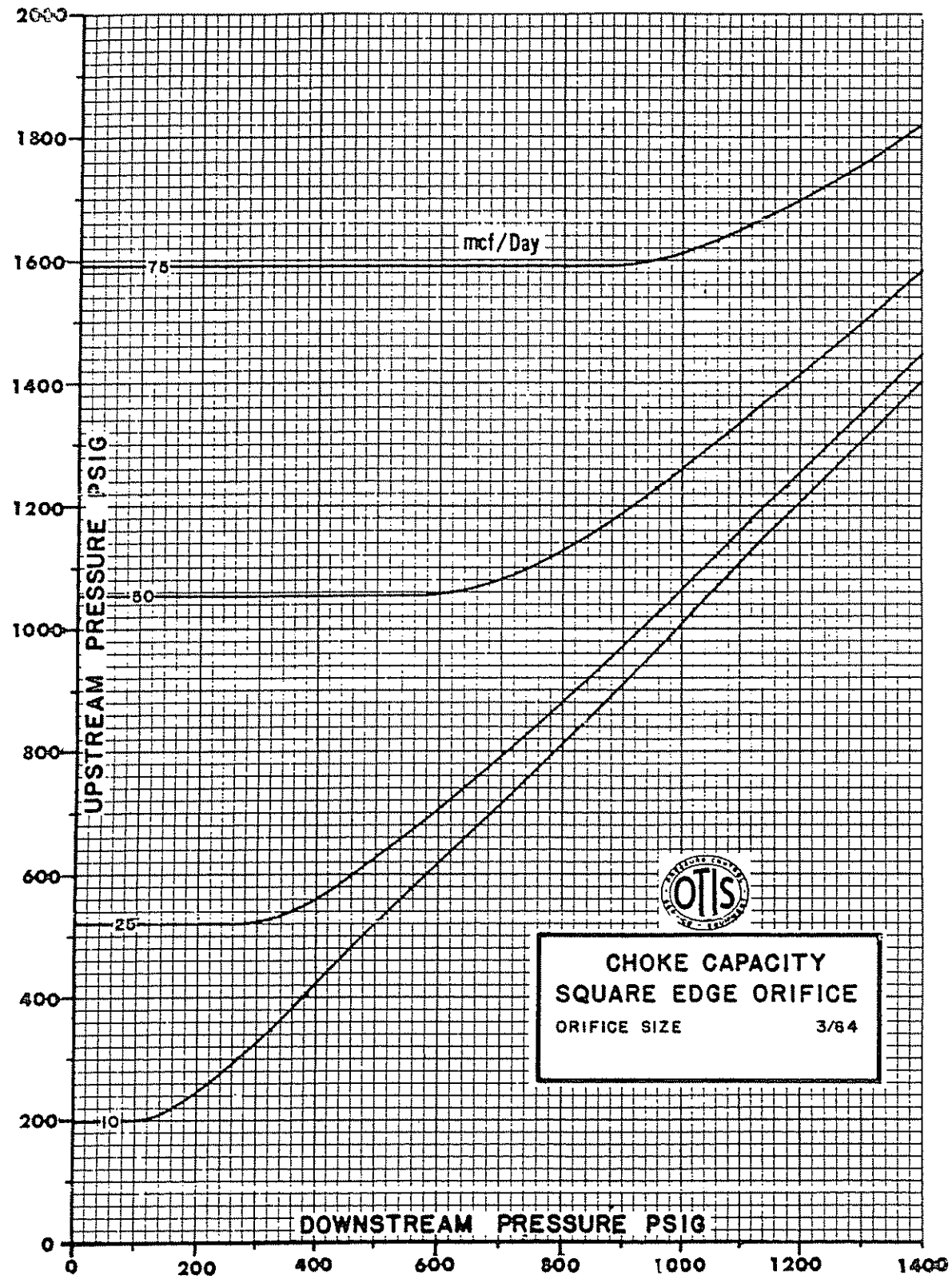


Fig. 3C-3 (Courtesy Otis Engineering Corp.)

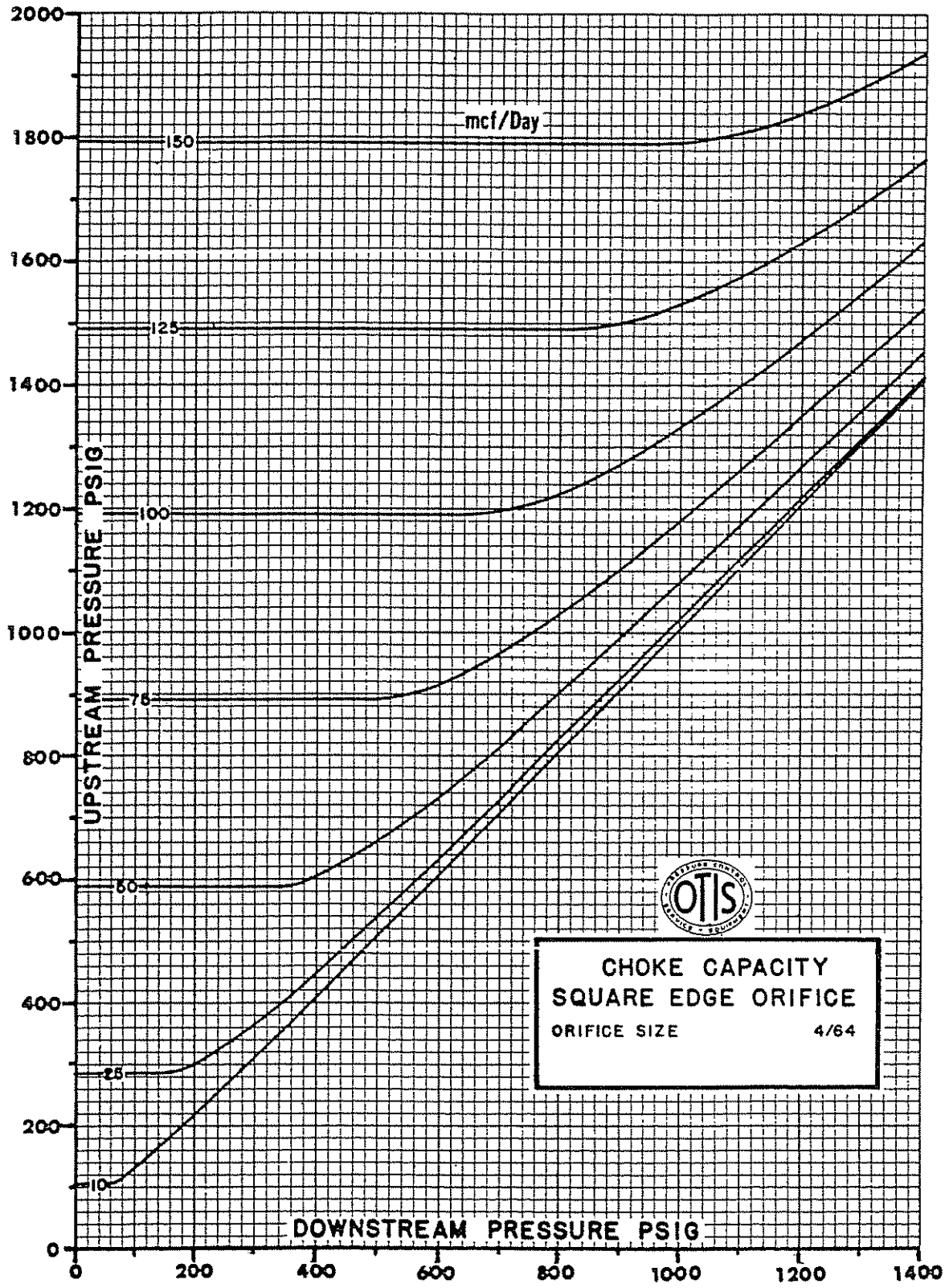


Fig. 3C-4 (Courtesy Otis Engineering Corp.)

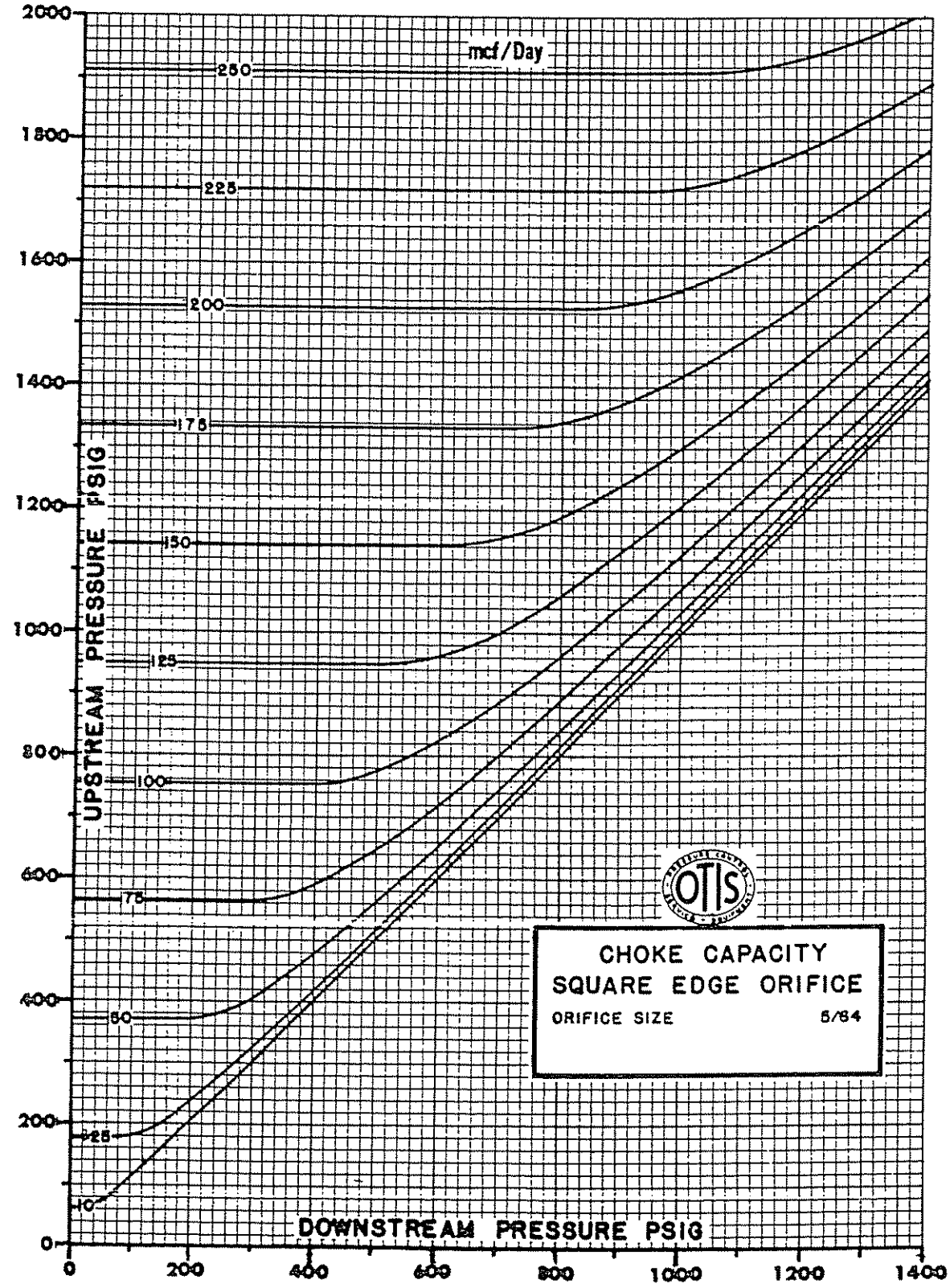


Fig. 3C-5 (Courtesy Otis Engineering Corp.)

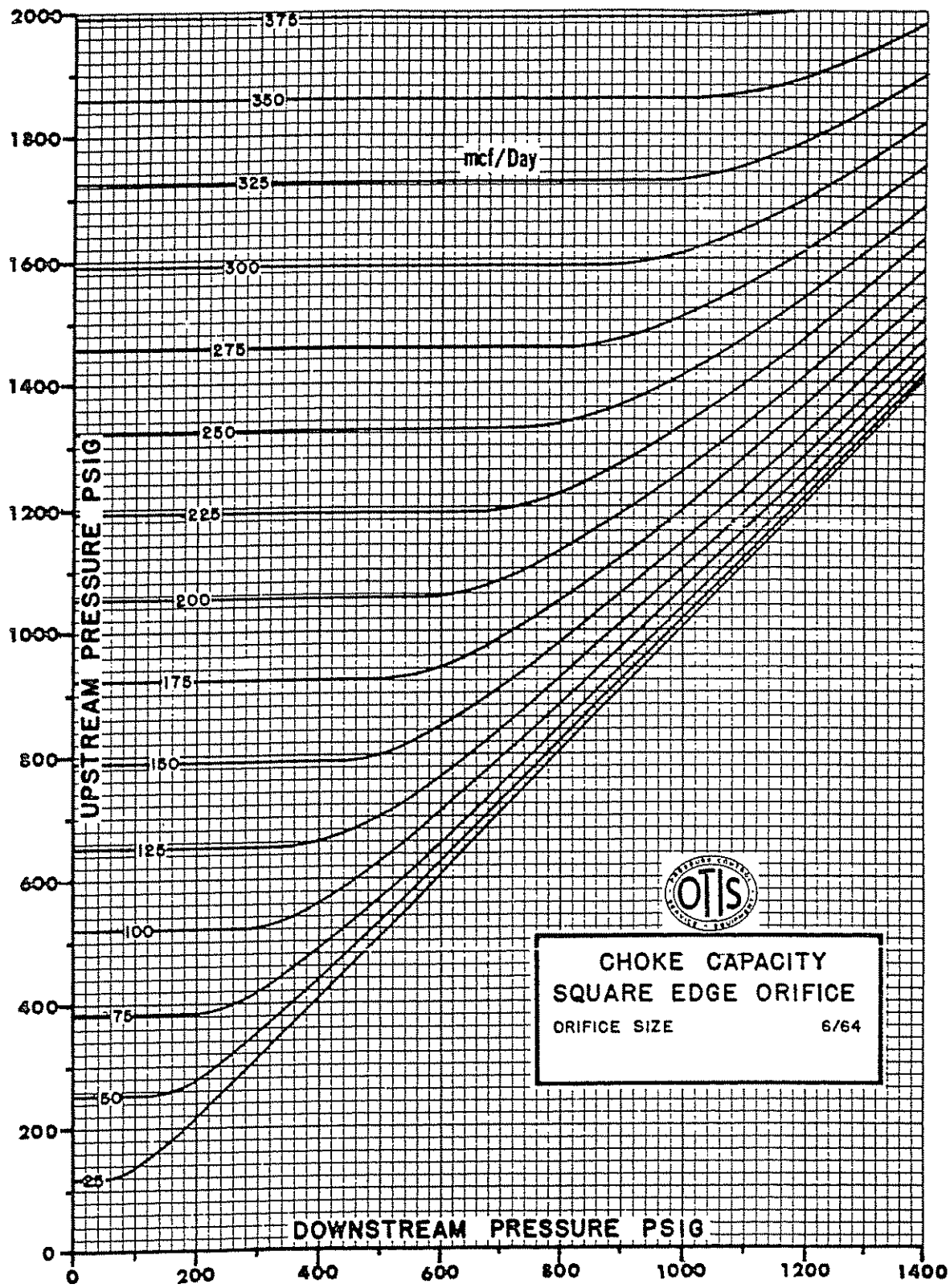


Fig. 3C-6 (Courtesy Otis Engineering Corp.)

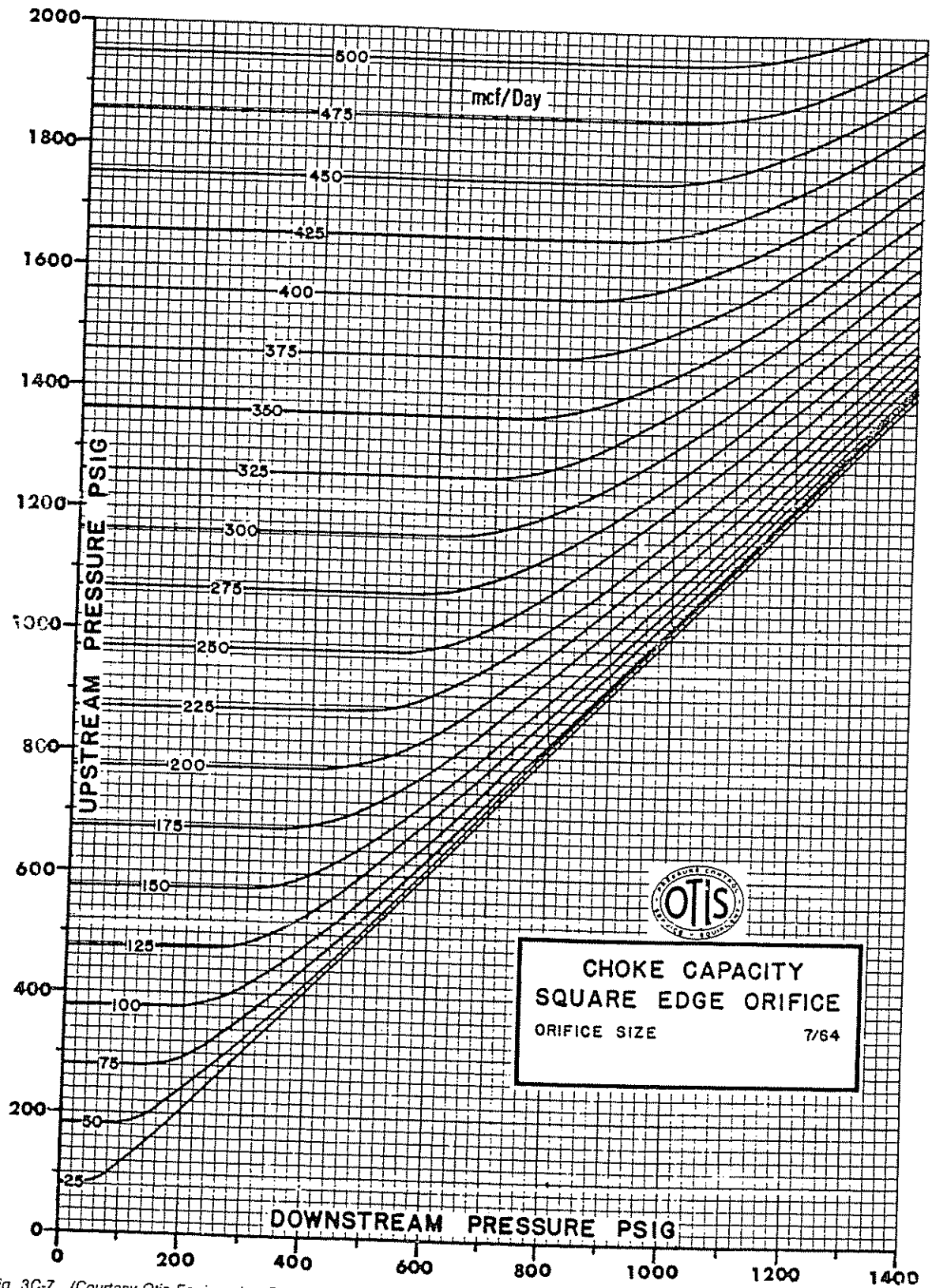


Fig. 3C-7 (Courtesy Otis Engineering Corp.)

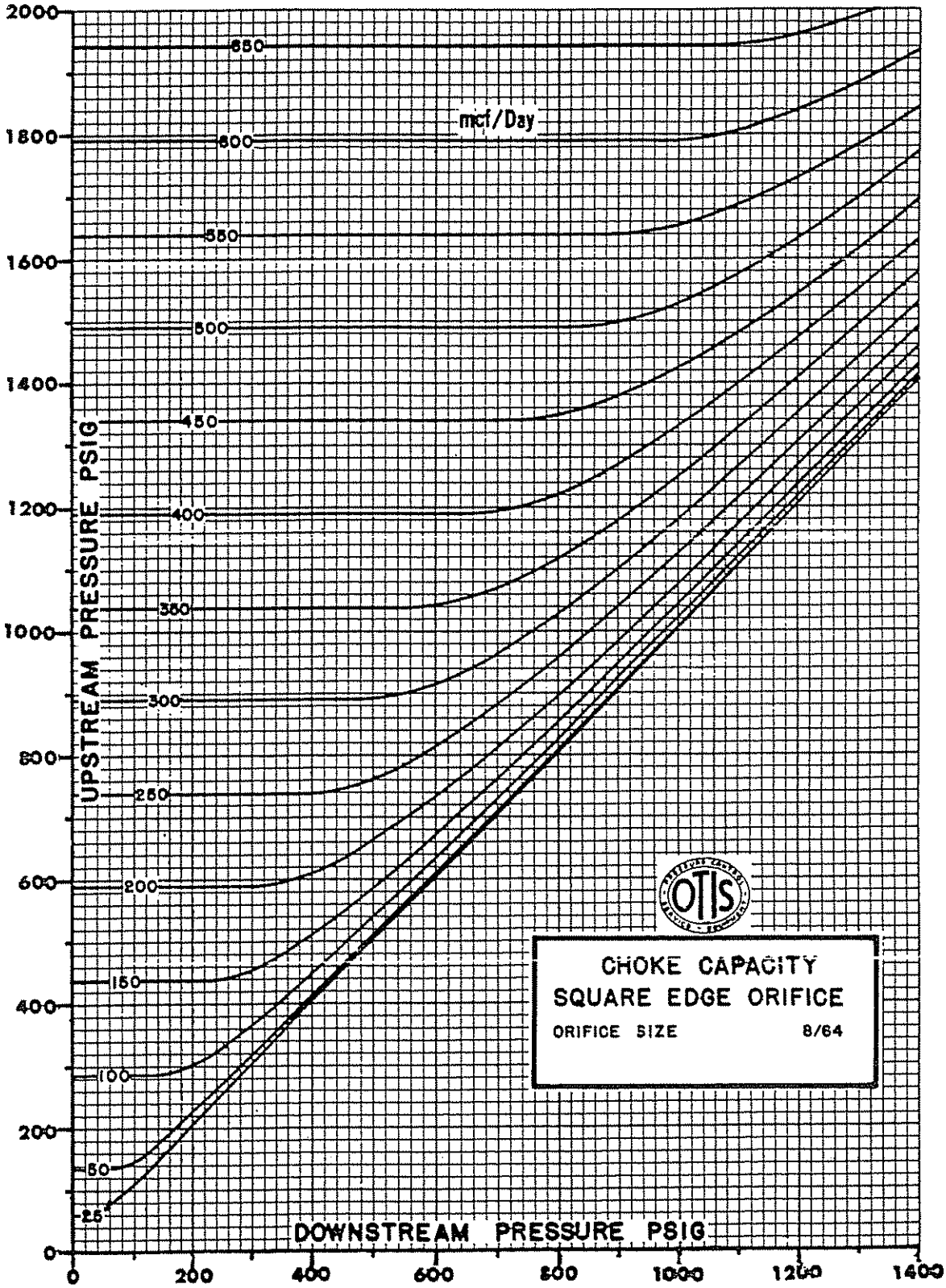


Fig. 3C-8 (Courtesy Otis Engineering Corp.)

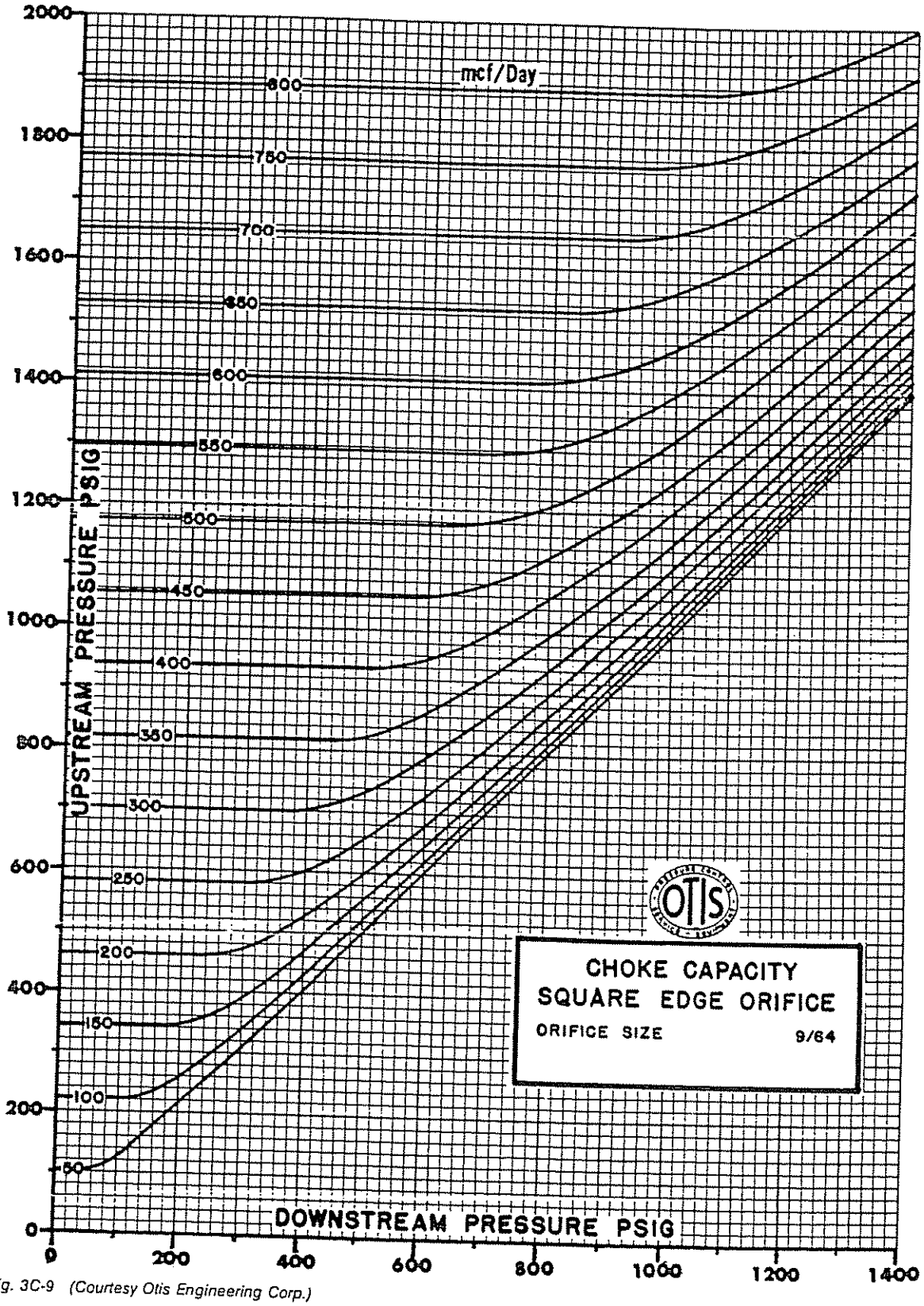


Fig. 3C-9 (Courtesy Otis Engineering Corp.)

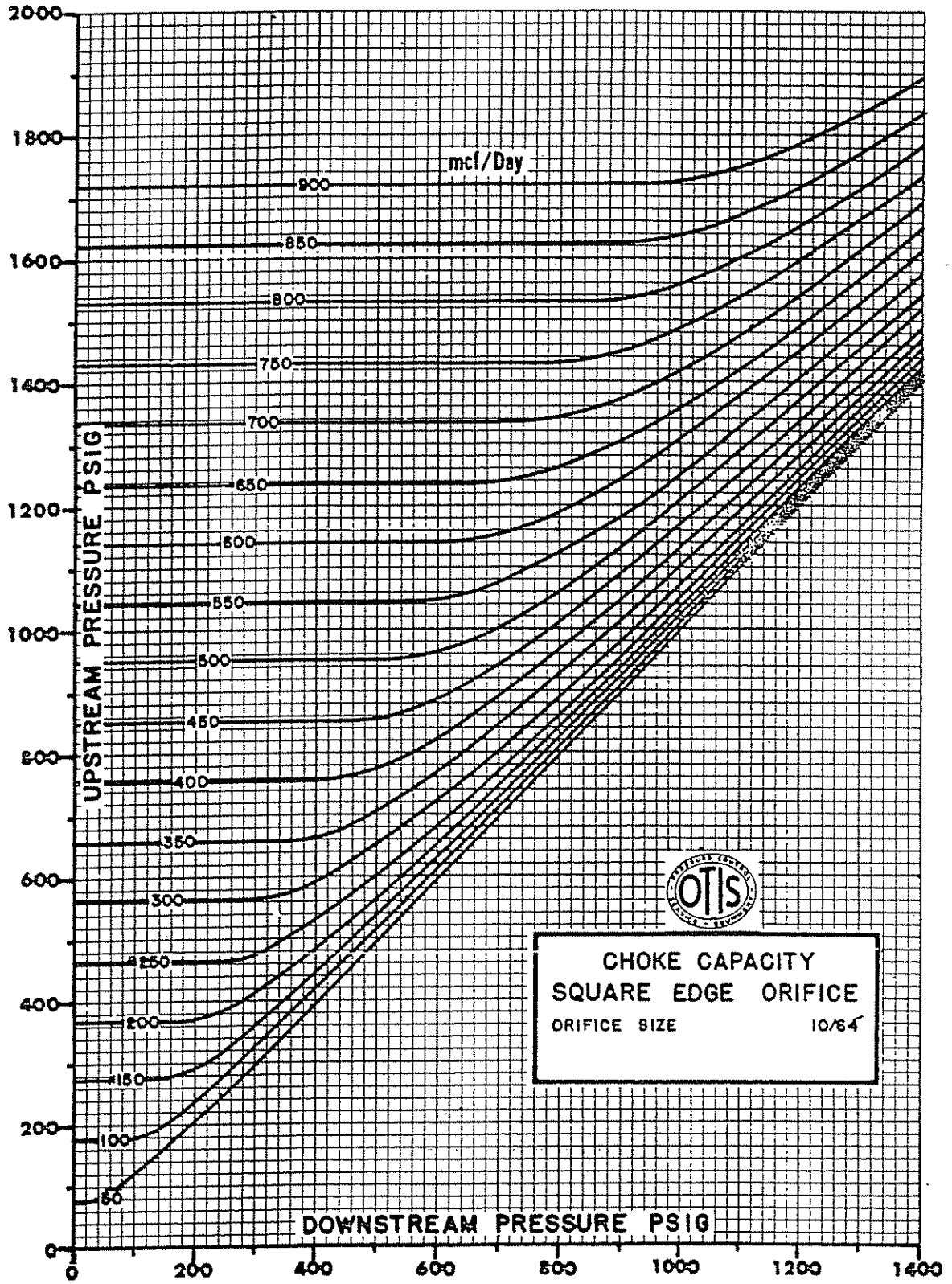


Fig. 3C-10 (Courtesy Otis Engineering Corp.)

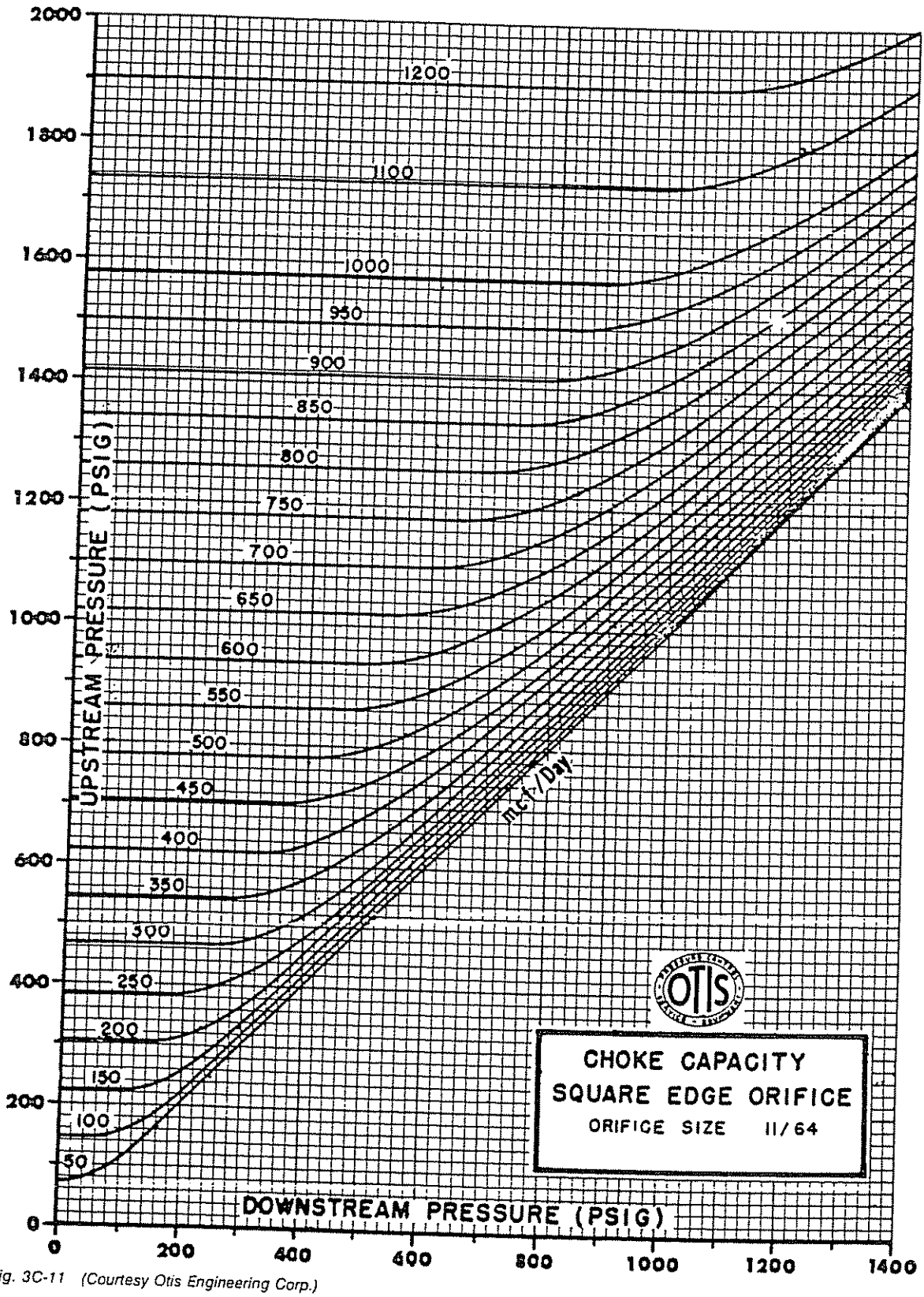


Fig. 3C-11 (Courtesy Otis Engineering Corp.)

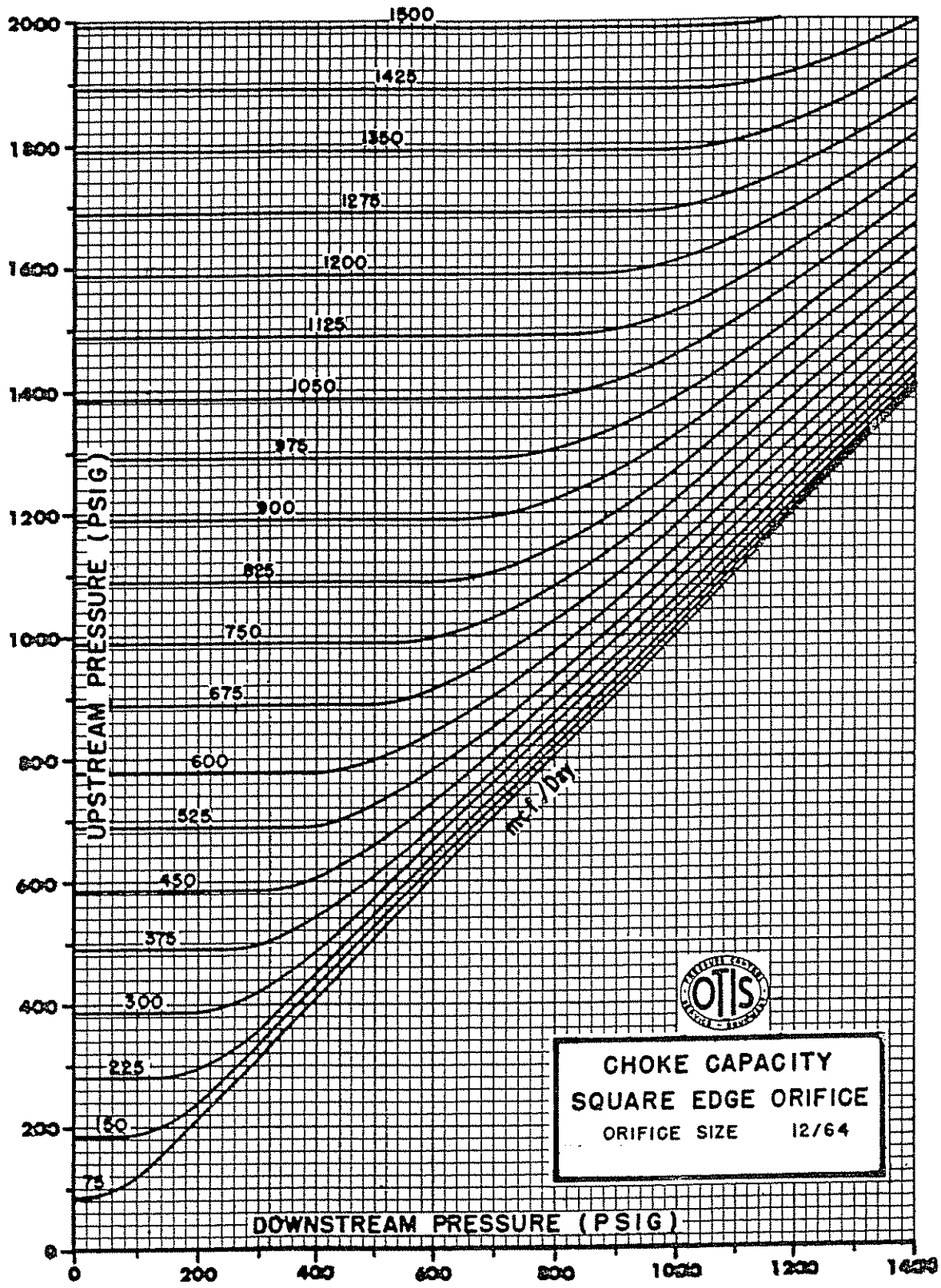


Fig. 3C-12 (Courtesy Otis Engineering Corp.)

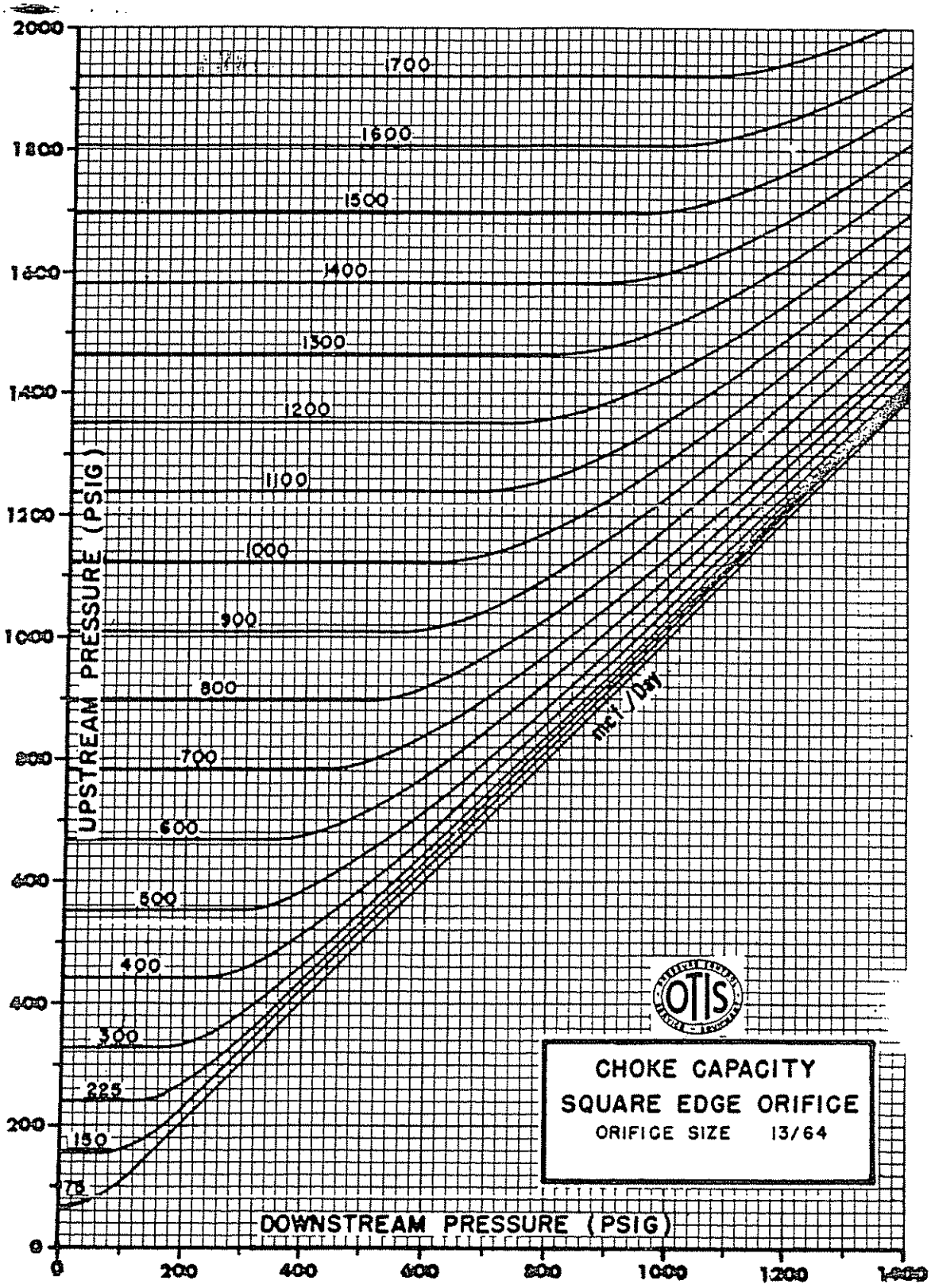


Fig. 3C-13 (Courtesy Otis Engineering Corp.)

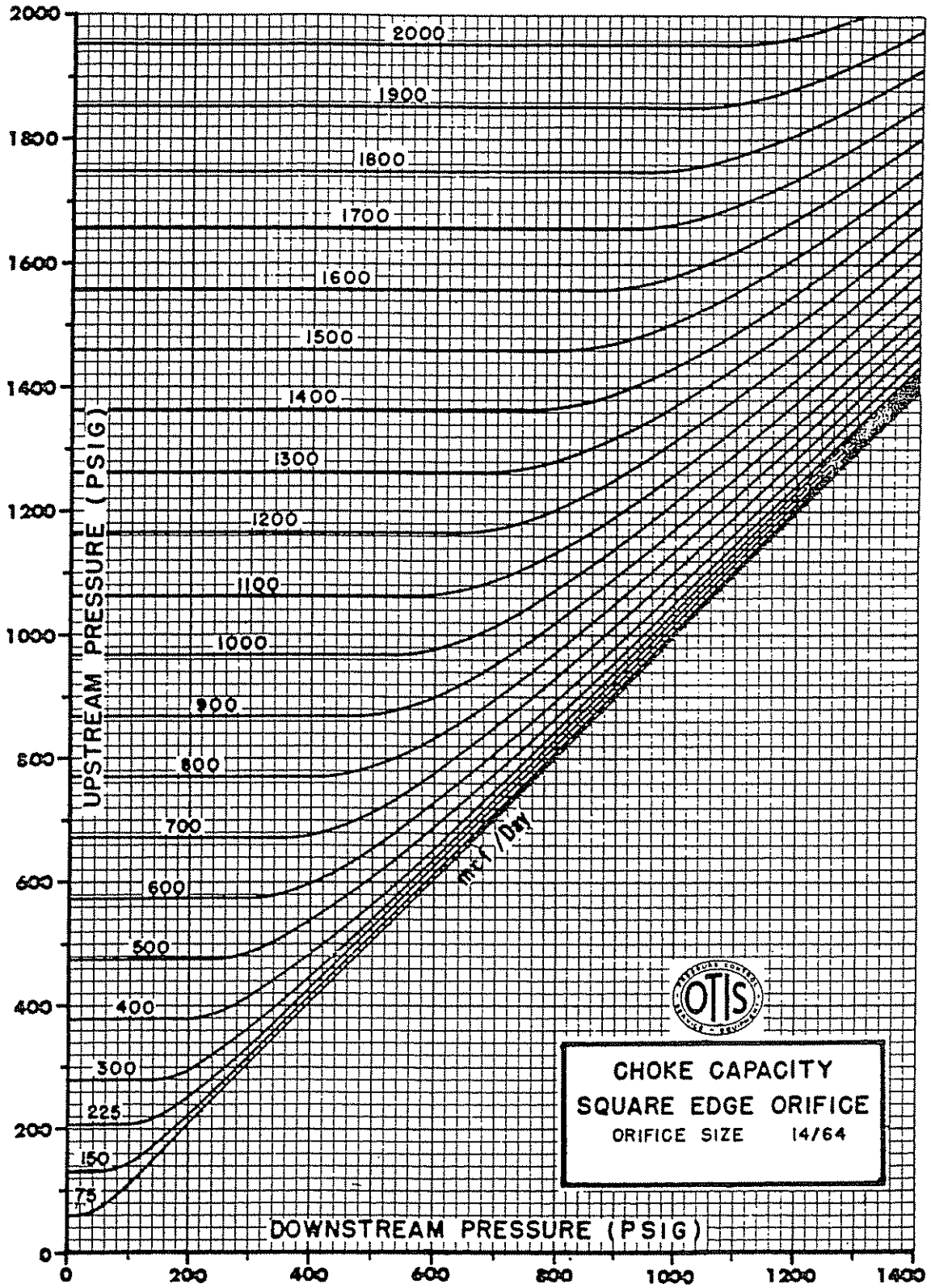


Fig. 3C-14 (Courtesy Otis Engineering Corp.)

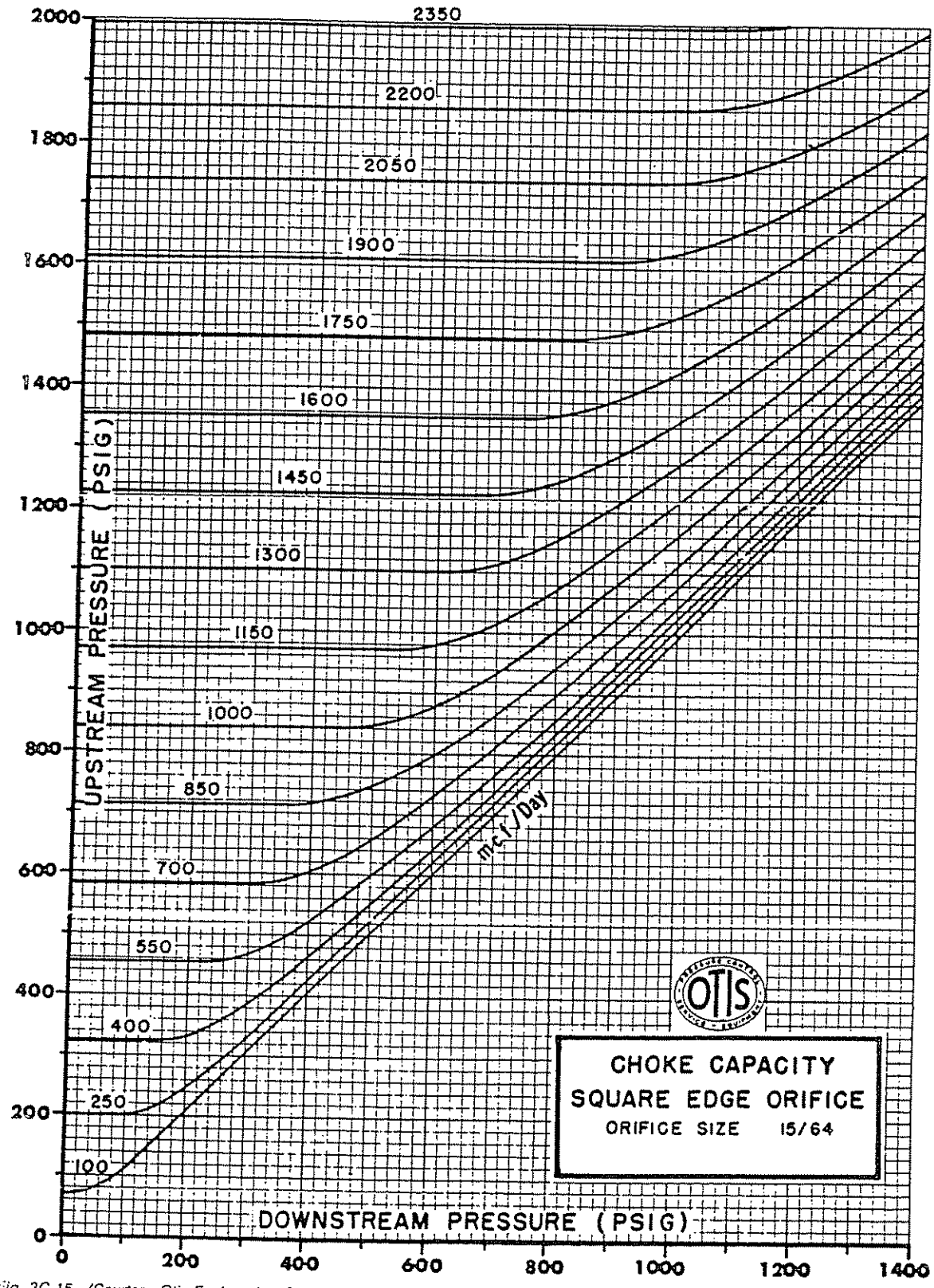


Fig. 3C-15 (Courtesy Otis Engineering Corp.)

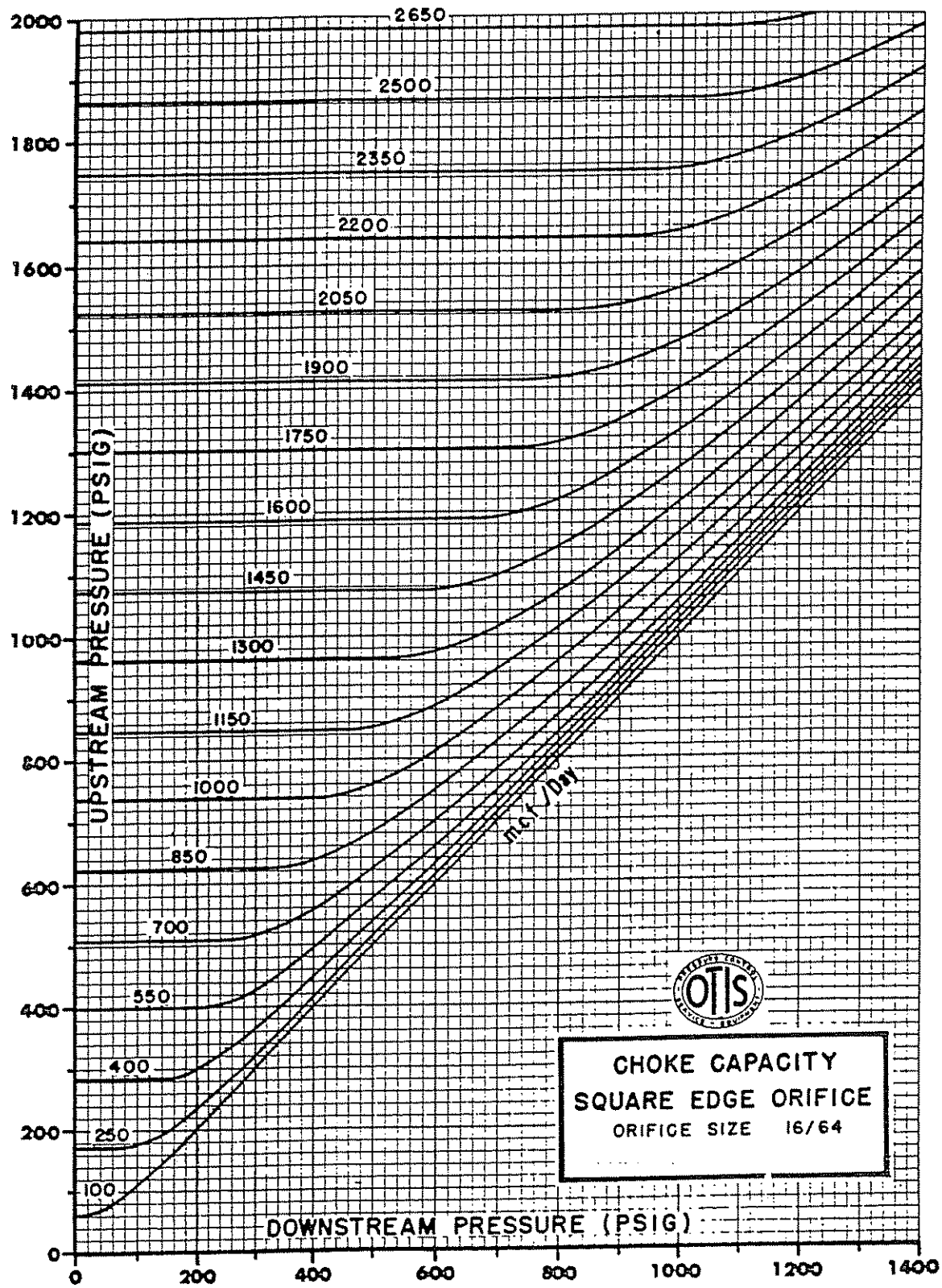


Fig. 3C-16 (Courtesy Otis Engineering Corp.)

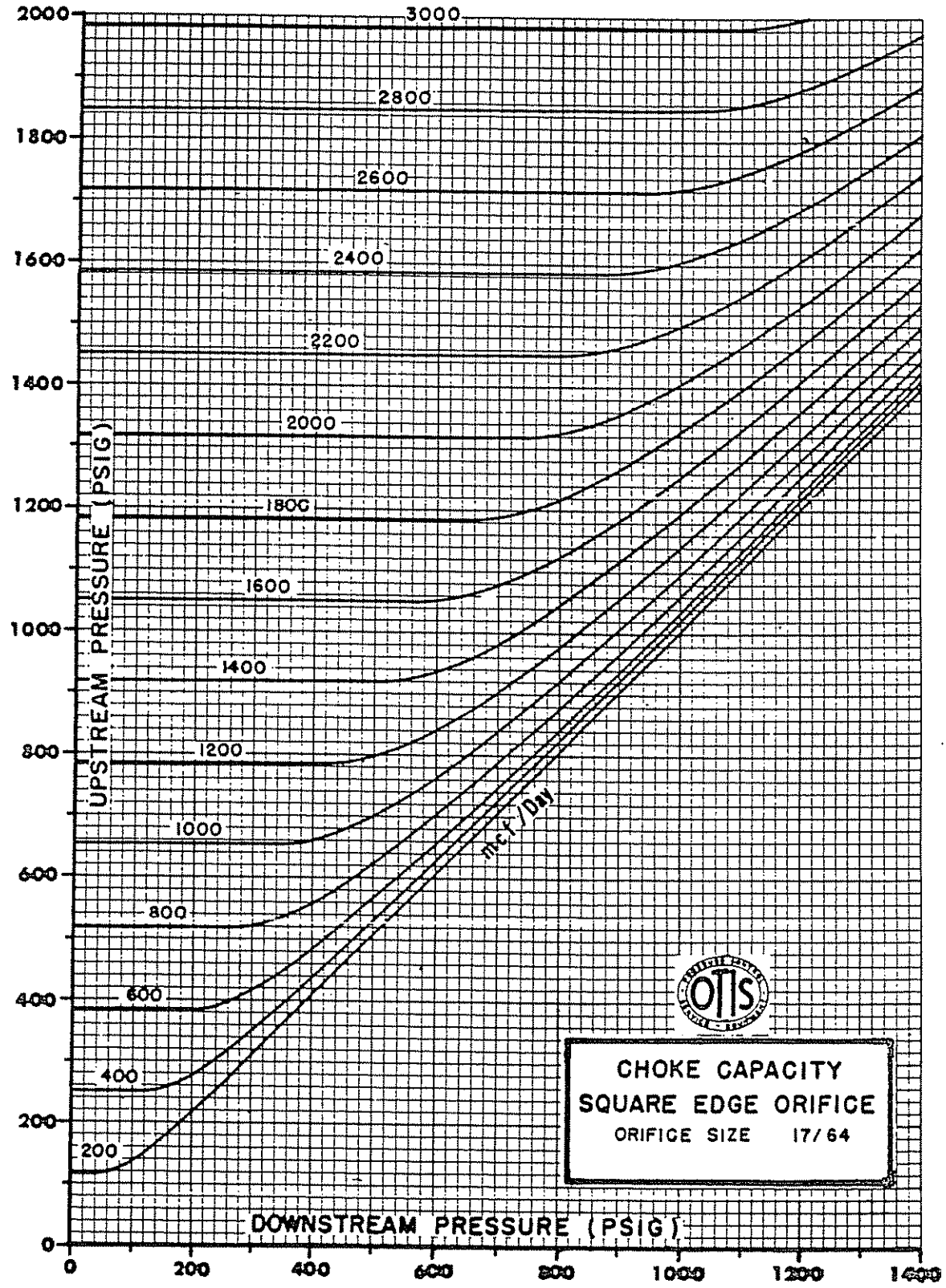


Fig. 3C-17 (Courtesy Otis Engineering Corp.)

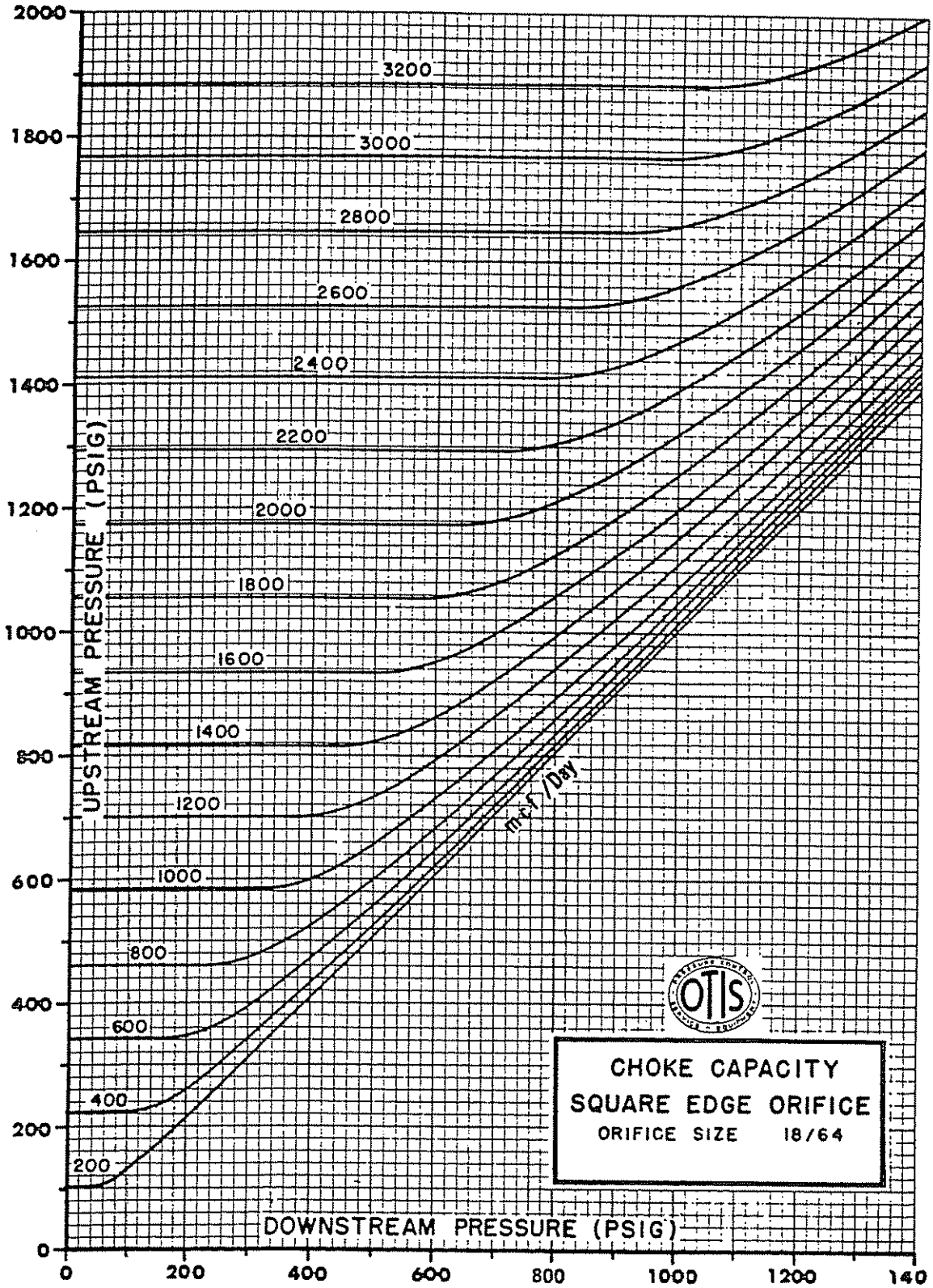


Fig. 3C-18 (Courtesy Otis Engineering Corp.)

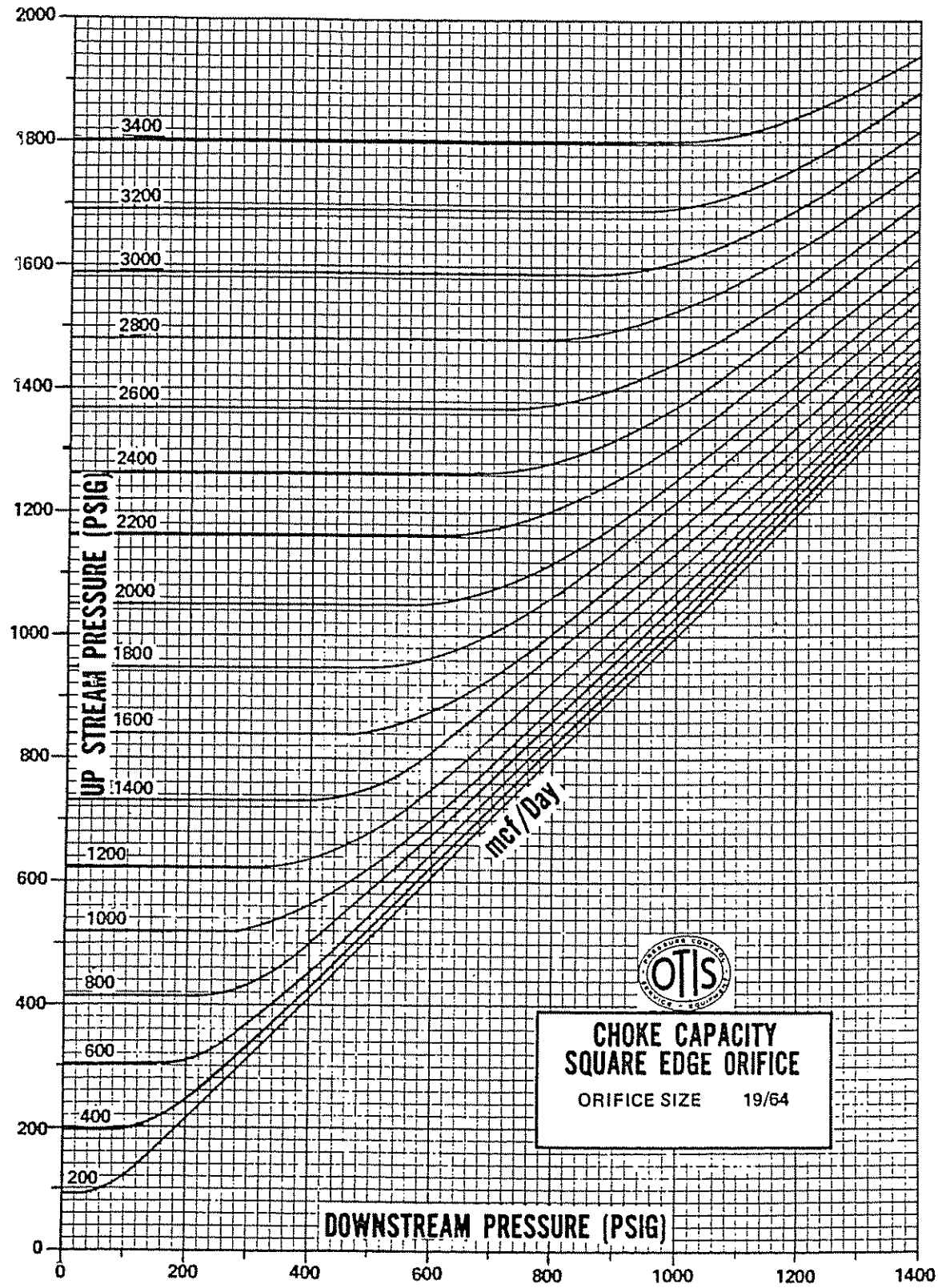


Fig. 3C-19 (Courtesy Otis Engineering Corp.)

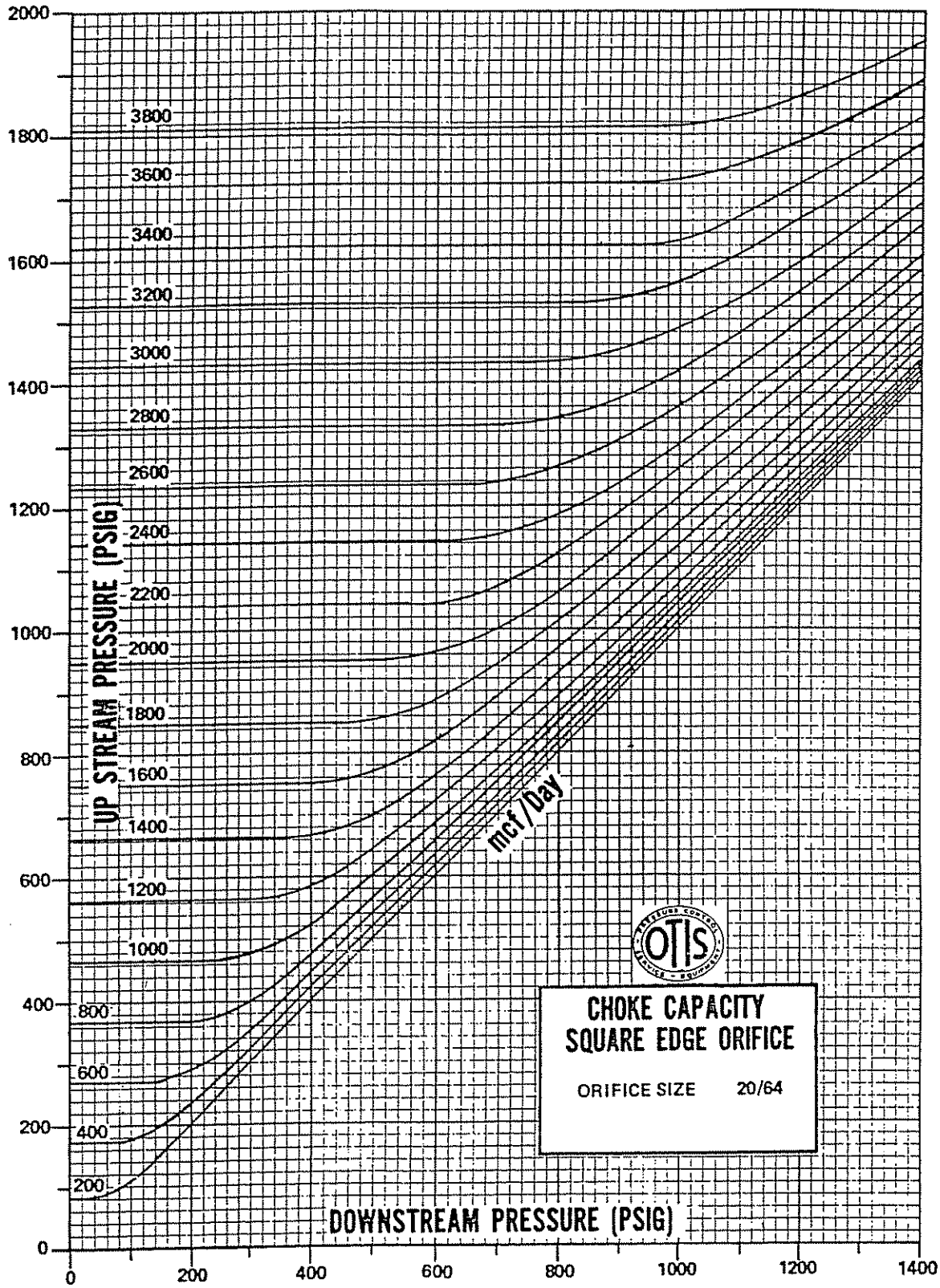


Fig. 3C-20 (Courtesy Otis Engineering Corp.)

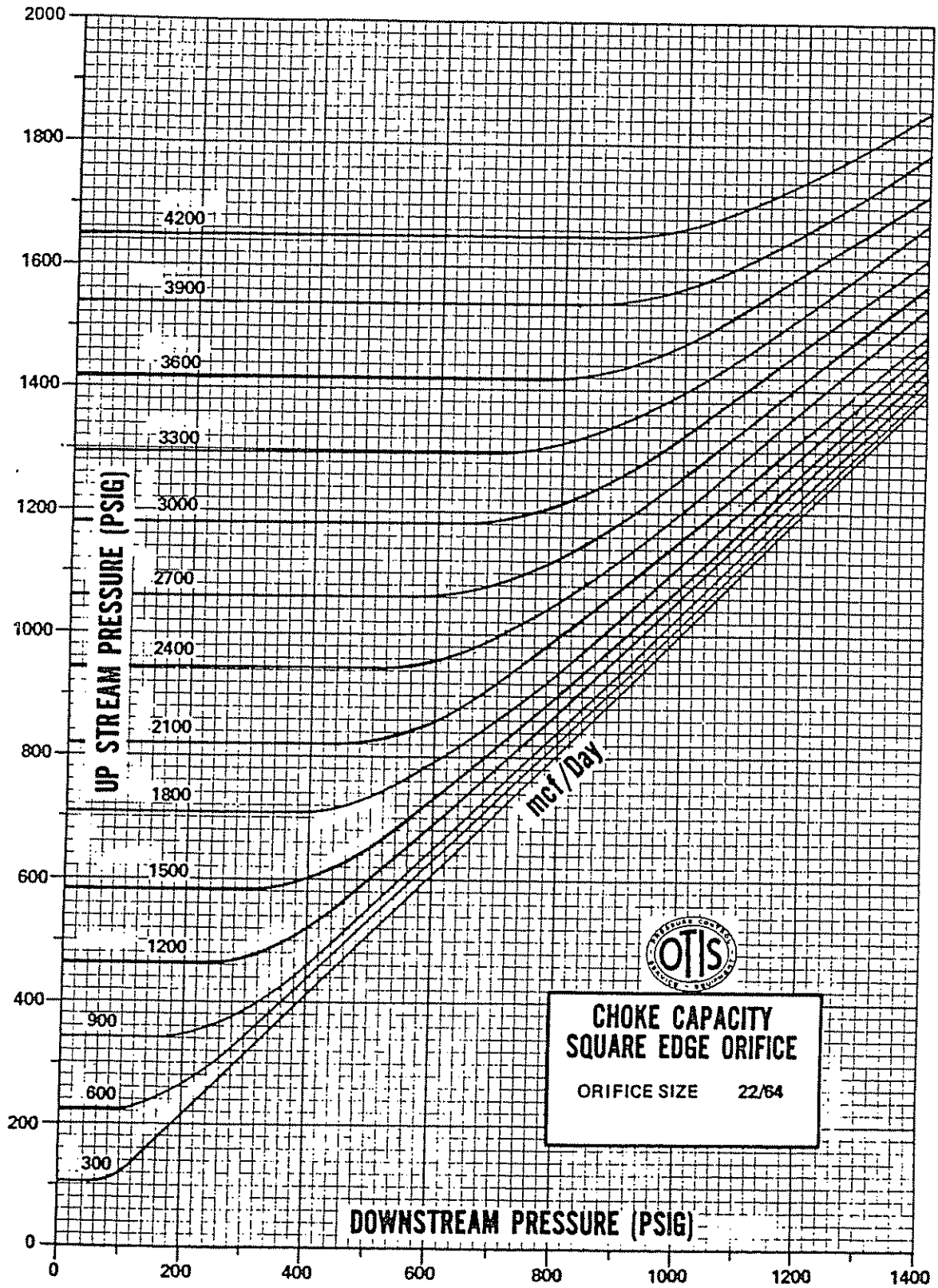


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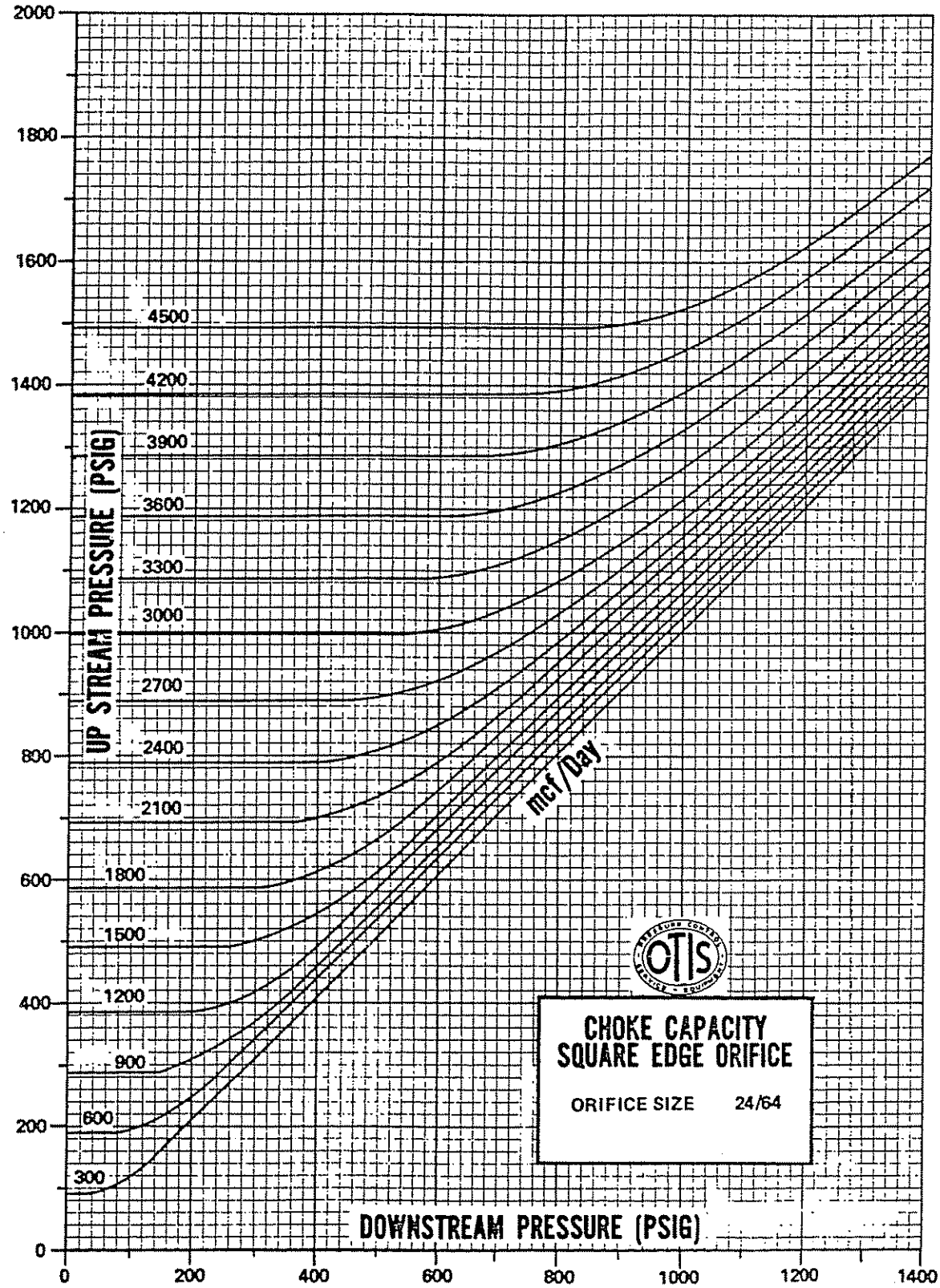


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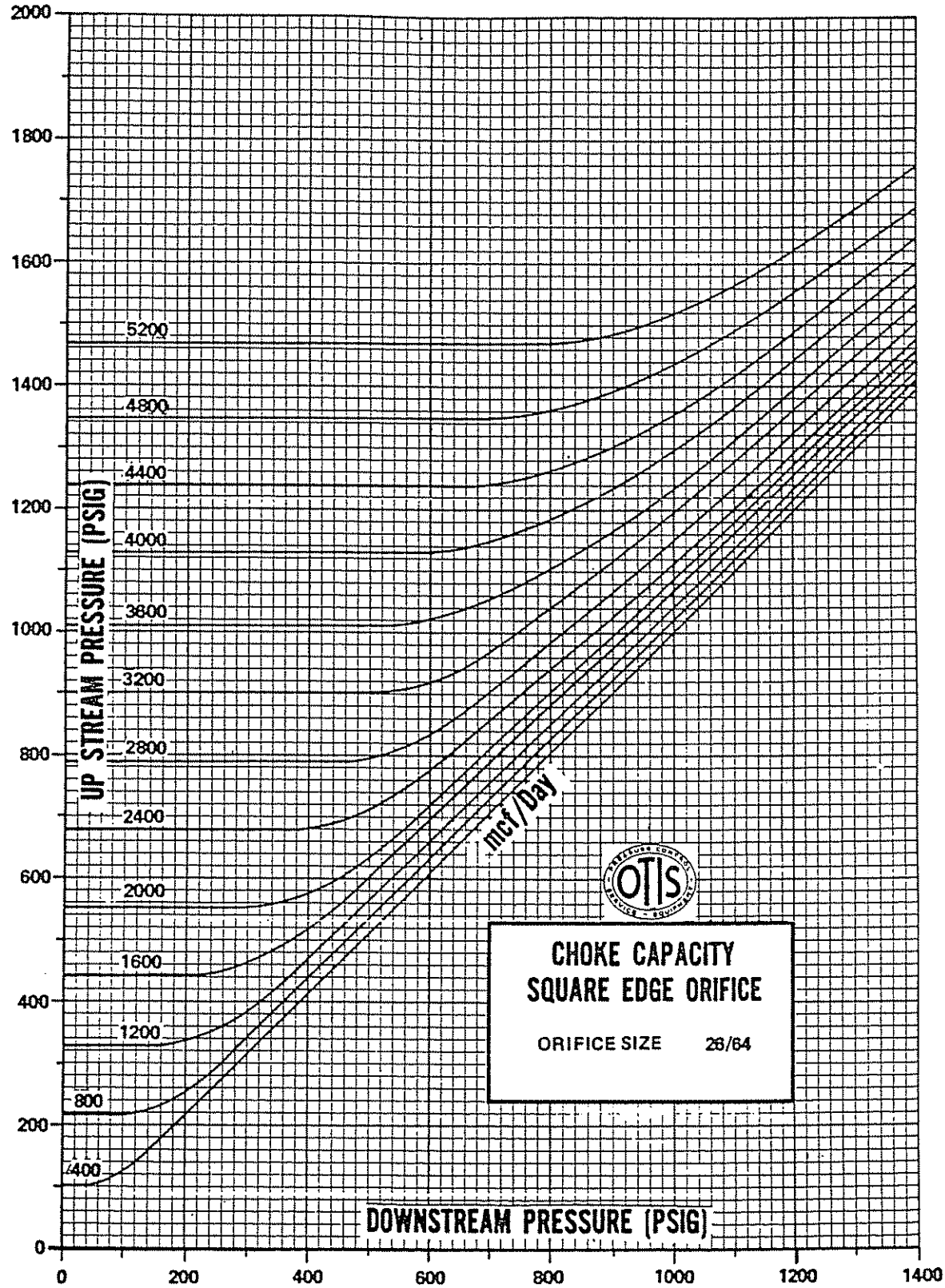


Fig. 3C-23 (Courtesy Otis Engineering Corp.)

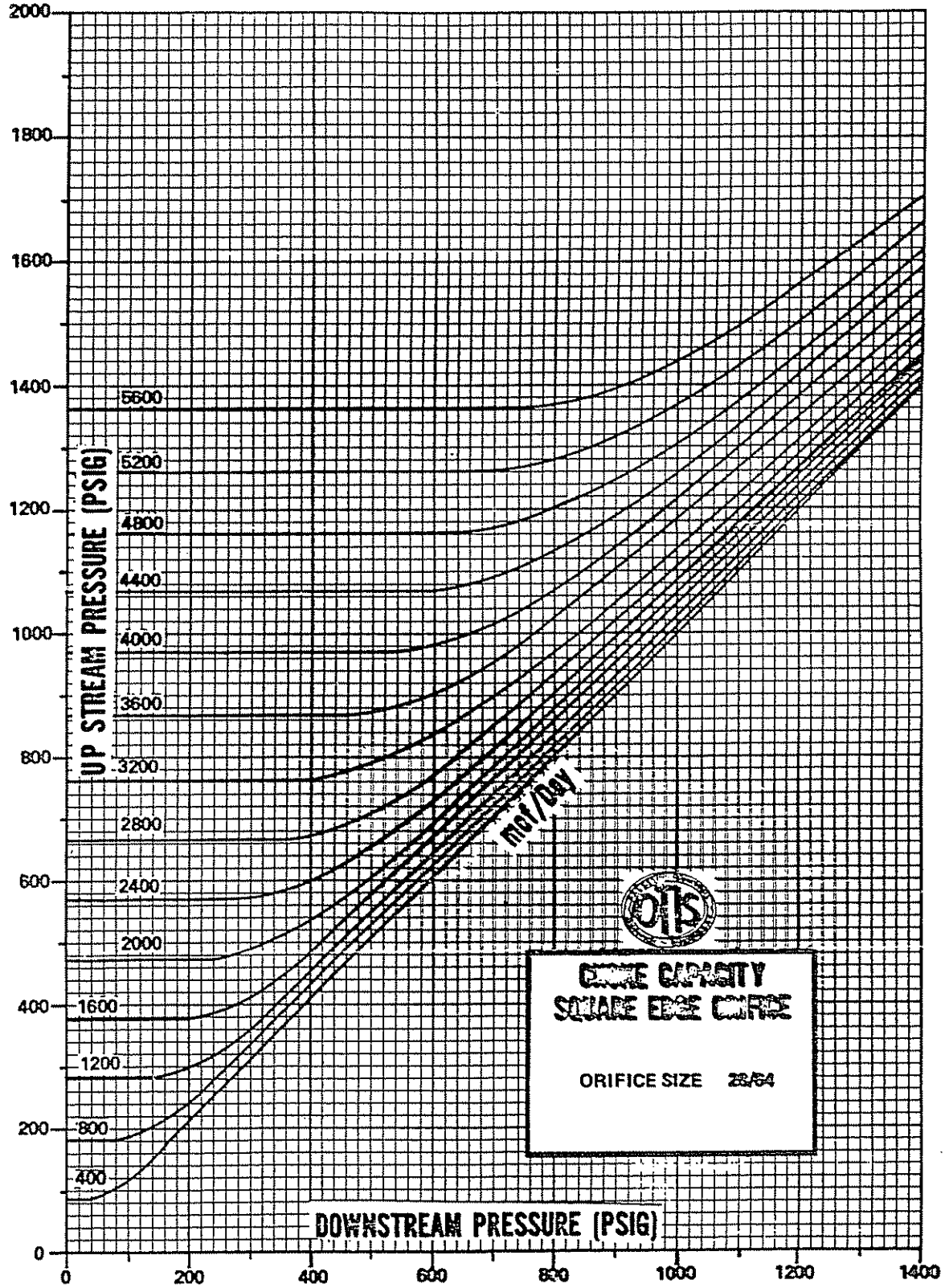


Fig. 3C-24 (Courtesy Otis Engineering Corp.)

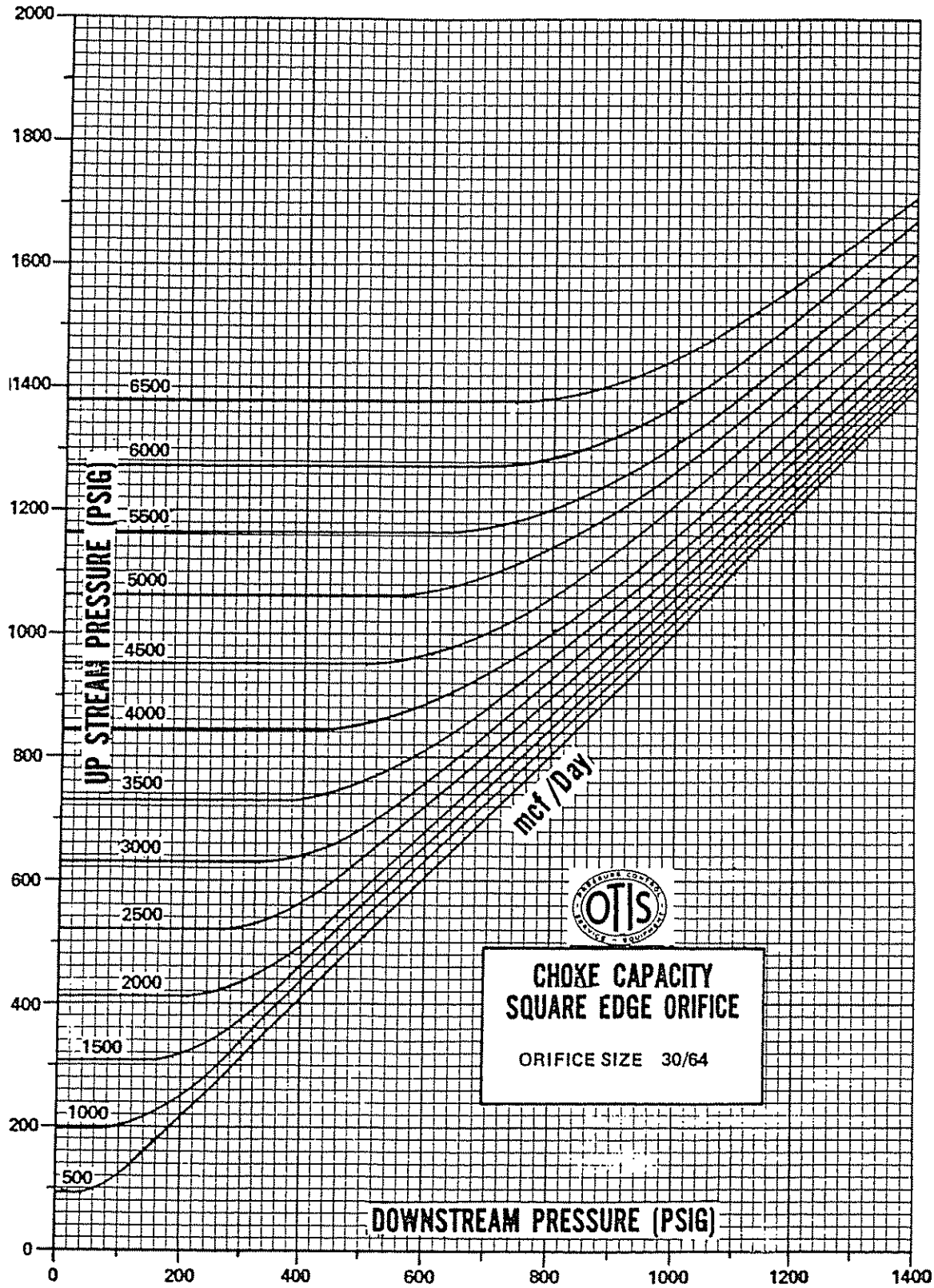


Fig. 3C-25 (Courtesy Otis Engineering Corp.)

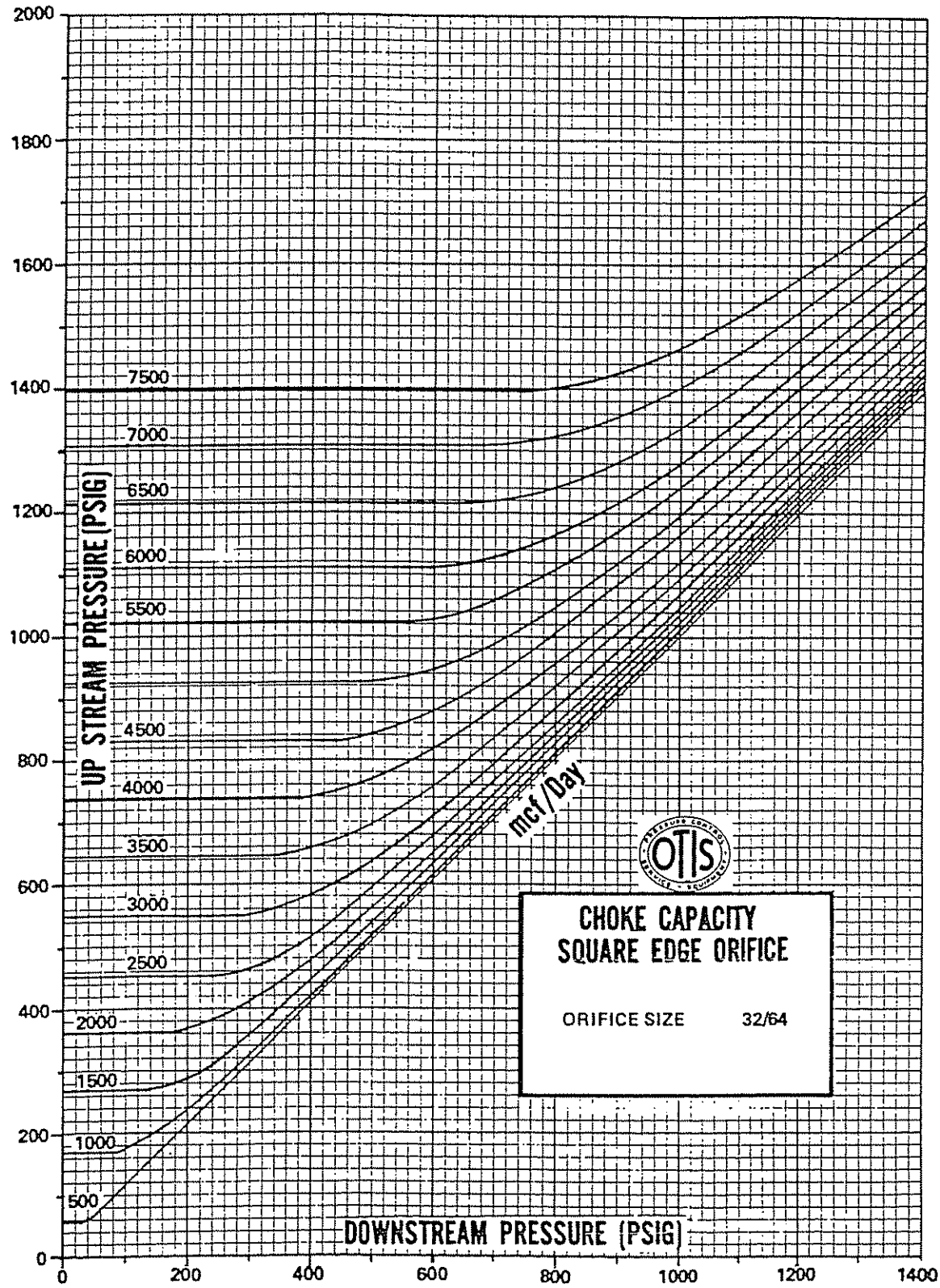


Fig. 3C-26 (Courtesy Otis Engineering Corp.)

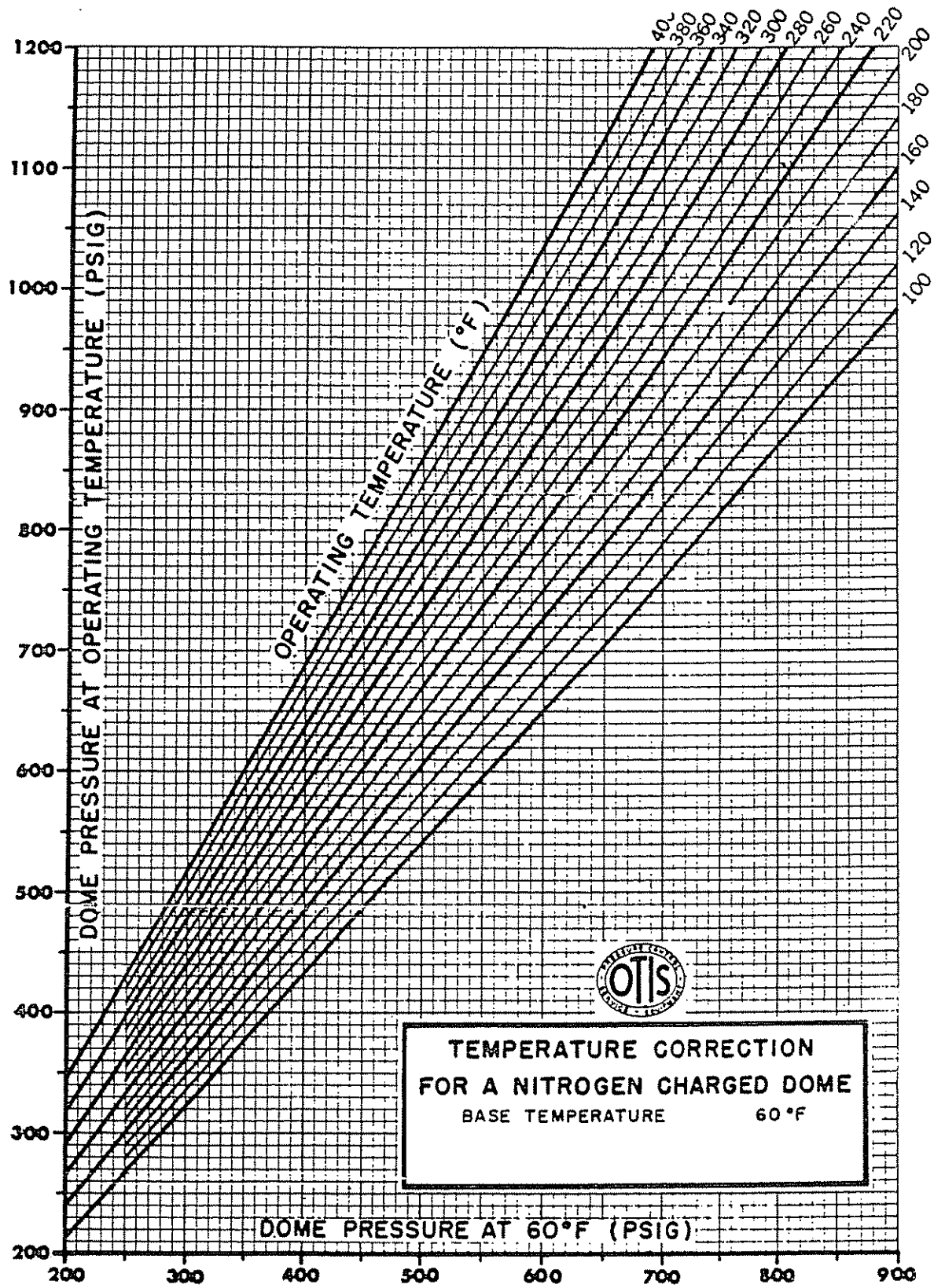


Fig. 3D-1 (Courtesy Otis Engineering Corp.)

TABLE 30.1
TEMPERATURE CORRECTION FACTORS FOR NITROGEN BASED ON 60 °F

°F	C_t	°F	C_t	°F	C_t	°F	C_t	°F	C_t	°F	C_t
61	.998	101	.919	141	.852	181	.794	221	.743	261	.698
62	.996	102	.917	142	.850	182	.792	222	.742	262	.697
63	.994	103	.915	143	.849	183	.791	223	.740	263	.696
64	.991	104	.914	144	.847	184	.790	224	.739	264	.695
65	.989	105	.912	145	.845	185	.788	225	.738	265	.694
66	.987	106	.910	146	.844	186	.787	226	.737	266	.693
67	.985	107	.908	147	.842	187	.786	227	.736	267	.692
68	.983	108	.906	148	.841	188	.784	228	.735	268	.691
69	.981	109	.905	149	.839	189	.783	229	.733	269	.690
70	.979	110	.903	150	.838	190	.782	230	.732	270	.689
71	.977	111	.901	151	.836	191	.780	231	.731	271	.688
72	.975	112	.899	152	.835	192	.779	232	.730	272	.687
73	.973	113	.898	153	.833	193	.778	233	.729	273	.686
74	.971	114	.896	154	.832	194	.776	234	.728	274	.685
75	.969	115	.894	155	.830	195	.775	235	.727	275	.684
76	.967	116	.893	156	.829	196	.774	236	.725	276	.683
77	.965	117	.891	157	.827	197	.772	237	.724	277	.682
78	.963	118	.889	158	.826	198	.771	238	.723	278	.681
79	.961	119	.887	159	.825	199	.770	239	.722	279	.680
80	.959	120	.886	160	.823	200	.769	240	.721	280	.679
81	.957	121	.884	161	.822	201	.767	241	.720	281	.678
82	.955	122	.882	162	.820	202	.766	242	.719	282	.677
83	.953	123	.881	163	.819	203	.765	243	.718	283	.676
84	.951	124	.879	164	.817	204	.764	244	.717	284	.675
85	.949	125	.877	165	.816	205	.762	245	.715	285	.674
86	.947	126	.876	166	.814	206	.761	246	.714	286	.673
87	.945	127	.874	167	.813	207	.760	247	.713	287	.672
88	.943	128	.872	168	.812	208	.759	248	.712	288	.671
89	.941	129	.871	169	.810	209	.757	249	.711	289	.670
90	.939	130	.869	170	.809	210	.756	250	.710	290	.699
91	.938	131	.686	171	.807	211	.755	251	.709	291	.668
92	.936	132	.866	172	.806	212	.754	252	.708	292	.667
93	.934	133	.864	173	.805	213	.752	253	.707	293	.666
94	.932	134	.863	174	.803	214	.751	254	.706	294	.665
95	.930	135	.861	175	.802	215	.750	255	.705	295	.664
96	.928	136	.860	176	.800	216	.749	256	.704	296	.663
97	.926	137	.858	177	.799	217	.748	257	.702	297	.662
98	.924	138	.856	178	.798	218	.746	258	.701	298	.662
99	.923	139	.855	179	.796	219	.745	259	.700	299	.661
100	.921	140	.853	180	.795	220	.744	260	.699	300	.660

$$C_t = \frac{\text{gas lift valve dome pressure at 60 °F}}{\text{gas lift valve dome pressure at well temperature}}$$

TABLE 30.2
COMPRESSIBILITY FACTORS FOR NITROGEN

PRESSURE		TEMPERATURES IN °F						
PSIG	PSIA	50	60	100	150	200	250	300
100	114.7	0.9975	0.9980	0.9999	1.0010	1.0018	1.0023	1.0028
200	214.7	0.9958	0.9966	0.9998	1.0020	1.0032	1.0045	1.0056
300	314.7	0.9943	0.9954	0.9999	1.0031	1.0052	1.0069	1.0083
400	414.7	0.9930	0.9946	1.0000	1.0042	1.0072	1.0092	1.0110
500	514.7	0.9921	0.9937	1.0003	1.0056	1.0093	1.0117	1.0135
600	614.7	0.9910	0.9930	1.0008	1.0069	1.0112	1.0141	1.0166
700	714.7	0.9903	0.9925	1.0013	1.0084	1.0133	1.0166	1.0194
800	814.7	0.9900	0.9924	1.0021	1.0101	1.0155	1.0192	1.0223
900	914.7	0.9900	0.9926	1.0031	1.0119	1.0181	1.0220	1.0252
1000	1014.7	0.9901	0.9930	1.0045	1.0141	1.0206	1.0251	1.0281
1100	1114.7	0.9902	0.9933	1.0056	1.0158	1.0229	1.0276	1.0311
1200	1214.7	0.9909	0.9941	1.0071	1.0179	1.0255	1.0305	1.0341
1300	1314.7	0.9918	0.9952	1.0088	1.0203	1.0283	1.0335	1.0372
1400	1414.7	0.9929	0.9965	1.0107	1.0226	1.0309	1.0365	1.0402
1500	1514.7	0.9942	0.9979	1.0128	1.0253	1.0337	1.0396	1.0433

The compressibility factors were determined by direct interpolation for even hundreds pressure gauge from even hundreds pressure absolute from Monograph on API Research Project 37 entitled, "Thermodynamic Properties of the Lighter Paraffin Hydrocarbons and Nitrogen," by Sage and Lacey.

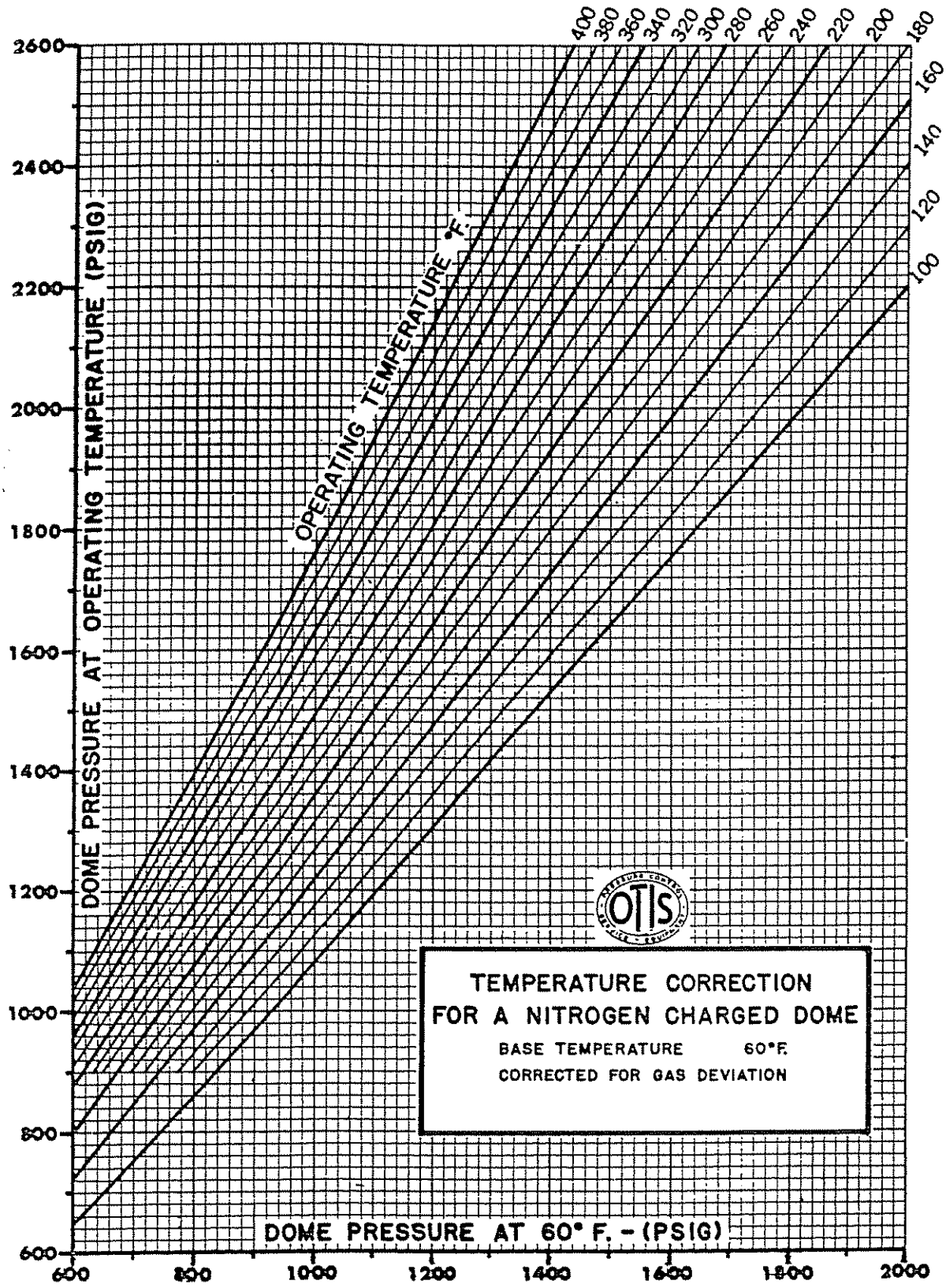


Fig. 3D-2 (Courtesy Otis Engineering Corp.)

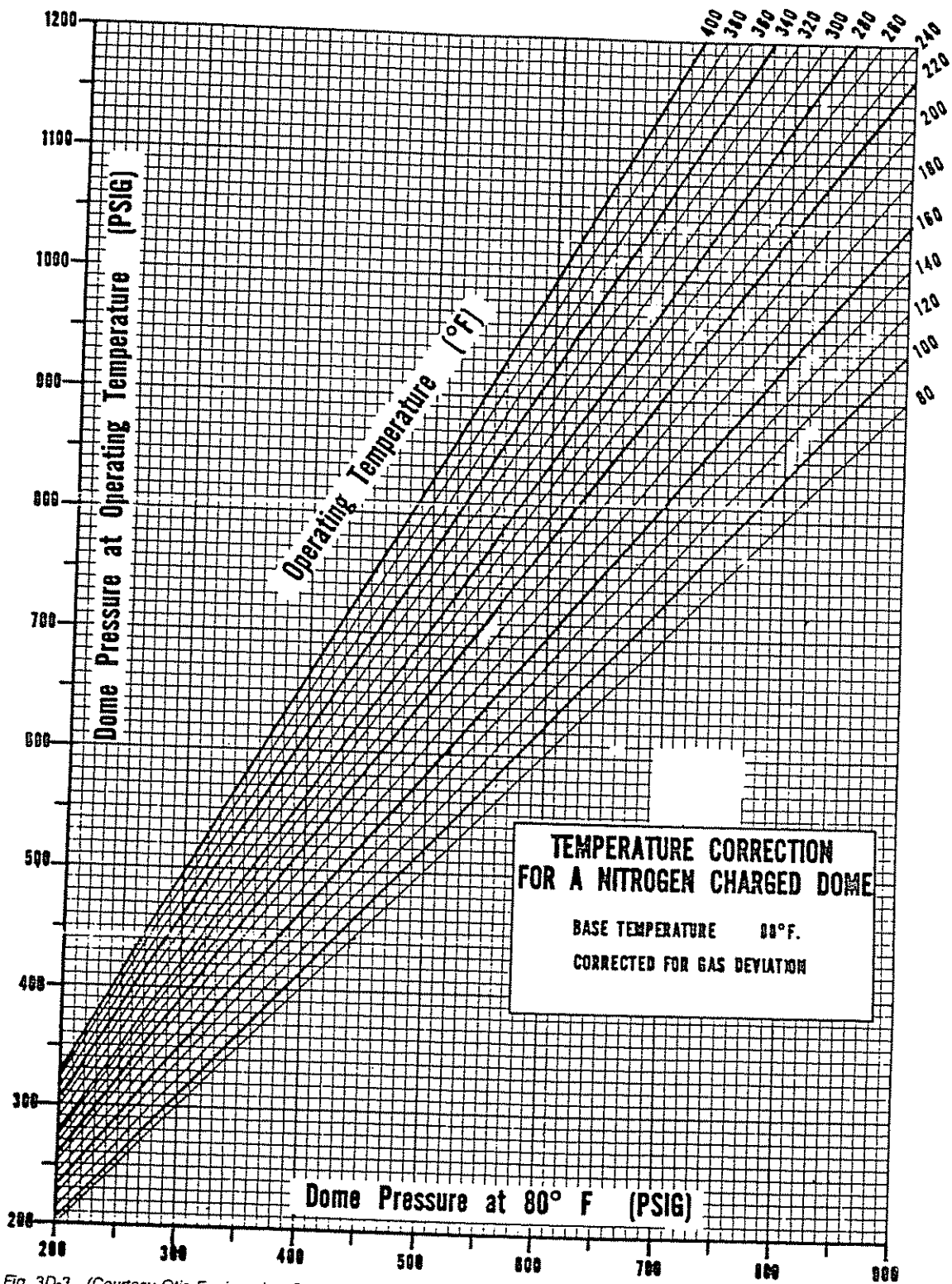


Fig. 3D-3 (Courtesy Otis Engineering Corp.)

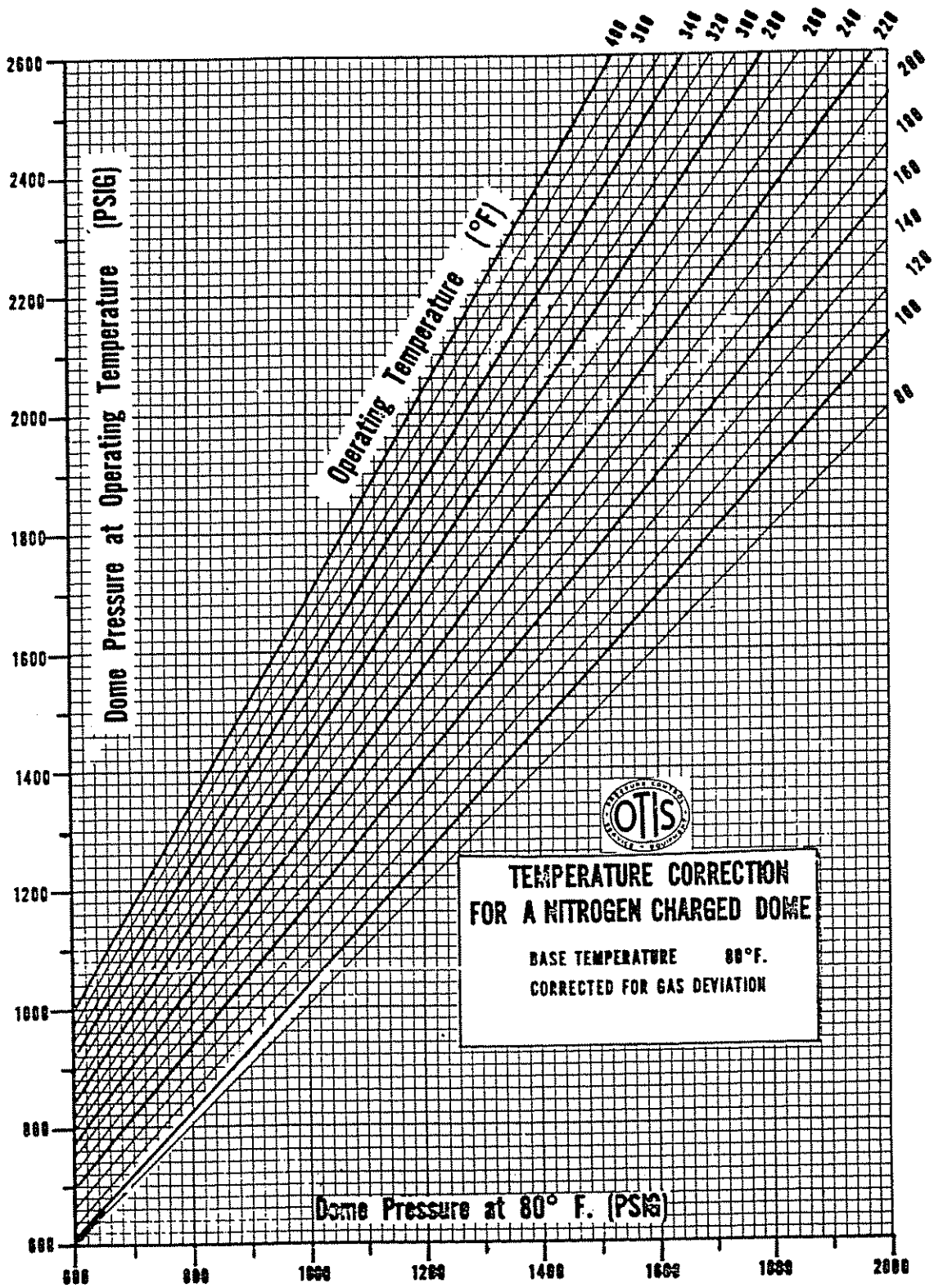


Fig. 3D-4 (Courtesy Otis Engineering Corp.)

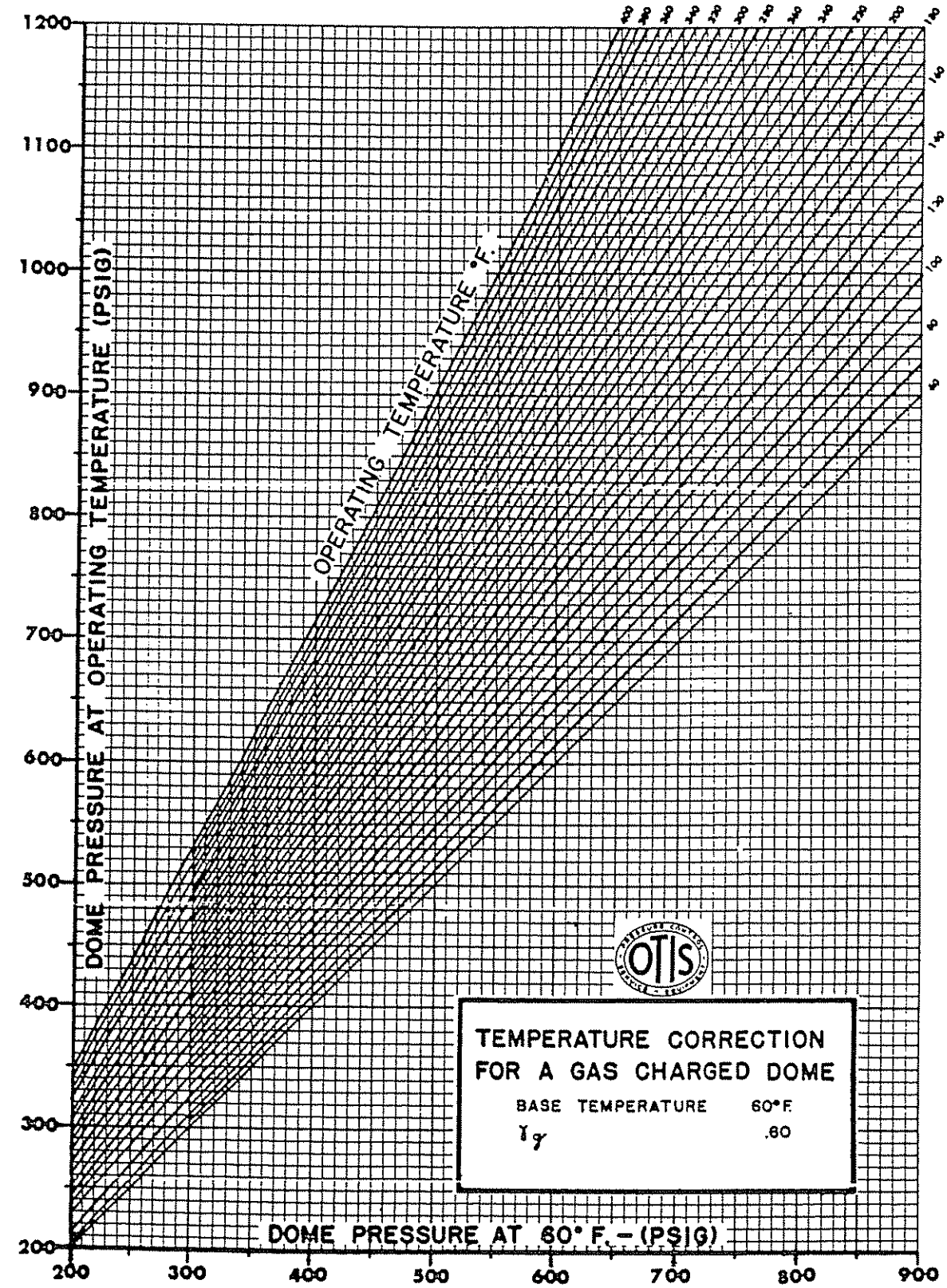


Fig. 3D-5 (Courtesy Otis Engineering Corp.)

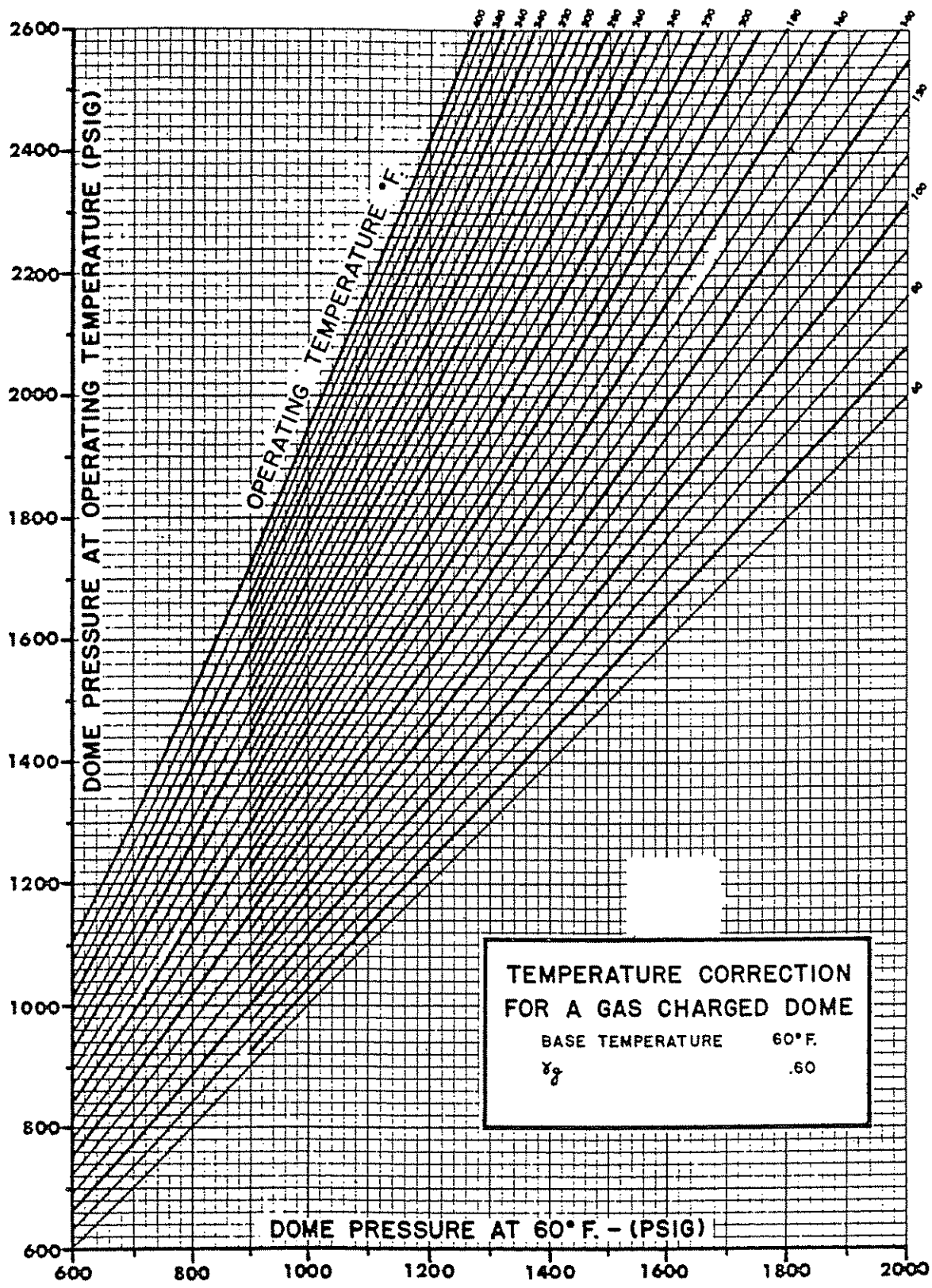


Fig. 3D-6 (Courtesy Otis Engineering Corp.)

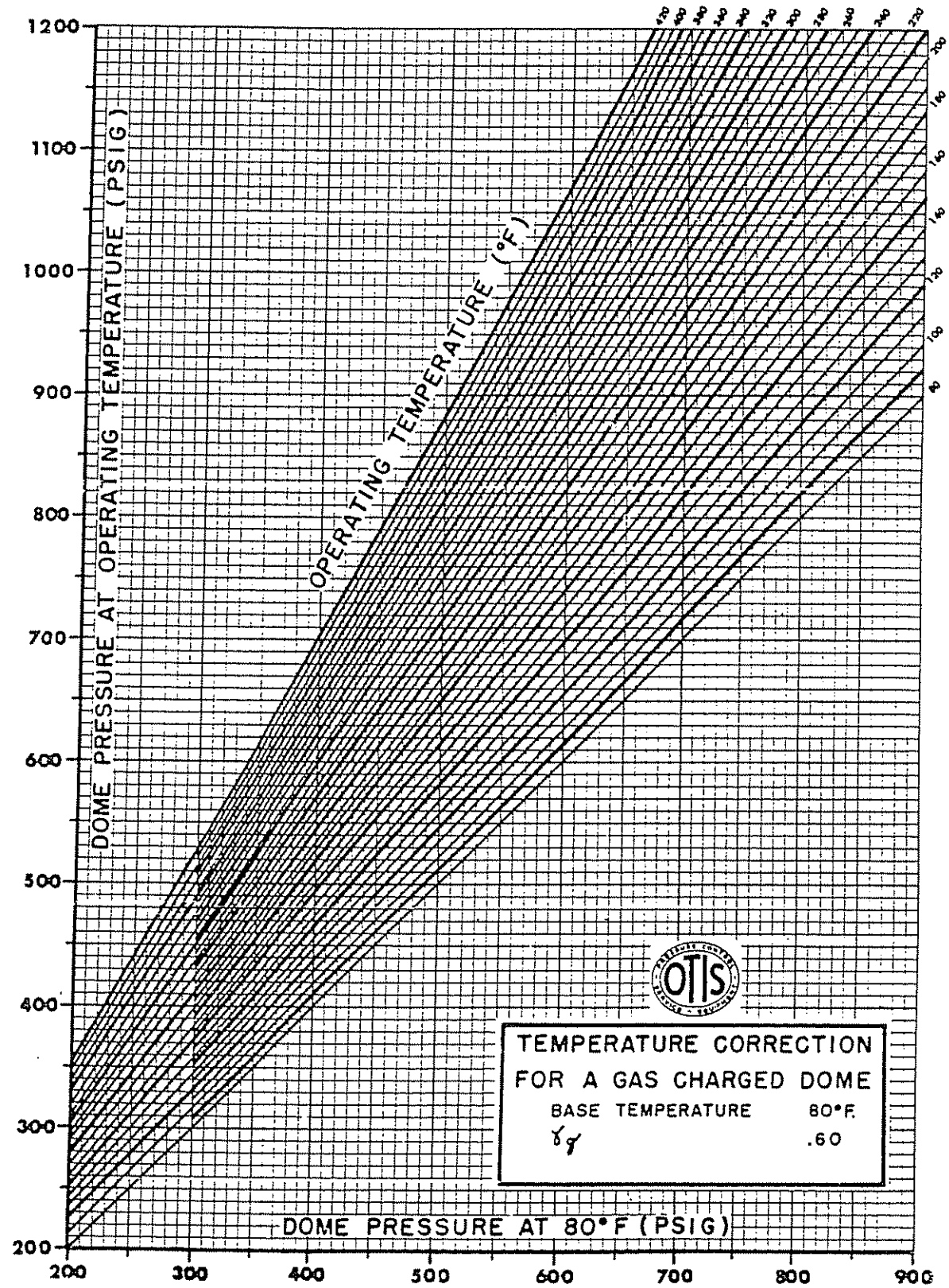


Fig. 3D-7 (Courtesy Otis Engineering Corp.)

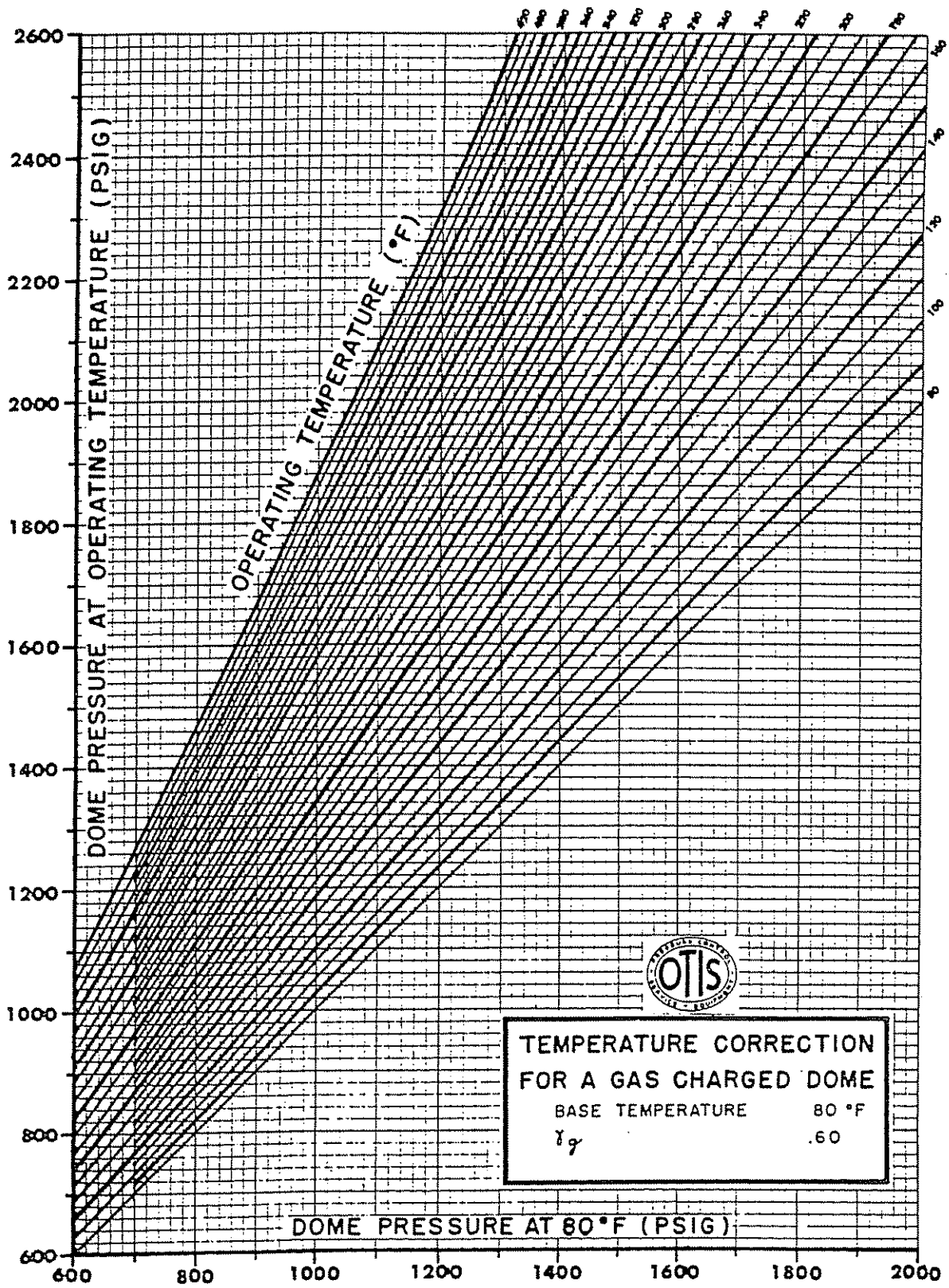


Fig. 3D-8 (Courtesy Otis Engineering Corp.)

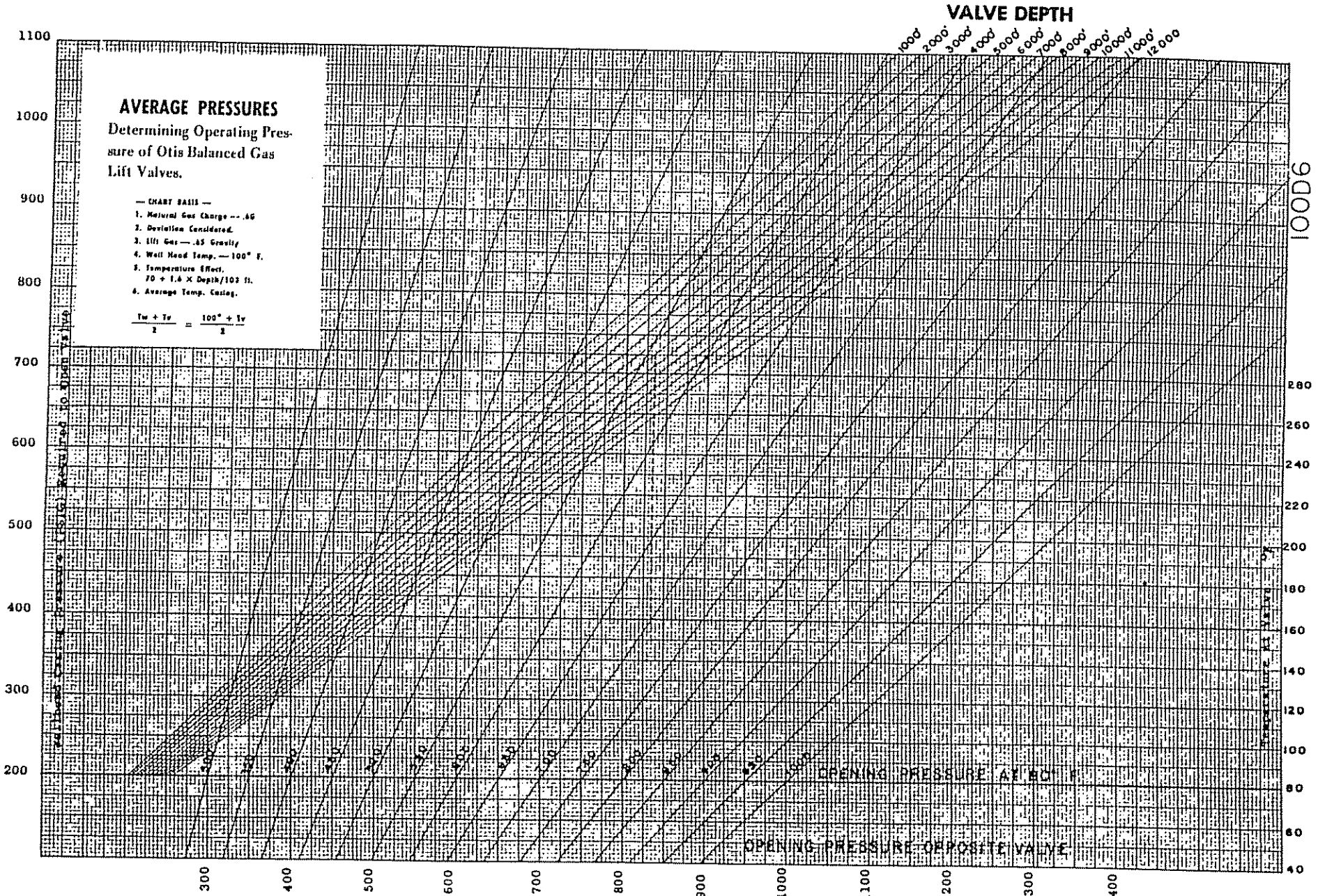


Fig. 3D-9 To correct for gas and temp. conditions other than Fig. basis (Courtesy Otis Engineering Corp.)

$$(\Delta P \text{ chart}) \left(\frac{\text{Chart avg. Temp.° Rankine}}{\text{Actual avg. Temp.° Rankine}}\right) \left(\frac{\text{gas grav. actual}}{\text{gas grav. chart}}\right)$$

(Courtesy Otis Engineering Corp.)

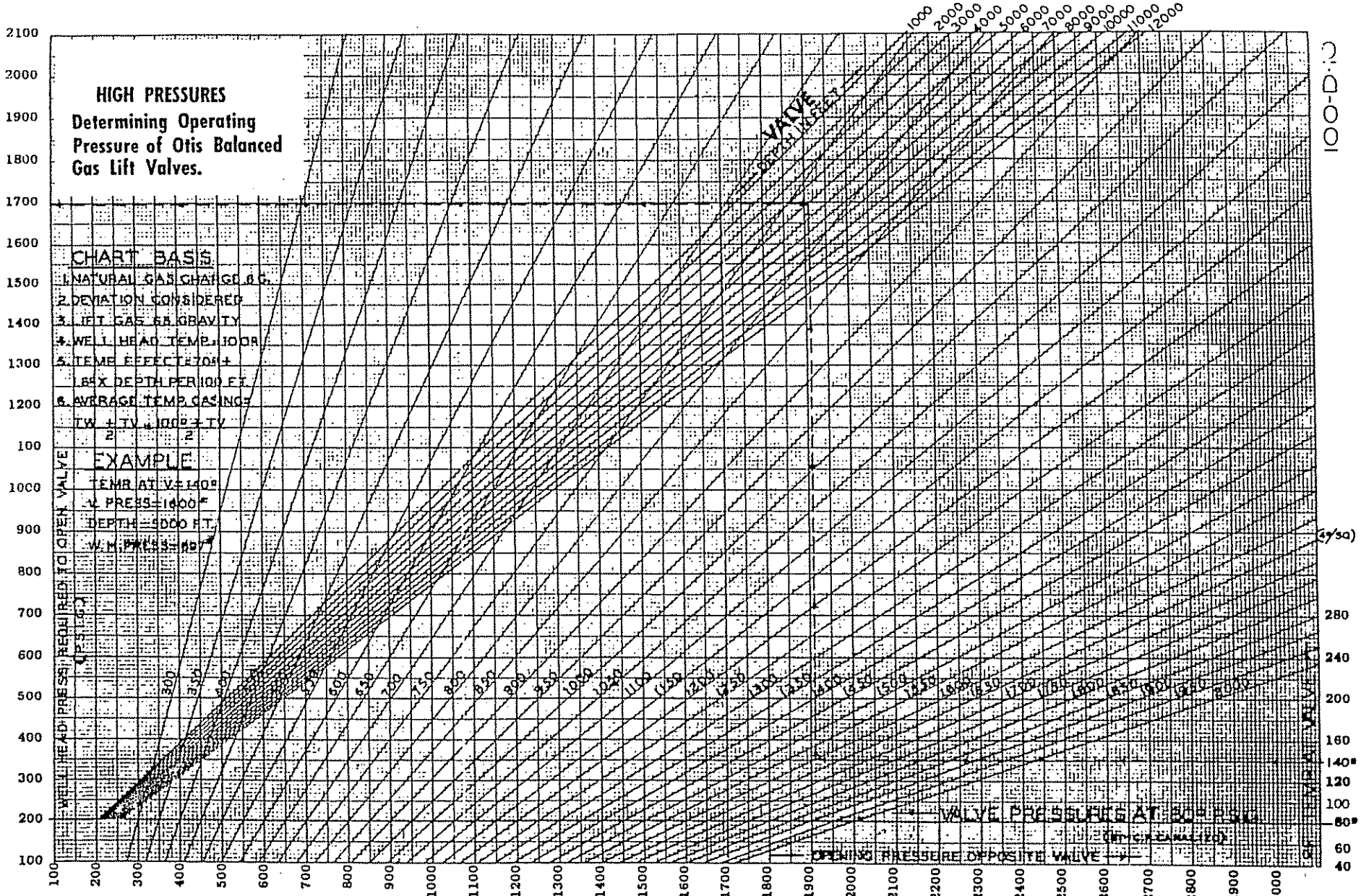


Fig. 3D-10 To correct for gas and temp. conditions other than Fig. basis (Courtesy Otis Engineering Corp.)

$$(\Delta P \text{ chart}) \left(\frac{\text{Chart avg. Temp.° Rankine}}{\text{Actual avg. Temp.° Rankine}}\right) \left(\frac{\text{gas grav. actual}}{\text{gas grav. chart}}\right)$$

(Courtesy Otis Engineering Corp.)

Fig. 3D.11

VARIATION OF NITROGEN BELLOWS CHARGE PRESSURE WITH TEMPERATURE

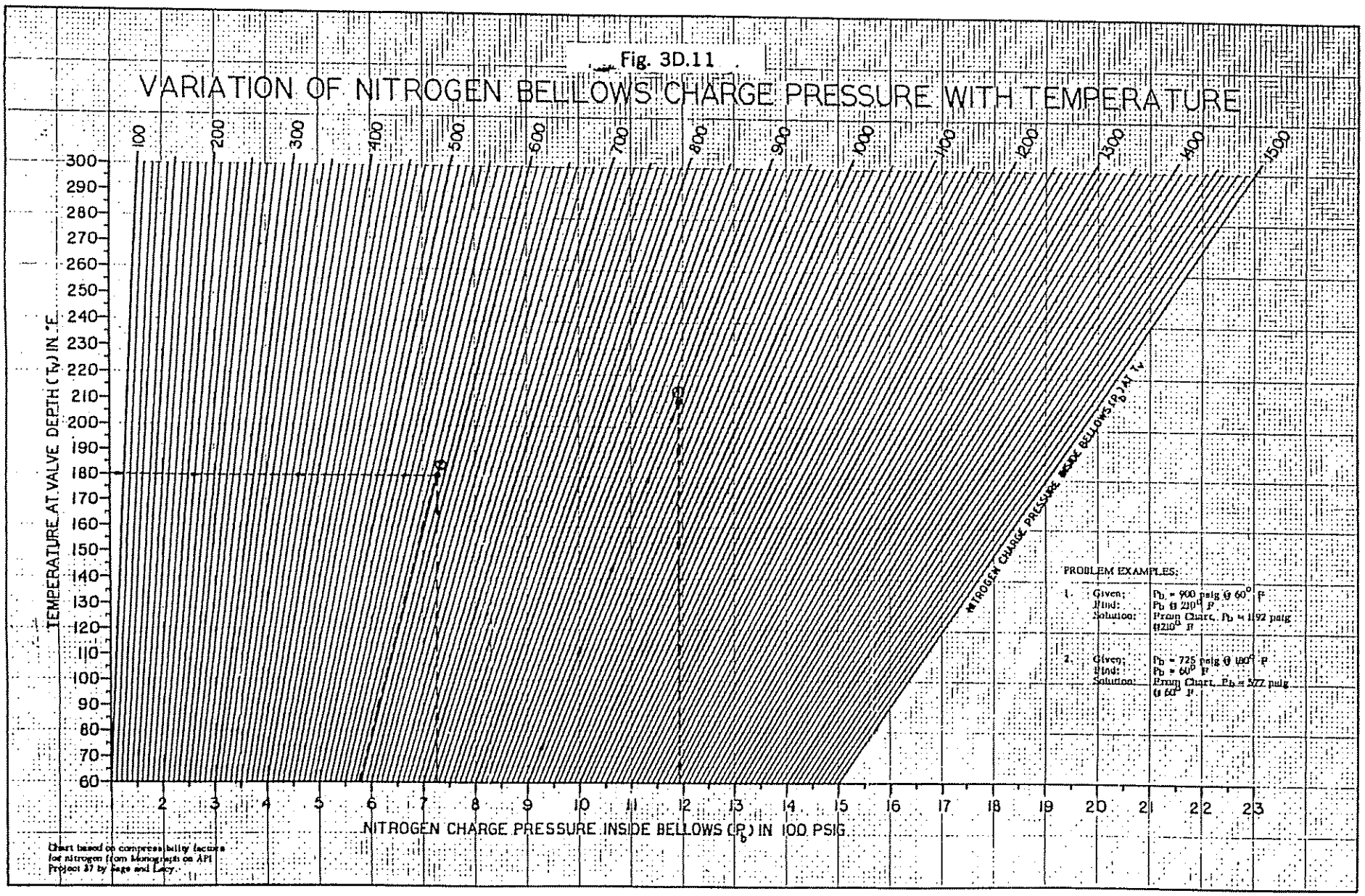


Chart based on compressibility factors for nitrogen from Monograph on API Project 37 by Sage and Lacy.

EXPLANATION OF FIGURES ENTITLED TUBING PRESSURE EFFECT — UNBALANCED VALVE 1½" and 1" Figures 3E.1 through 3E.15

These charts are prepared to determine the tubing pressure effect on both 1½" and 1" unbalanced gas lift valves. Once the dome pressure has been determined, the opening pressure at any tubing pressure can be determined. The test rack opening pressure can be determined by noting the opening pressure at zero tubing pressure.

The following problems can be solved by use of these charts:

(1) If the dome pressure or closing pressure is known the opening pressure at any tubing pressure can be selected.
(2) If the dome pressure at any temperature is known, the test rack opening pressure can be obtained at zero tubing pressure.
(3) If the opening pressure is known and the tubing load to be lifted is known, the dome pressure or closing pressure can be obtained.

The following procedure should be followed:

(1) For a known dome pressure (closing pressure) determine the opening pressure under a tubing load:
   (a) Note the dome pressure on the diagonal line labeled closing pressure = dome pressure.
   (b) Note the tubing pressure on the left vertical margin.
   (c) Proceed upward to the right of the dome pressure on a line parallel to the drawn lines until intersecting the line coming horizontally to the right from the tubing pressure.
   (d) From this intersection proceed on a true vertical to the top and read the opening pressure.
(2) For a known dome pressure select the test rack opening pressure.
   (a) Note the dome pressure on the diagonal line labeled closing pressure = dome pressure.
   (b) Proceed upward to the right on a line parallel to the drawn lines all the way to the top.
   (c) Read the test rack opening pressure.
(3) For a known opening pressure and a known tubing load to be lifted, determine the dome pressure or closing pressure of the valve.
   (a) From the opening at top dome proceed vertically downward until intersecting the tubing pressure coming horizontally from the left.
   (b) From this point of intersection proceed downward to the left following parallel to the drawn lines until reaching the closing pressure = dome pressure line. This is the closing pressure of the valve as read vertically upward from this point.

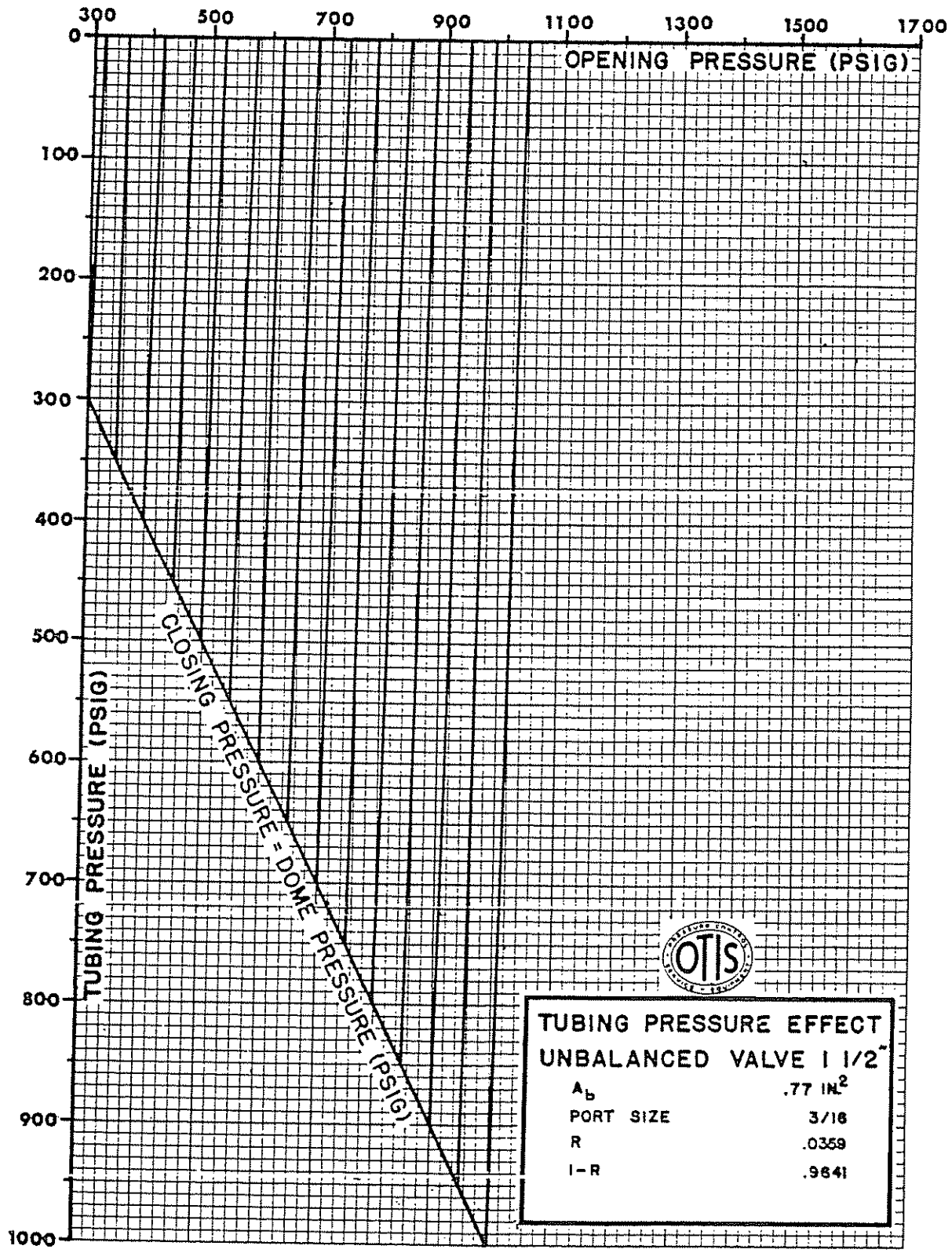


Fig. 3E-1 (Courtesy Otis Engineering Corp.)

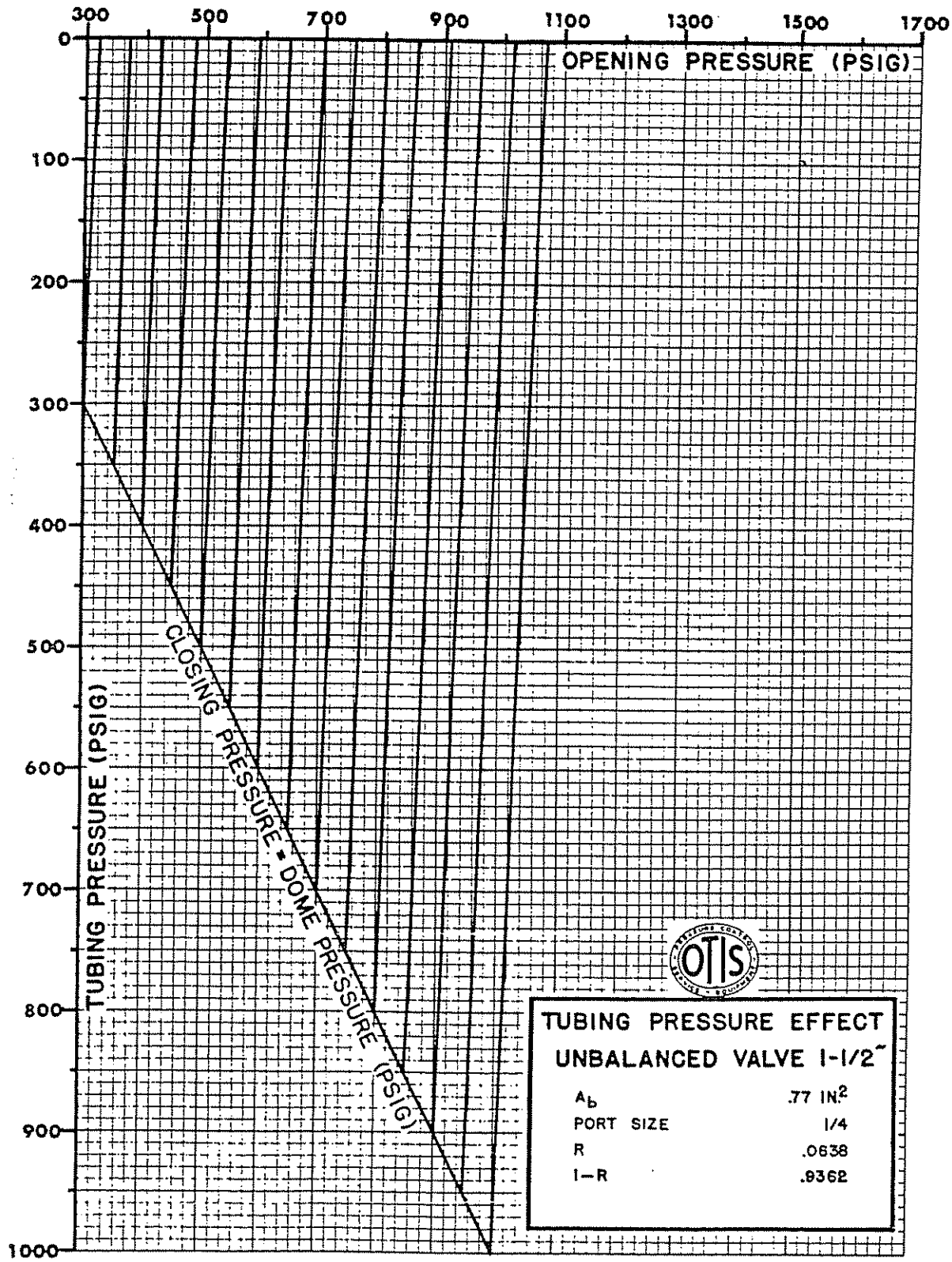


Fig. 3E-2 (Courtesy Otis Engineering Corp.)

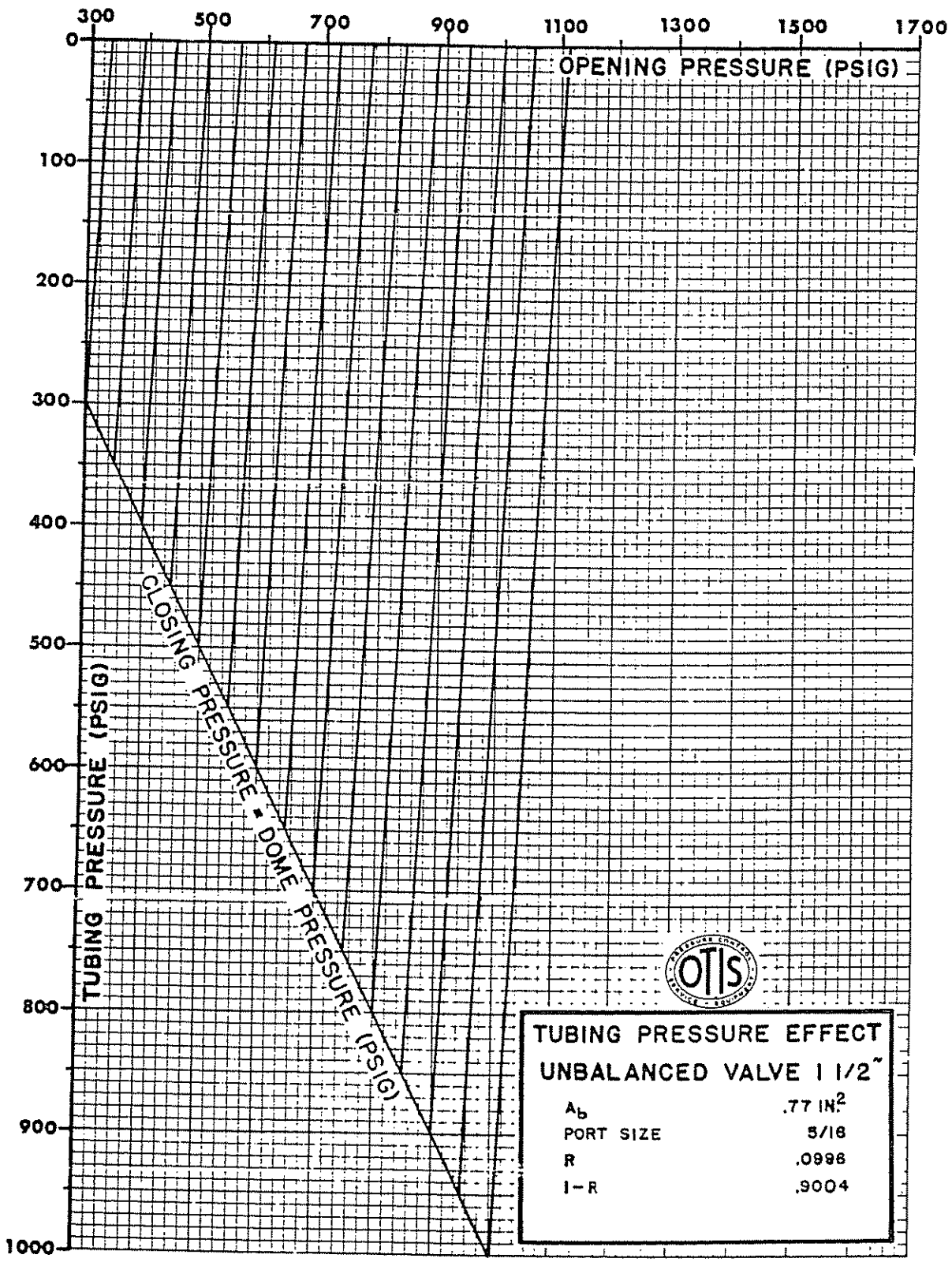


Fig. 3E-3 (Courtesy Otis Engineering Corp.)

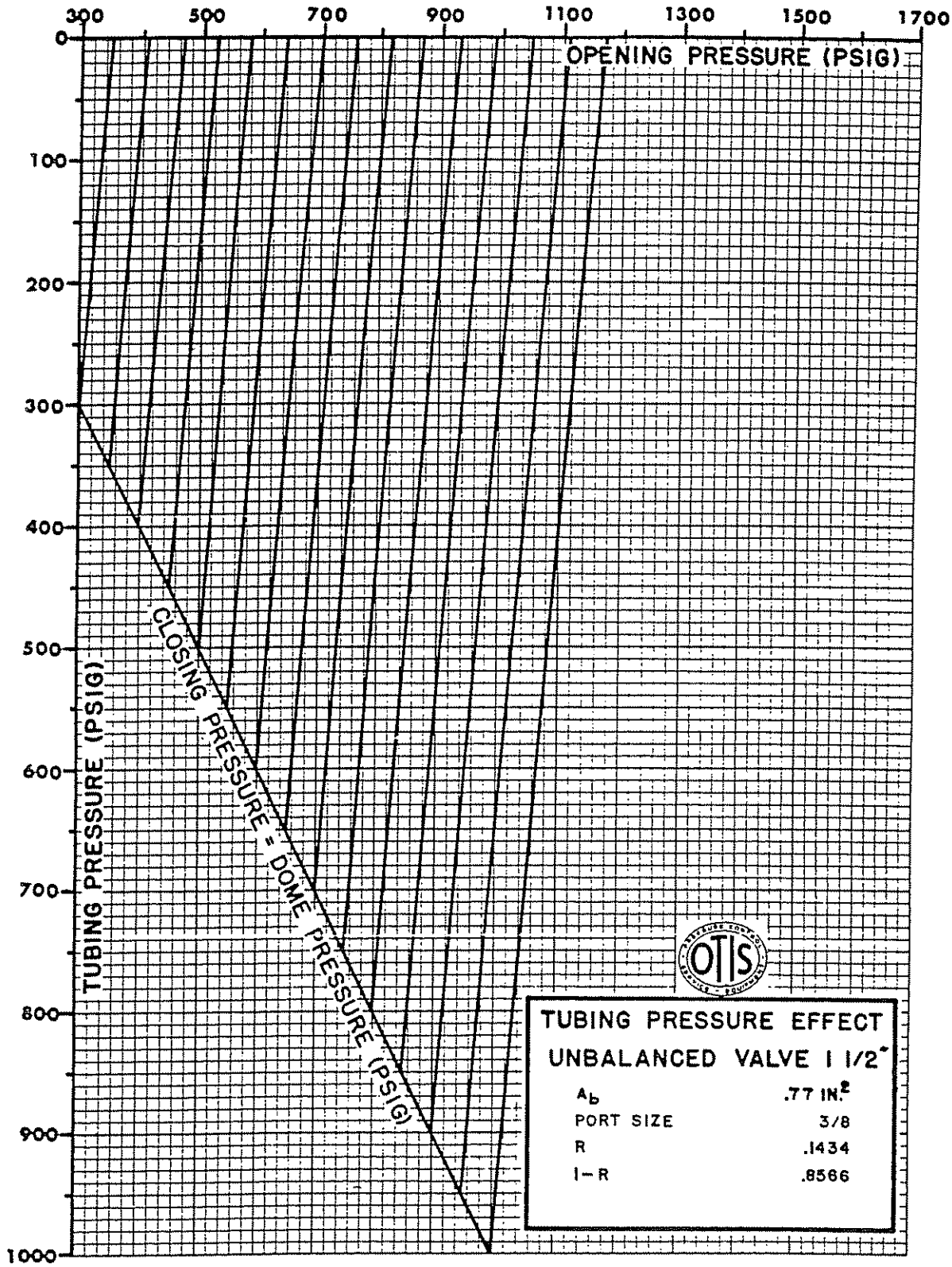


Fig. 3E-4 (Courtesy Otis Engineering Corp.)

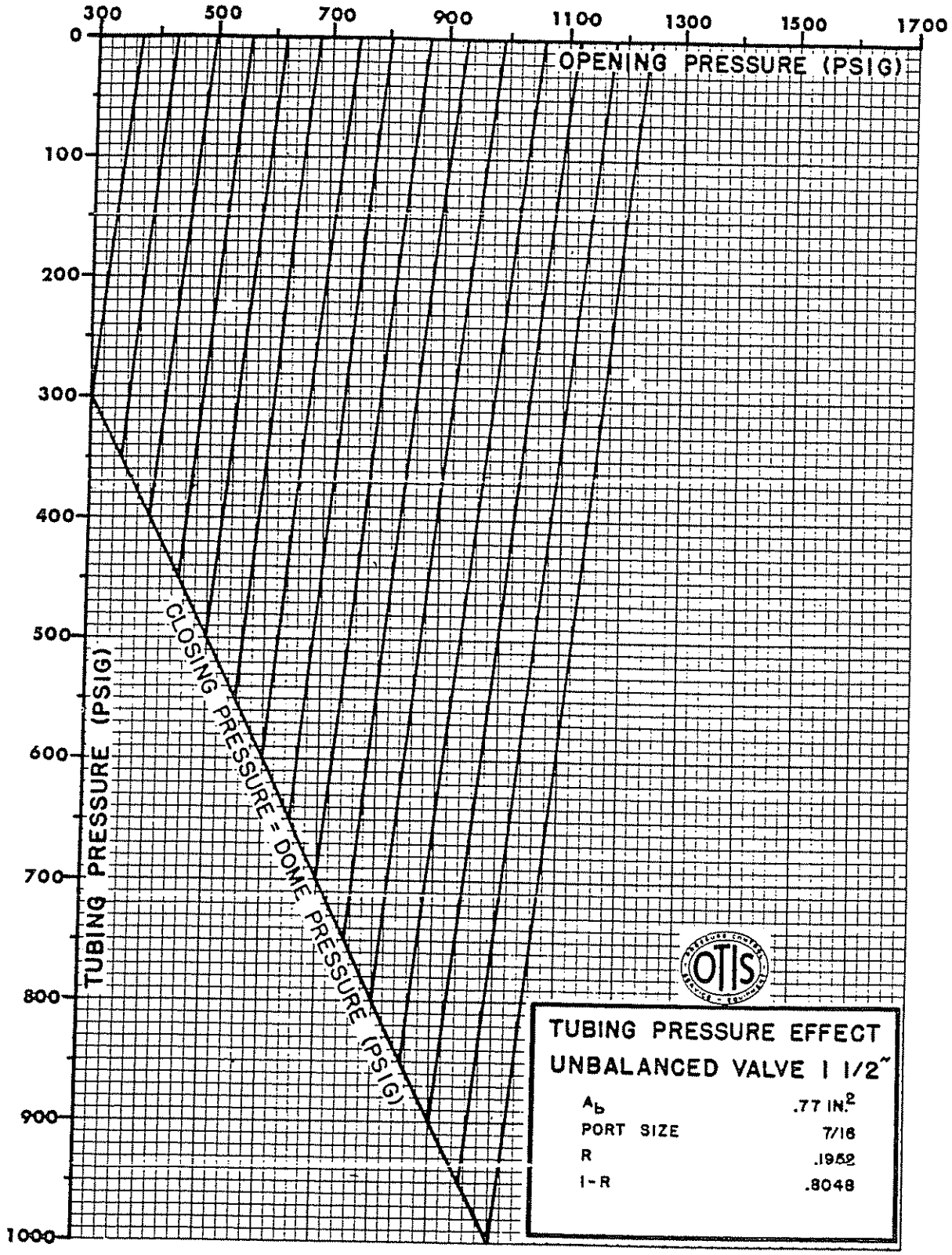


Fig. 3E-5 (Courtesy Otis Engineering Corp.)

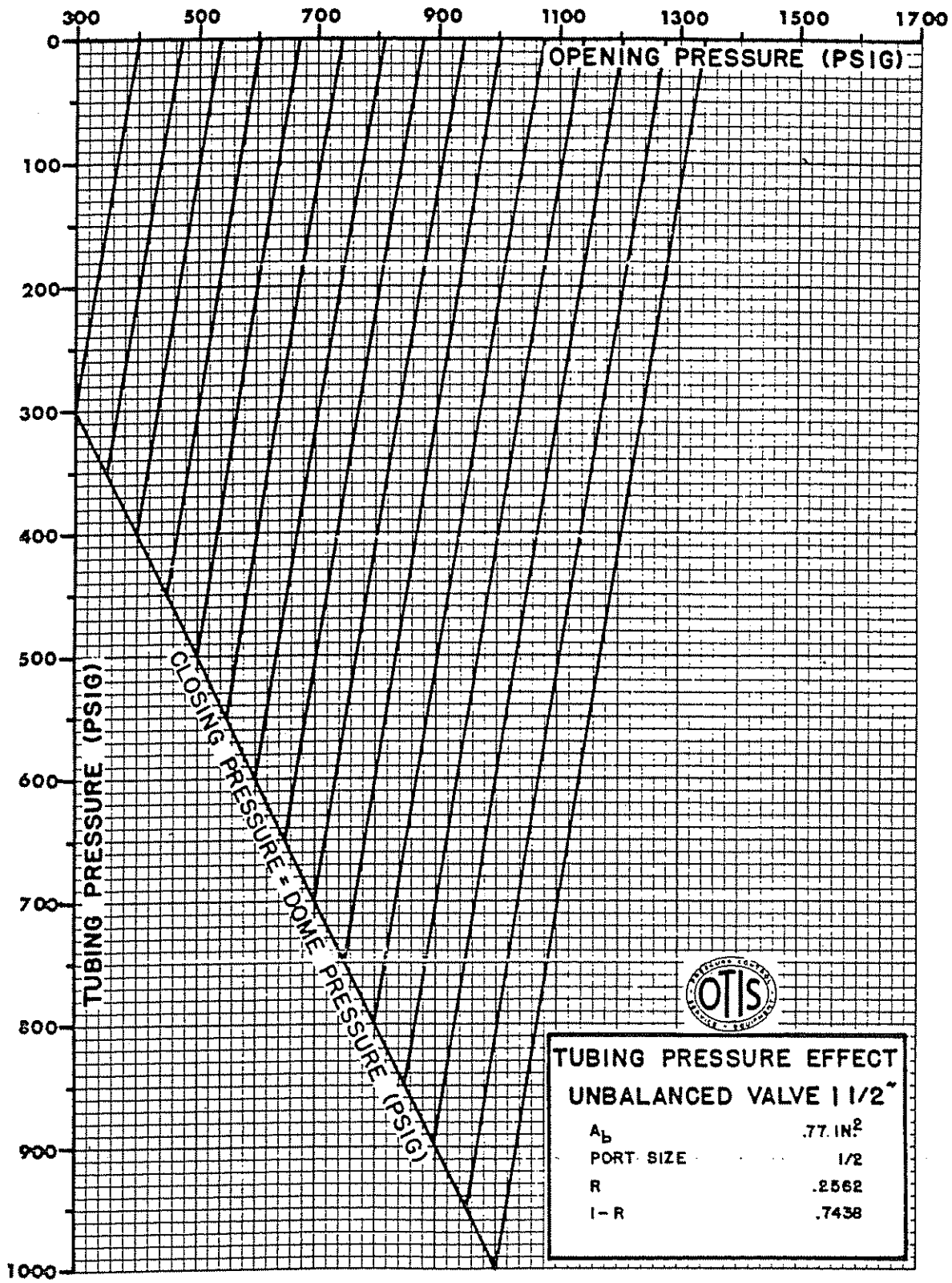


Fig. 3E-6 (Courtesy Otis Engineering Corp.)

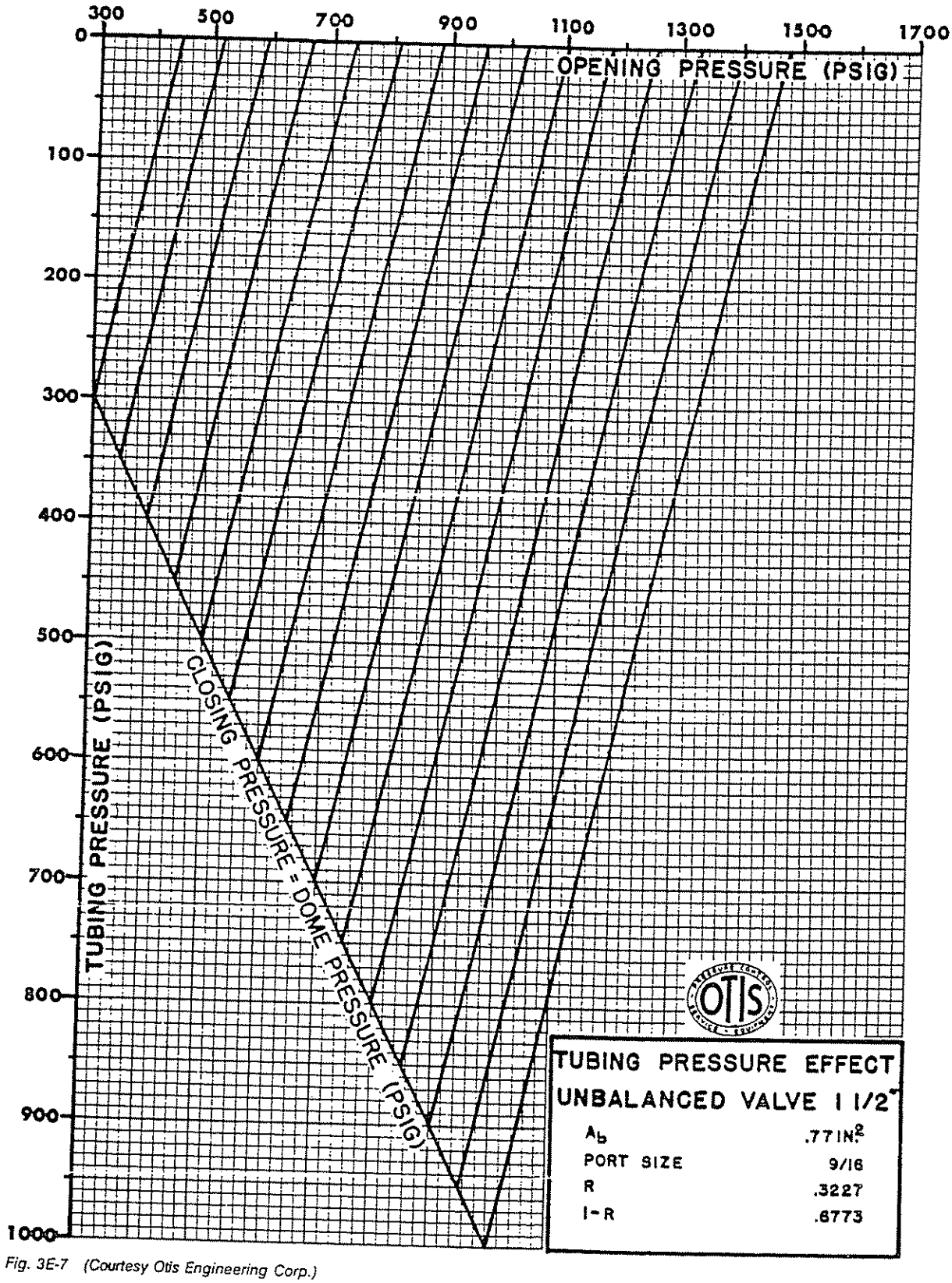


Fig. 3E-7 (Courtesy Otis Engineering Corp.)

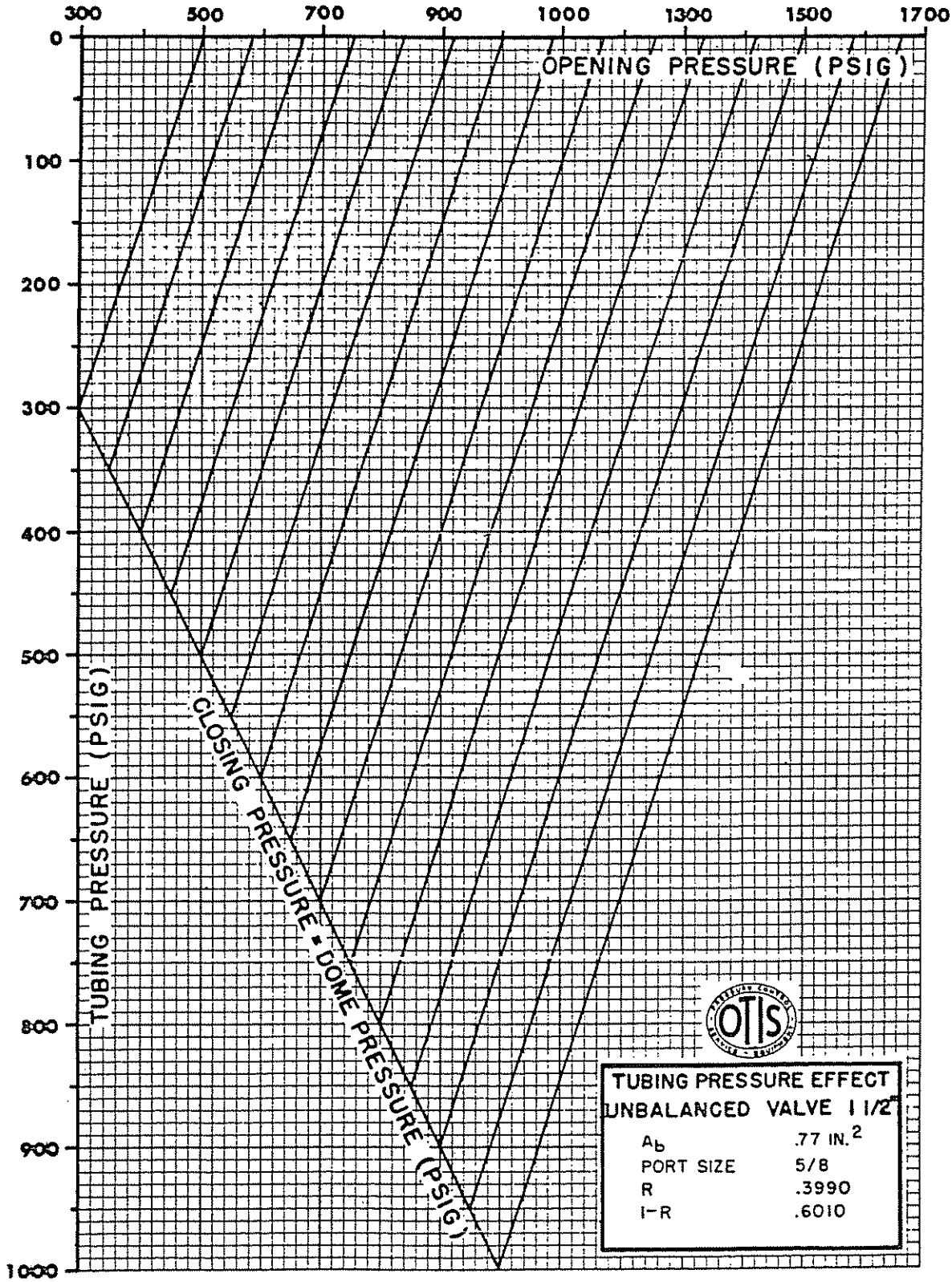


Fig. 3E-8 (Courtesy Otis Engineering Corp.)

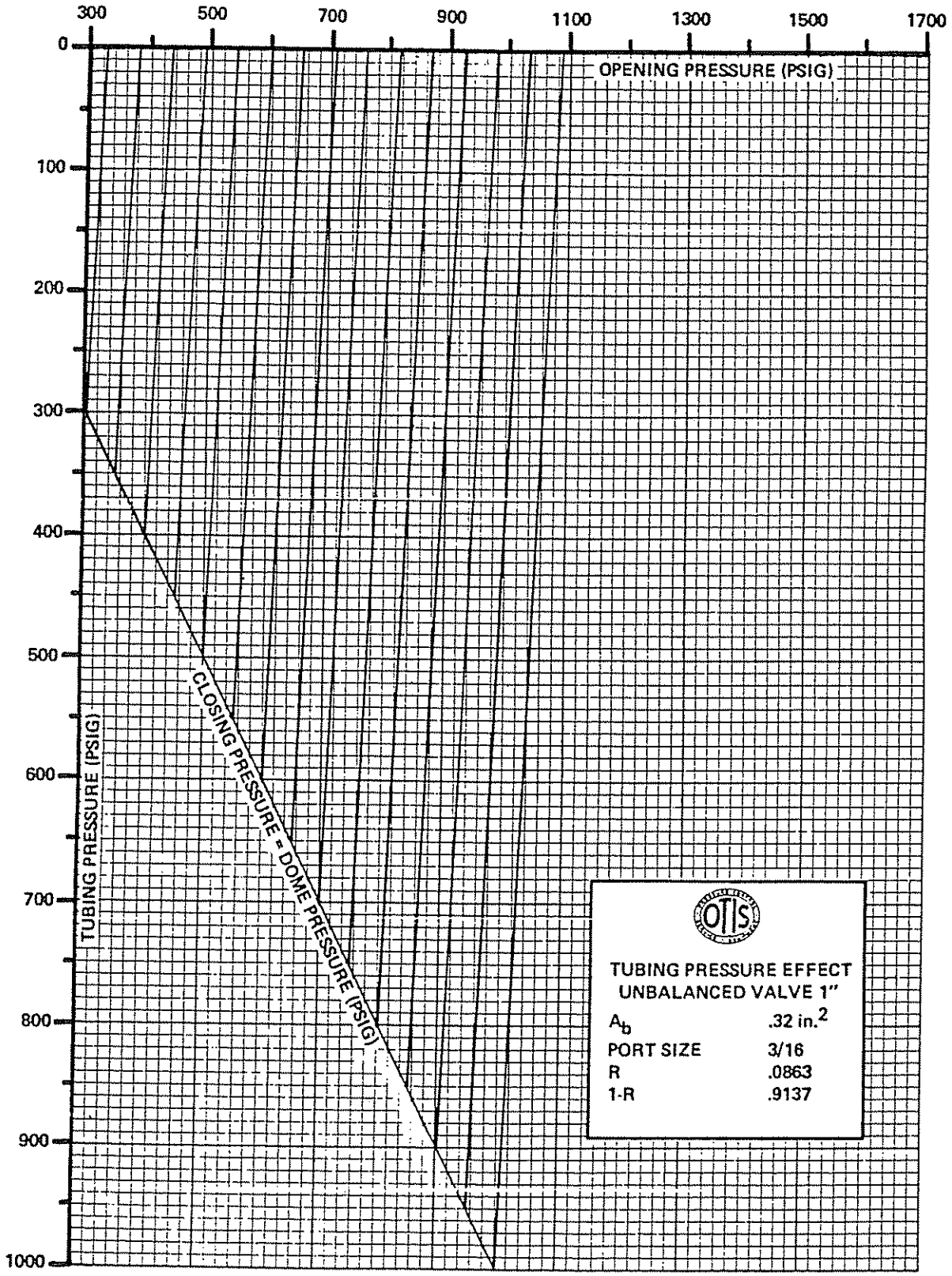


Fig. 3E-9 (Courtesy Otis Engineering Corp.)

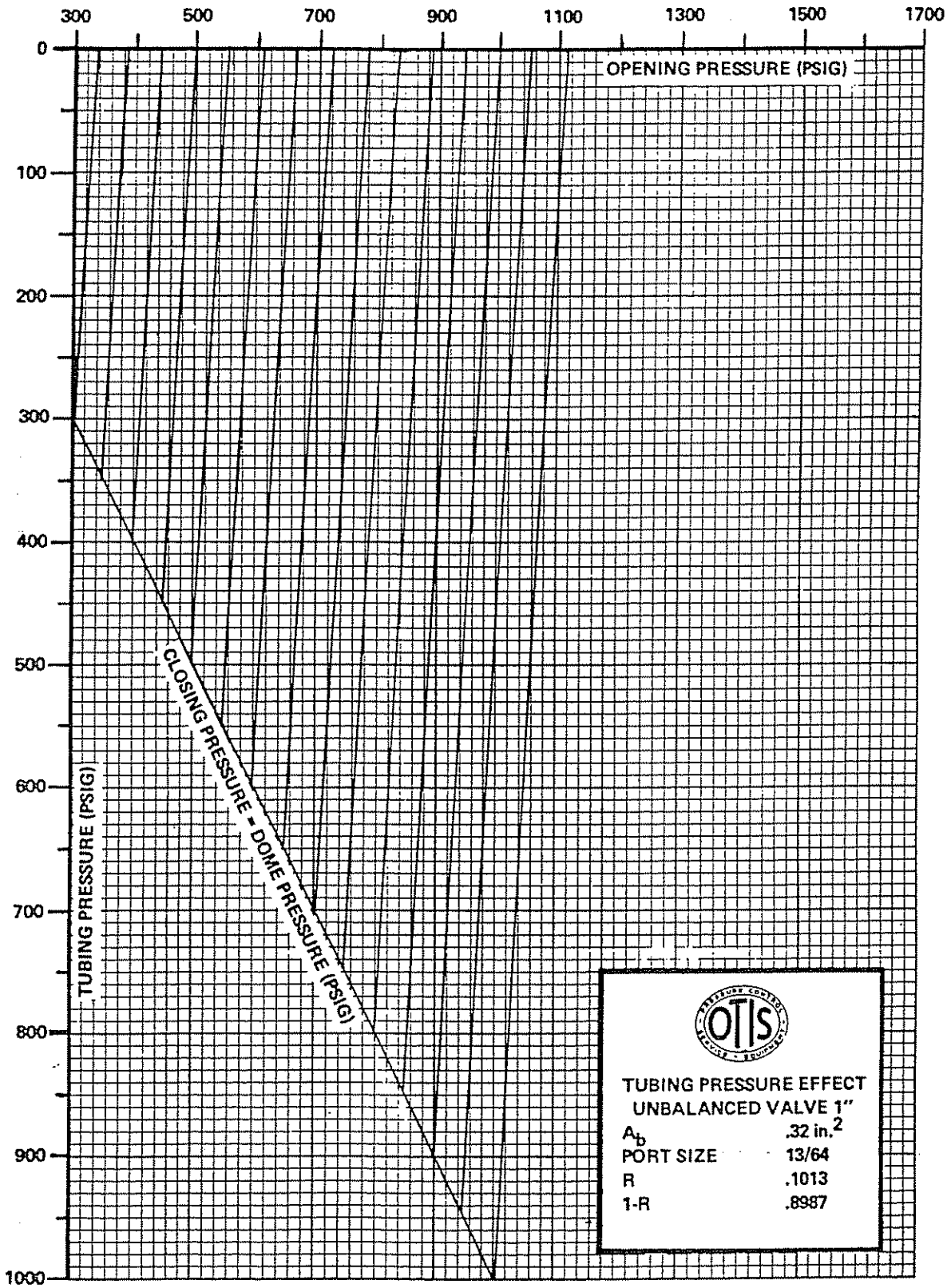


Fig. 3E-10 (Courtesy Otis Engineering Corp.)

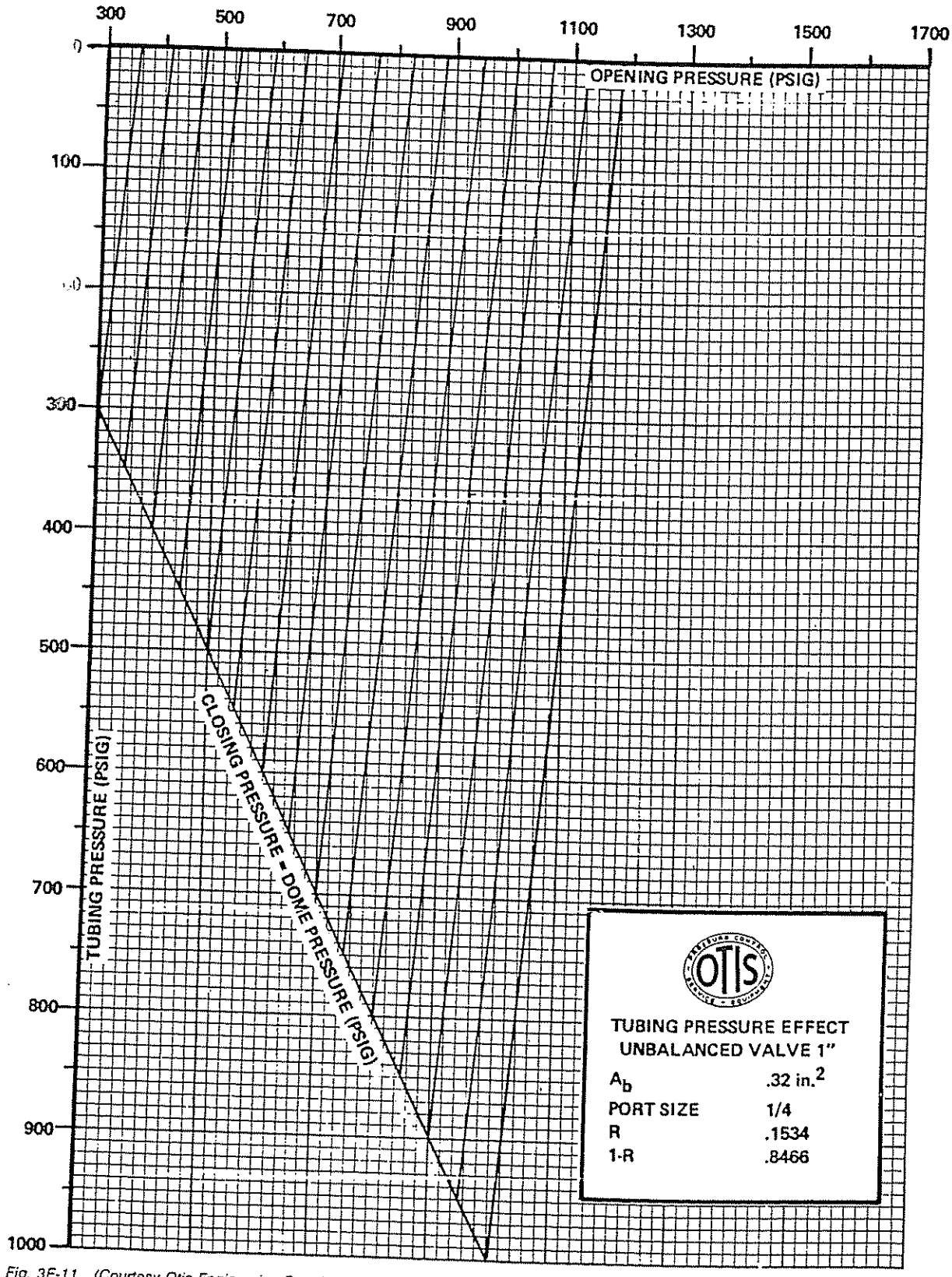


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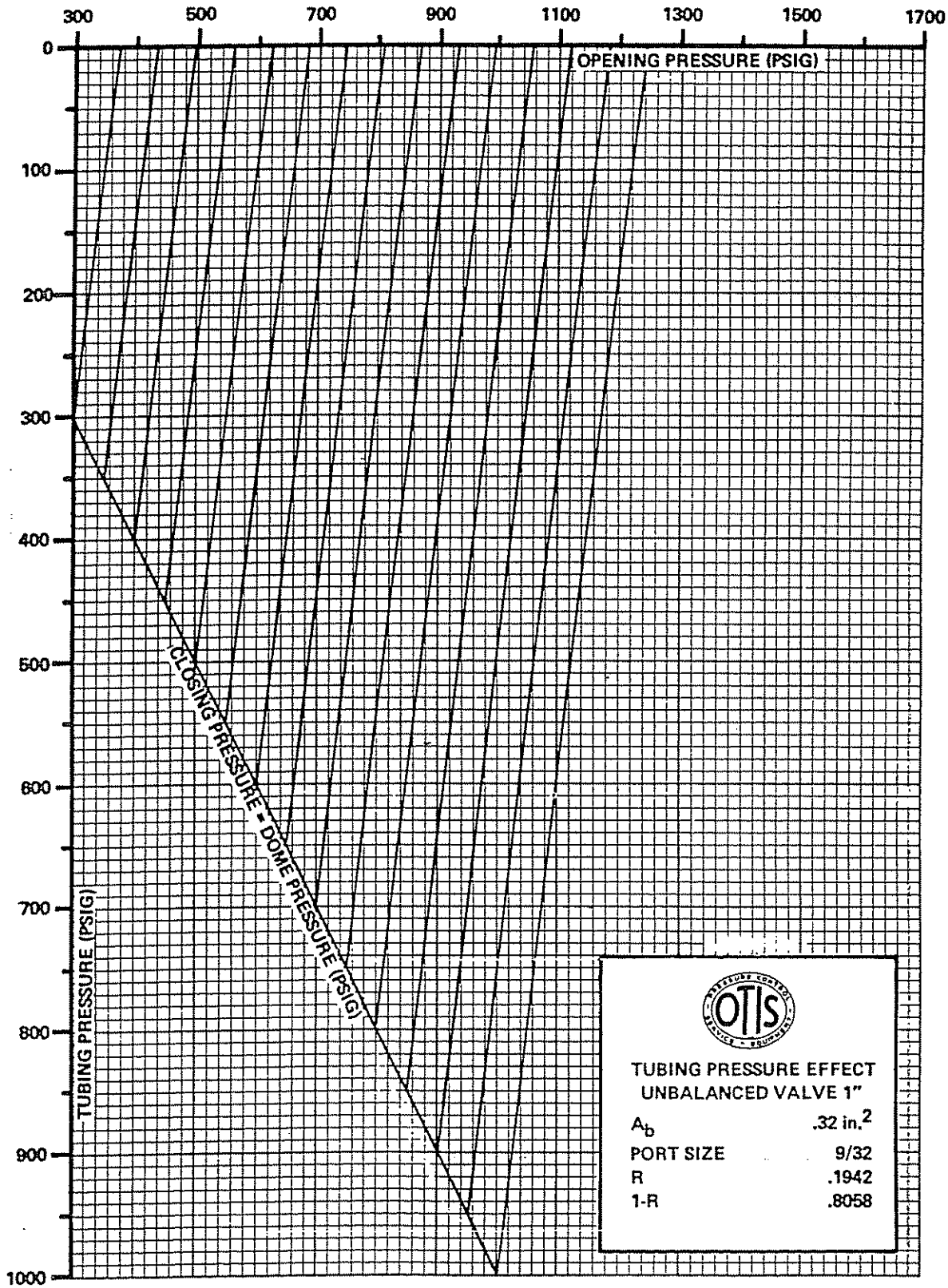


Fig. 3E-12 (Courtesy Otis Engineering Corp.)

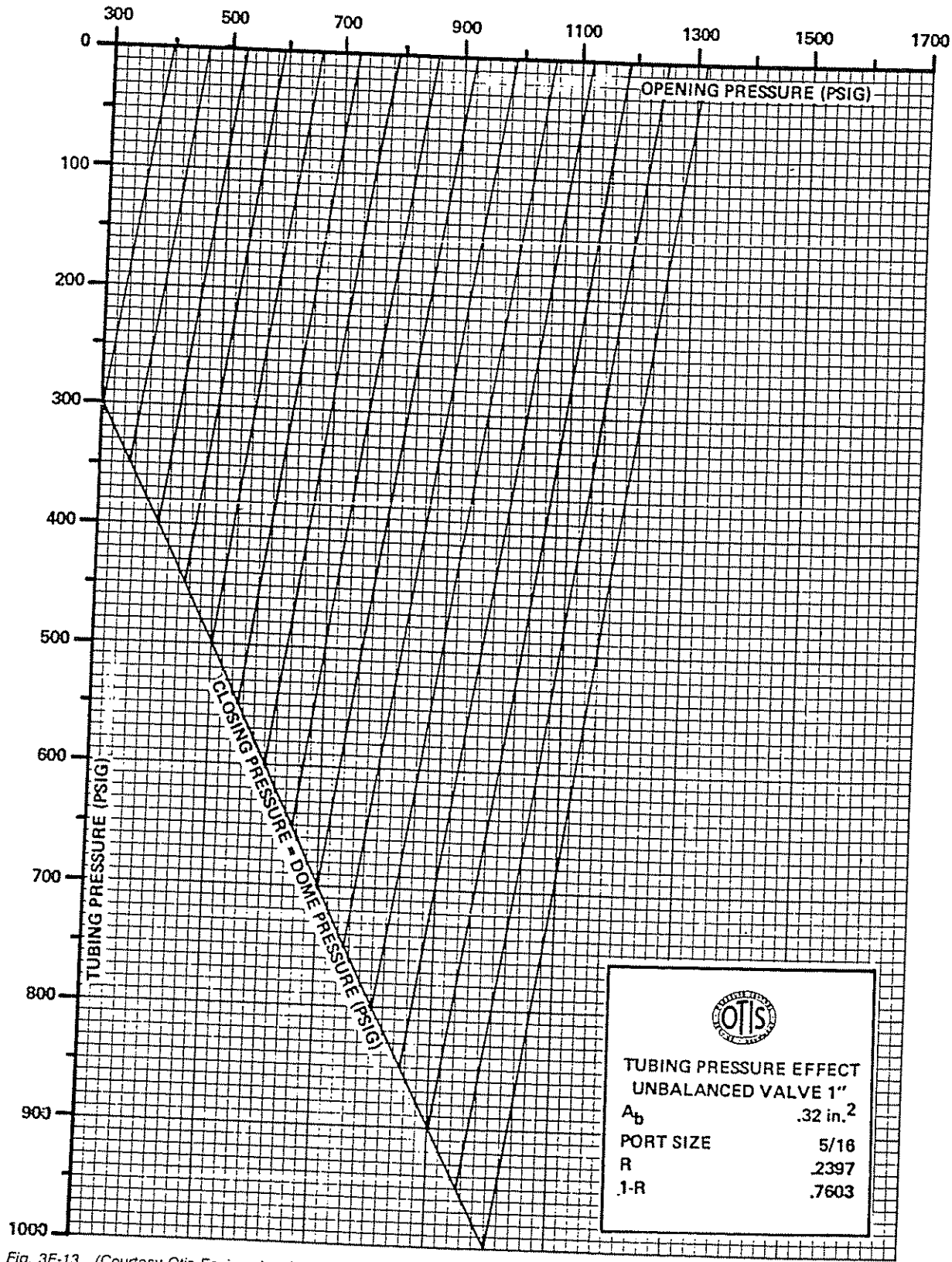


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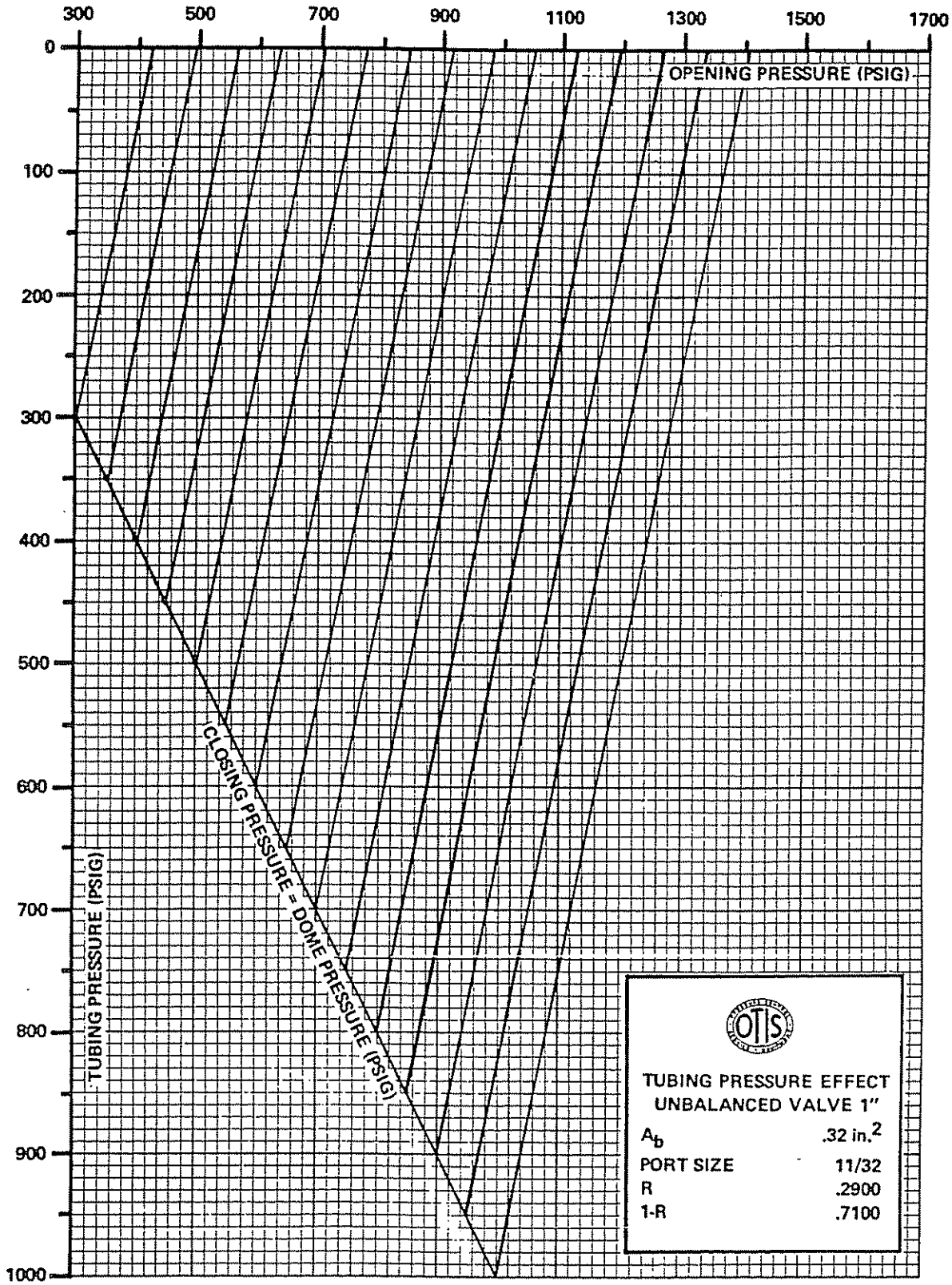


Fig. 3E-14 (Courtesy Otis Engineering Corp.)

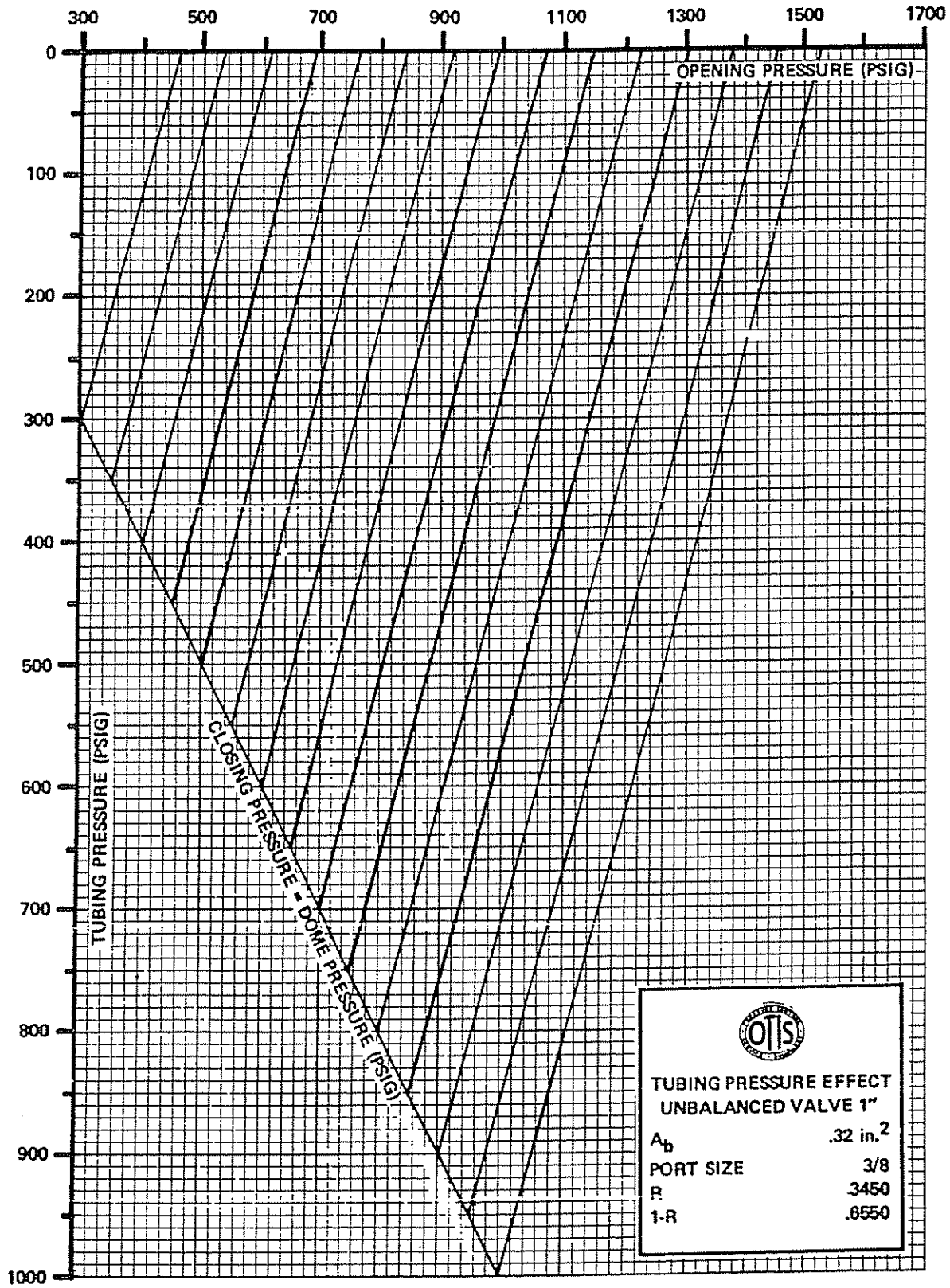


Fig. 3E-15 (Courtesy Otis Engineering Corp.)

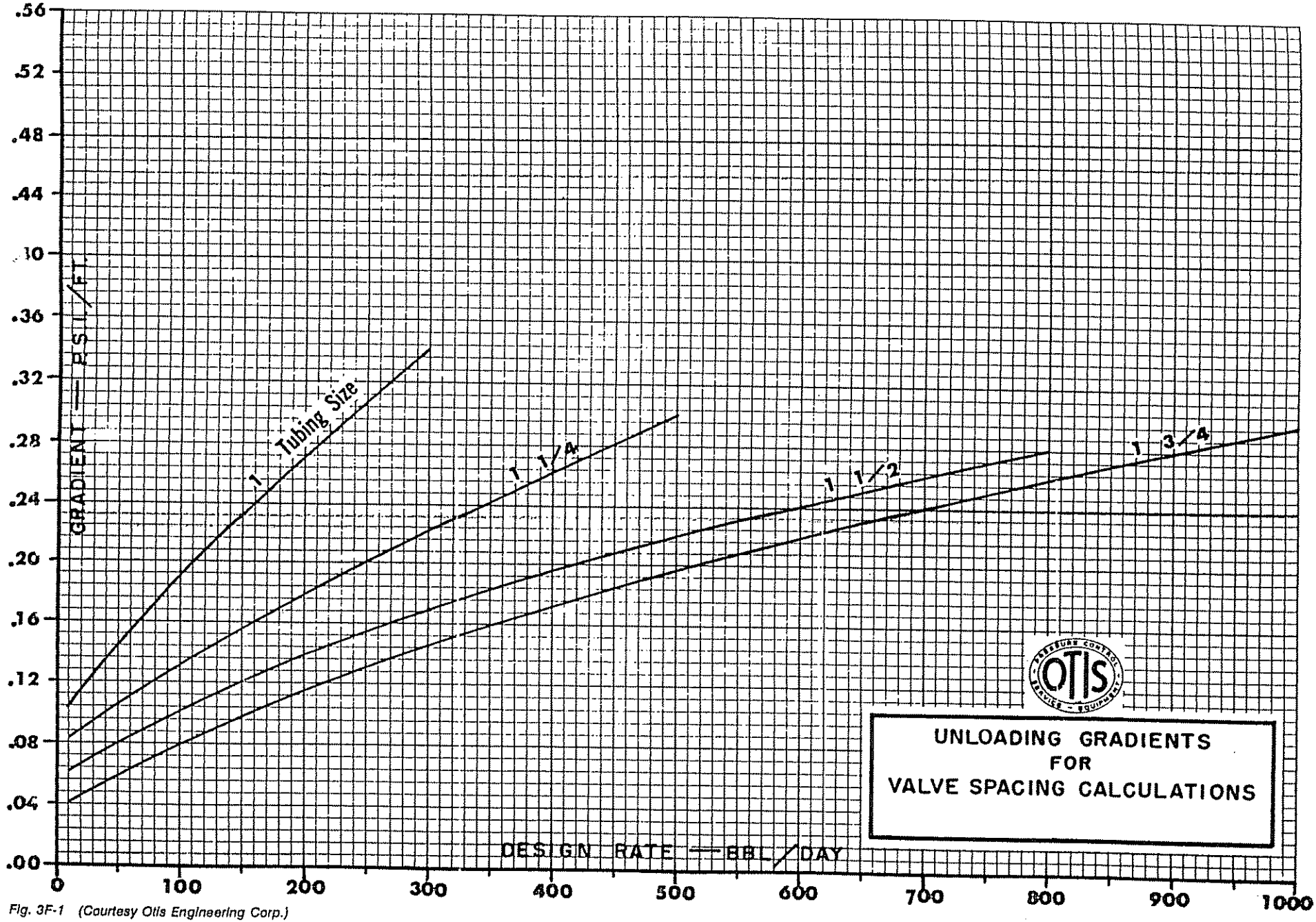
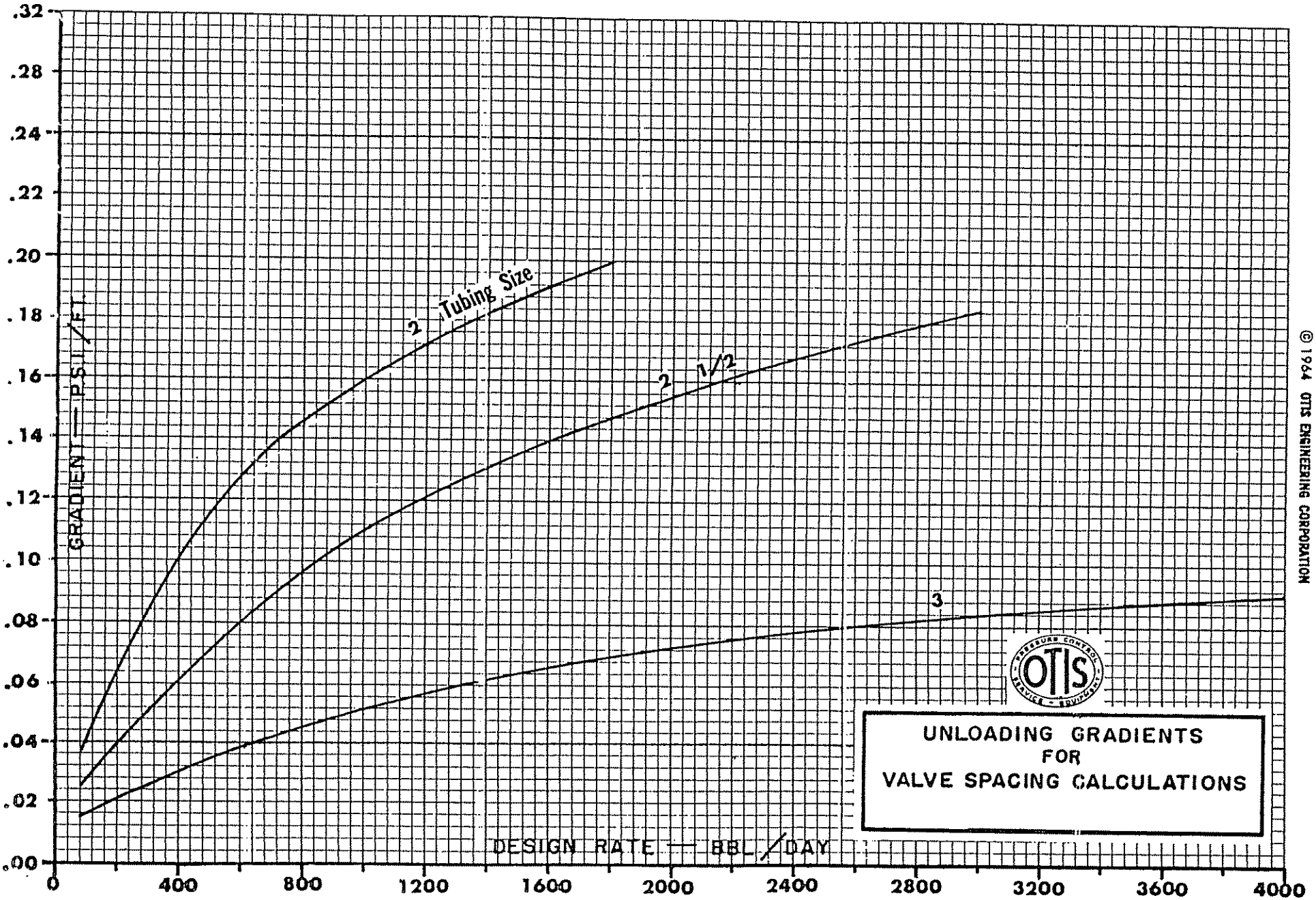


Fig. 3F-1 (Courtesy Otis Engineering Corp.)



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Fig 3F-2 (Courtesy Otis Engineering Corp.)

EXPLANATION OF FIGURES 3G.1 THROUGH 3G.16 ENTITLED "Intermittent Gas Volume Requirements"

The gas volume requirements were determined by calculating the number of SCF of gas necessary to fill the tubing string less the remaining slug length in the tubing as it reaches the surface at the average pressure and temperature underneath the liquid slug from bottom to top.

The following constants were utilized:

(1) Surface temperature — 80°F
(2) Temperature gradient — 1.6°F/100 feet
(3) Gas specific gravity — 0.65
(4) Gas deviation was accounted for
(5) A 35° API oil was lifted
(6) The gas pressure placed underneath the slug at the operating valve is the opening pressure of the valve.
(7) The gas pressure in the tubing as the slug is being produced at the surface is the tubing back pressure plus the hydrostatic head pressure of the remaining slug.
(8) The difference between the tubing load to be lifted and operating lift pressure is 200 psi.
(9) The fall-back is assumed to be 7% per 1000.
(10) The volume that the liquid fall-back in the tubing string occupies is neglected and assumed to be occupied by gas.
(11) The gas column weight in the tubing string was accounted for in order to determine the initial length of the liquid slug.

Recommended procedure for use of Fig Series 3G

(1) Calculate valve opening pressure at depth (P_{vo})
(2) Select correct Figure in accordance with tubing size and separator pressure.
(3) Beginning at depth of valve on vertical scale, proceed horizontally to the valve opening pressure at depth (P_{vo}) determined in step 1.
(4) Proceed vertically to read total gas volume requirement on upper horizontal scale.

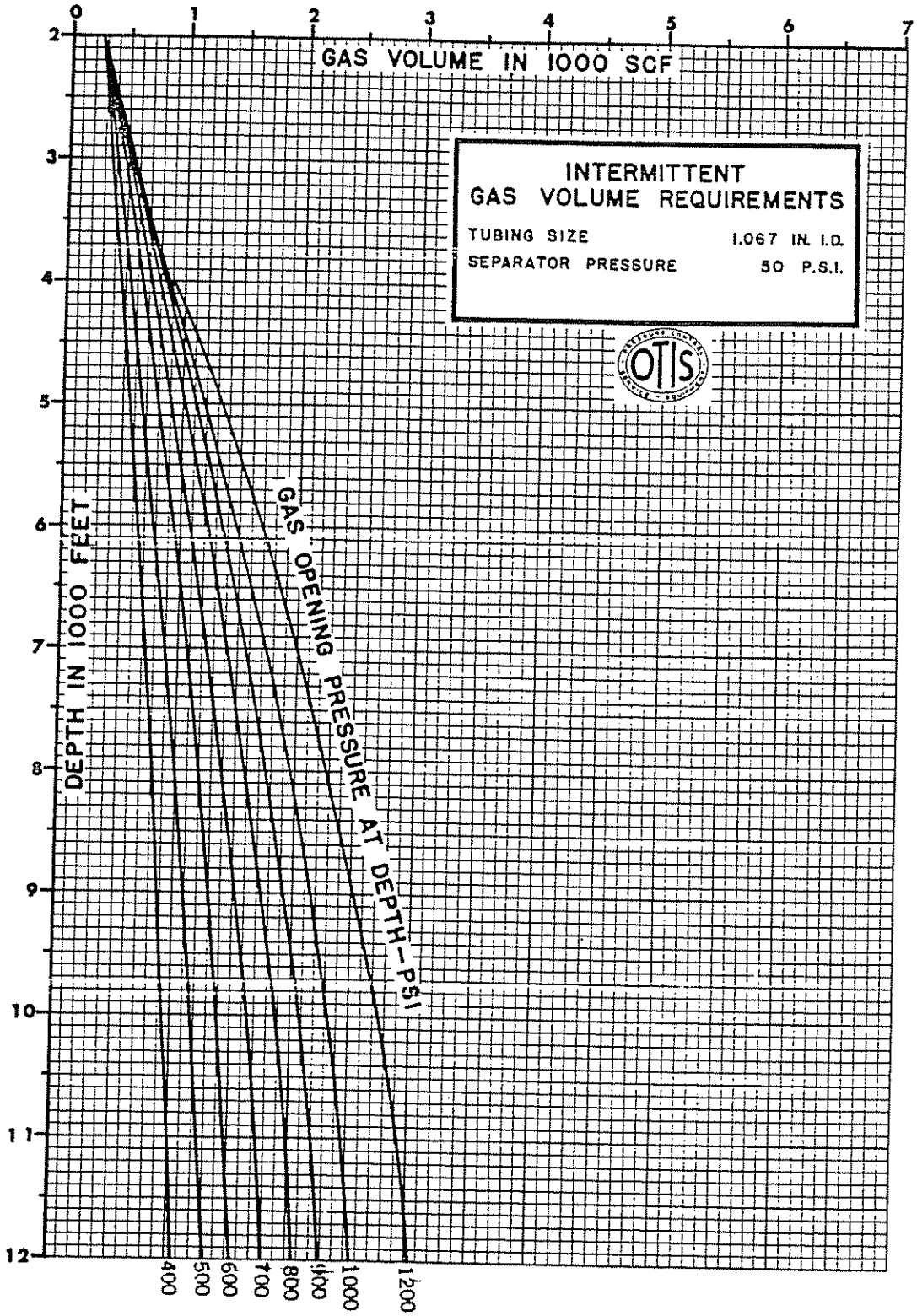


Fig. 3G-1 (Courtesy Otis Engineering Corp.

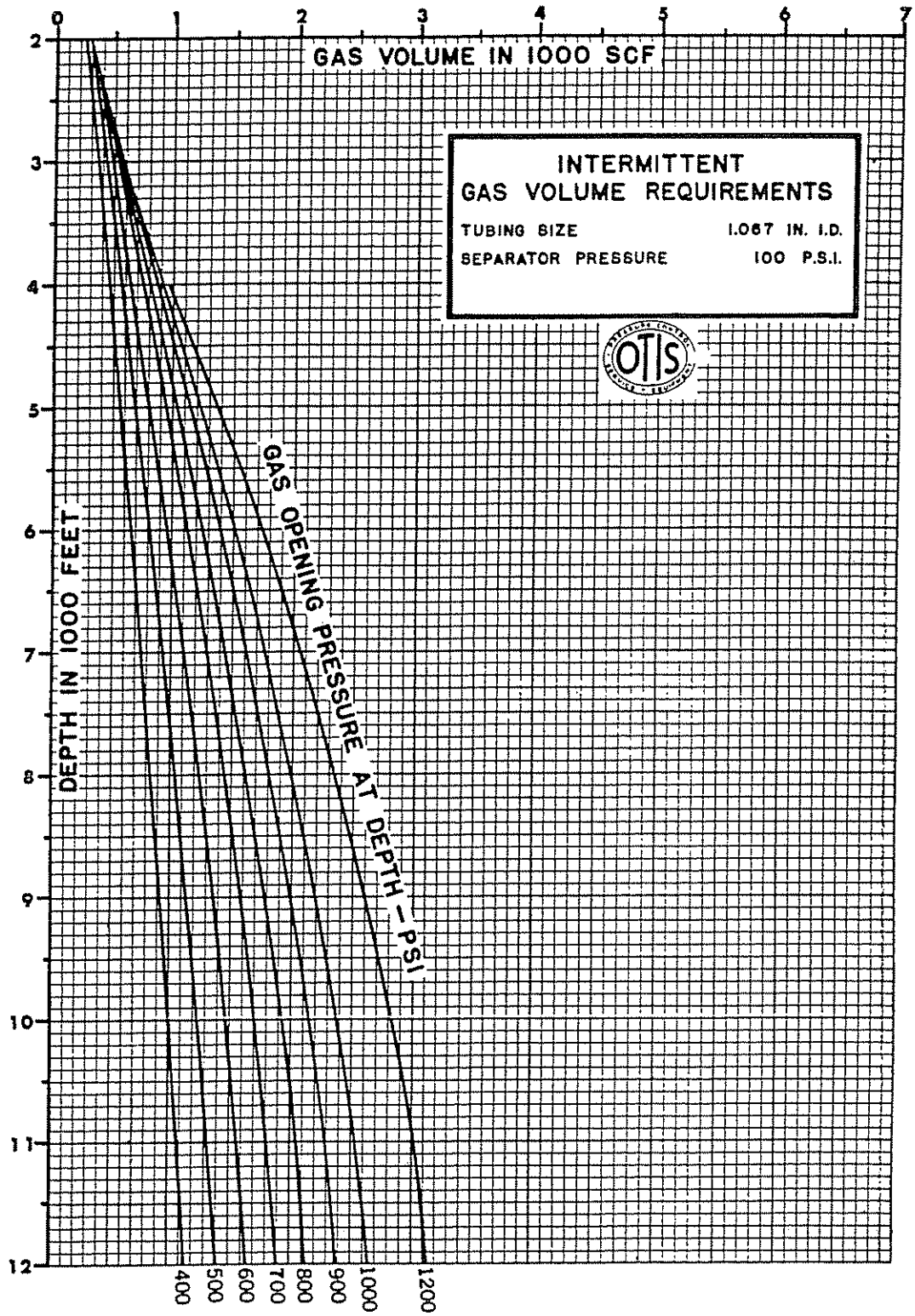


Fig. 3G-2 (Courtesy Otis Engineering Corp.)

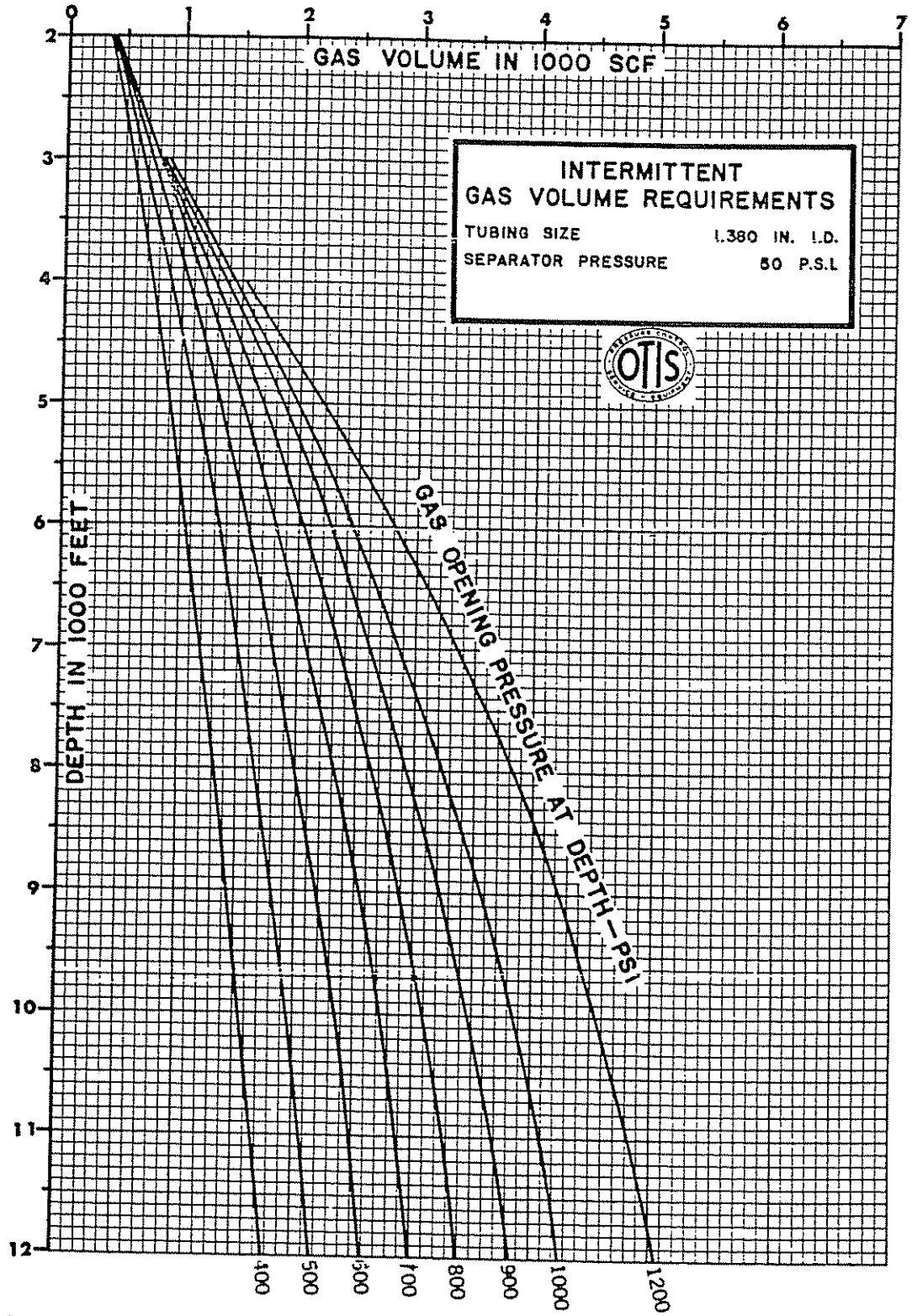


Fig. 3G-3 (Courtesy Otis Engineering Corp.)

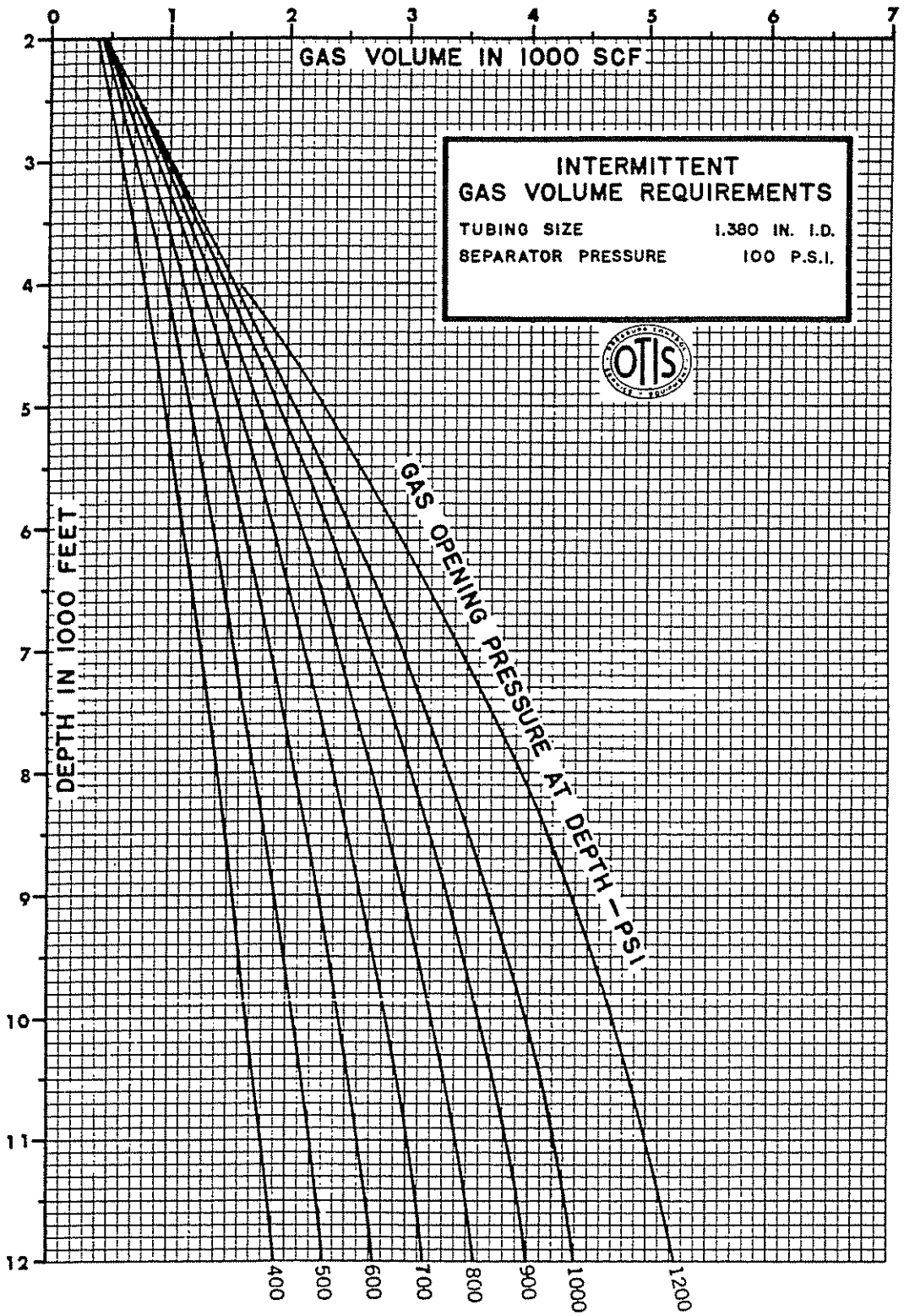


Fig. 3G-4 (Courtesy Otis Engineering Corp.)

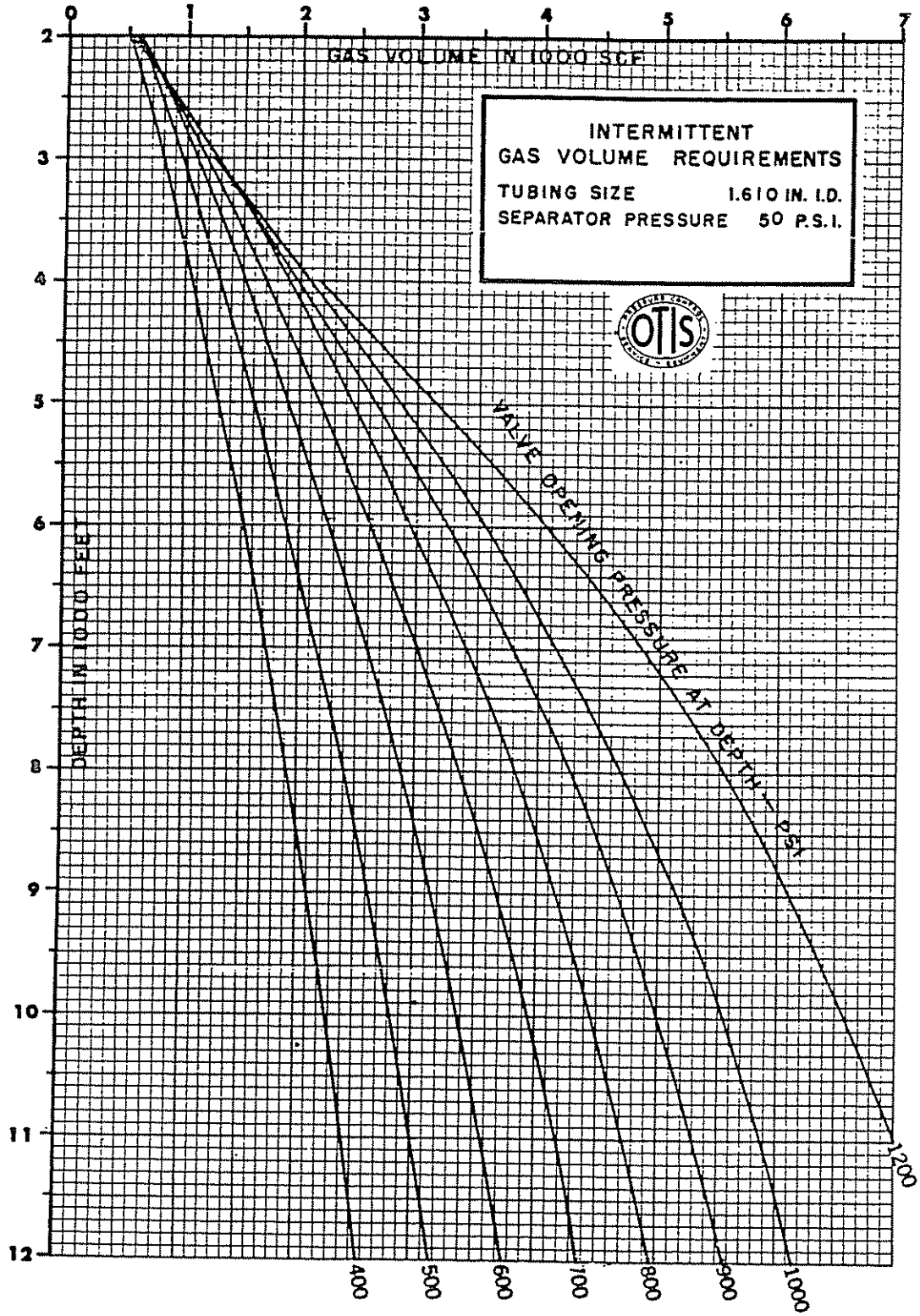


Fig. 3G-5 (Courtesy Otis Engineering Corp.)

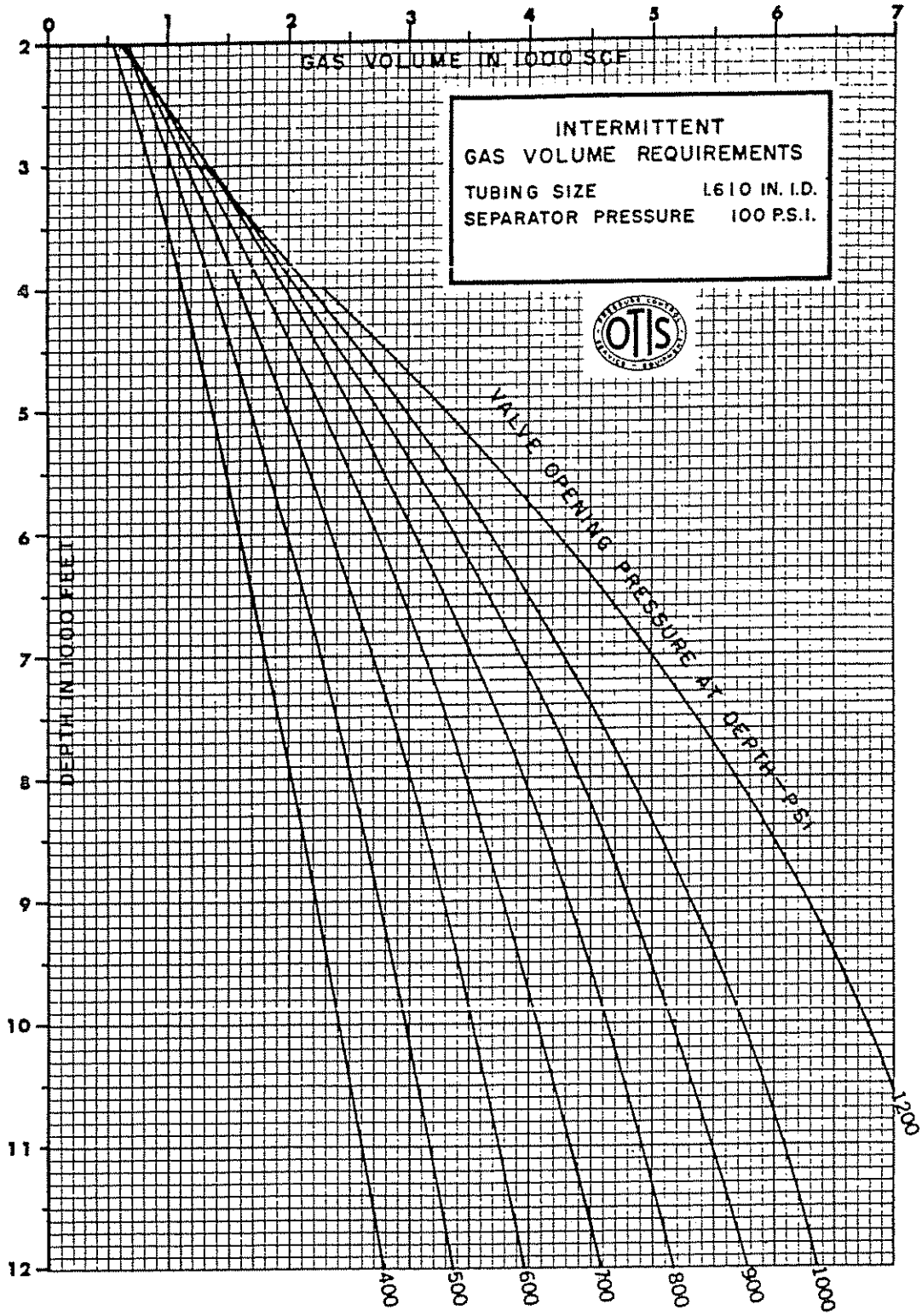


Fig. 3G-6 (Courtesy Otis Engineering Corp.)

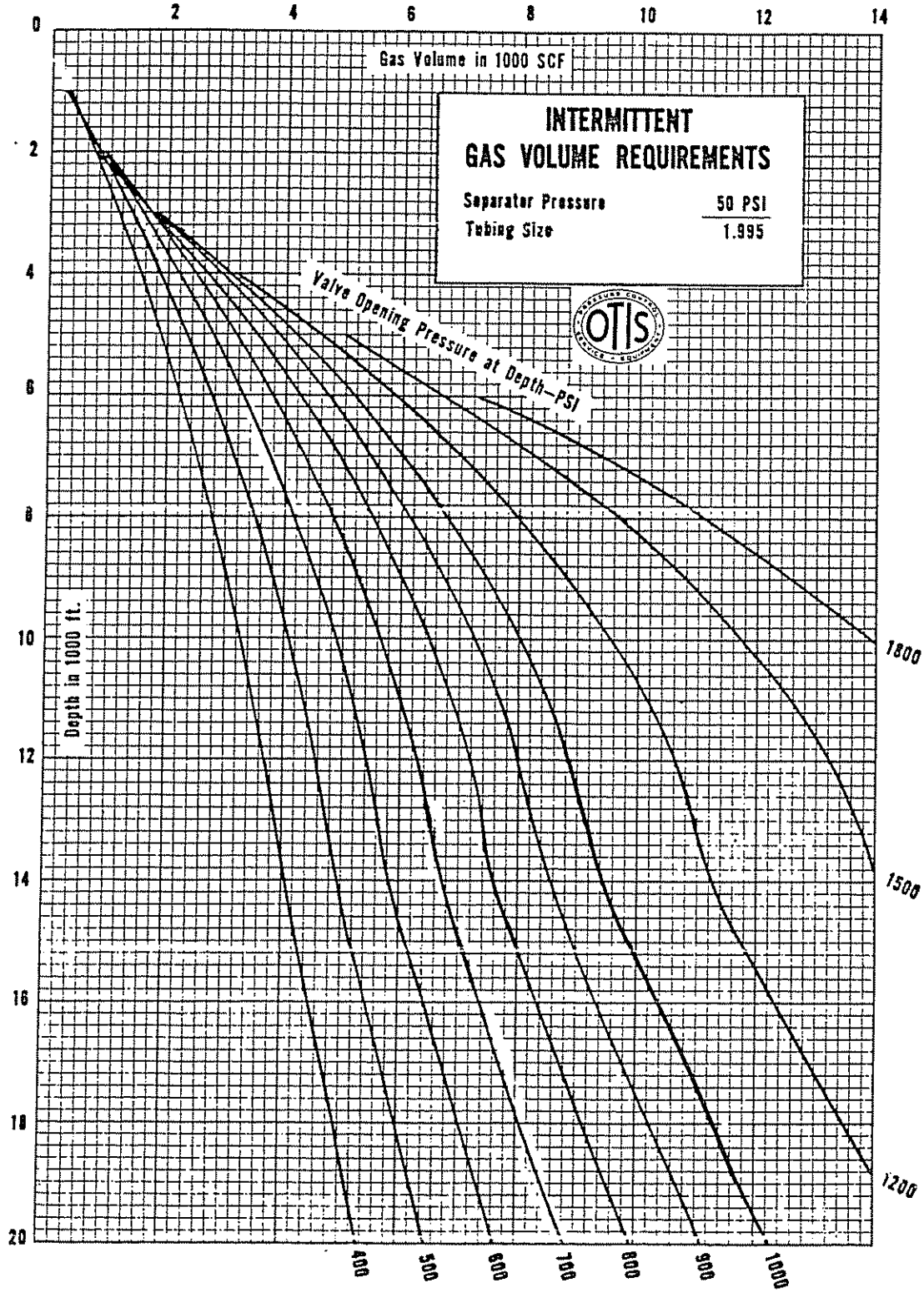


Fig. 3G-7 (Courtesy Otis Engineering Corp.)

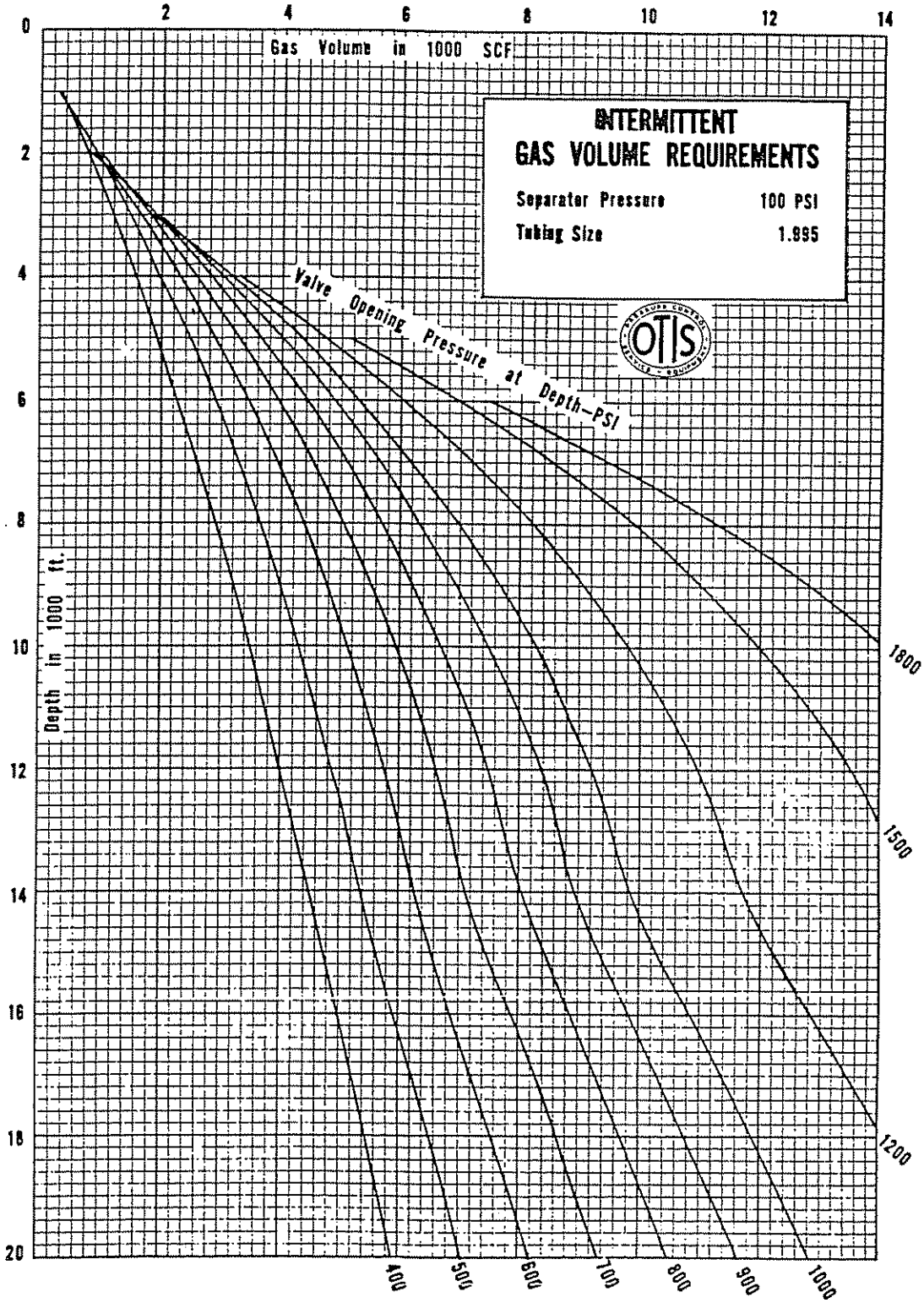


Fig. 3G-8 (Courtesy Otis Engineering Corp.)

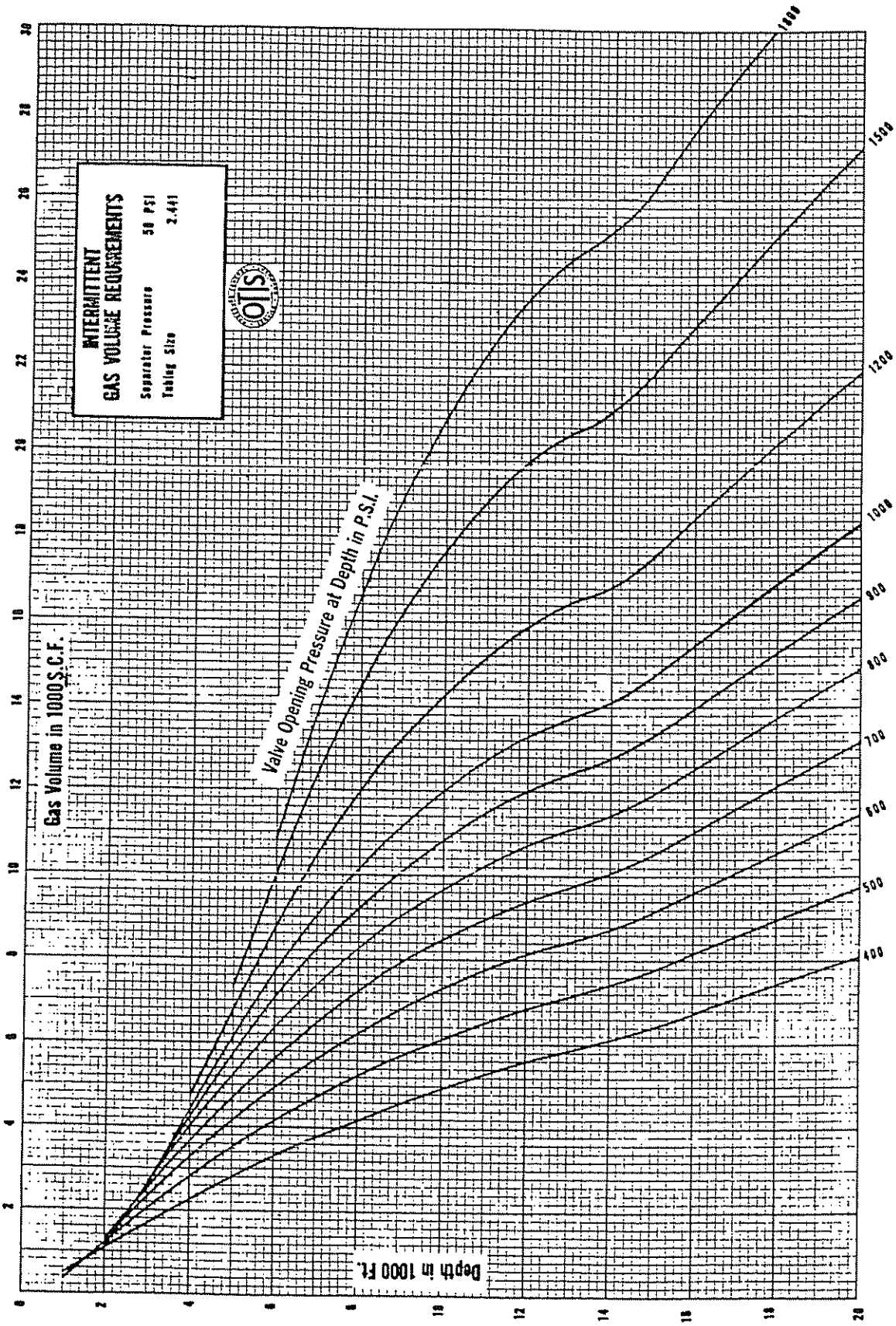


Fig. 3G-9 (Courtesy Otis Engineering Corp.)

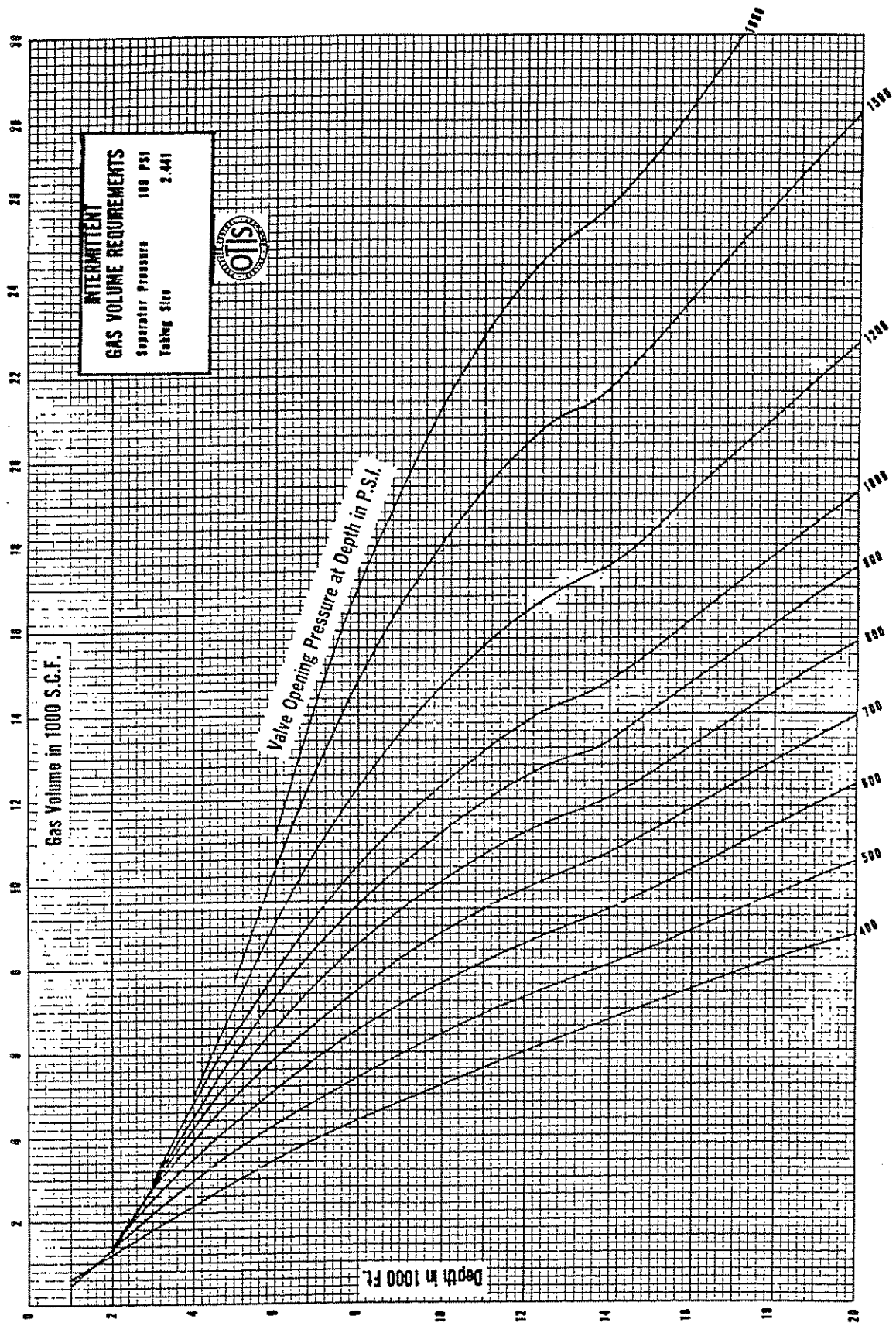


Fig. 3G-10 (Courtesy Otis Engineering Corp.)

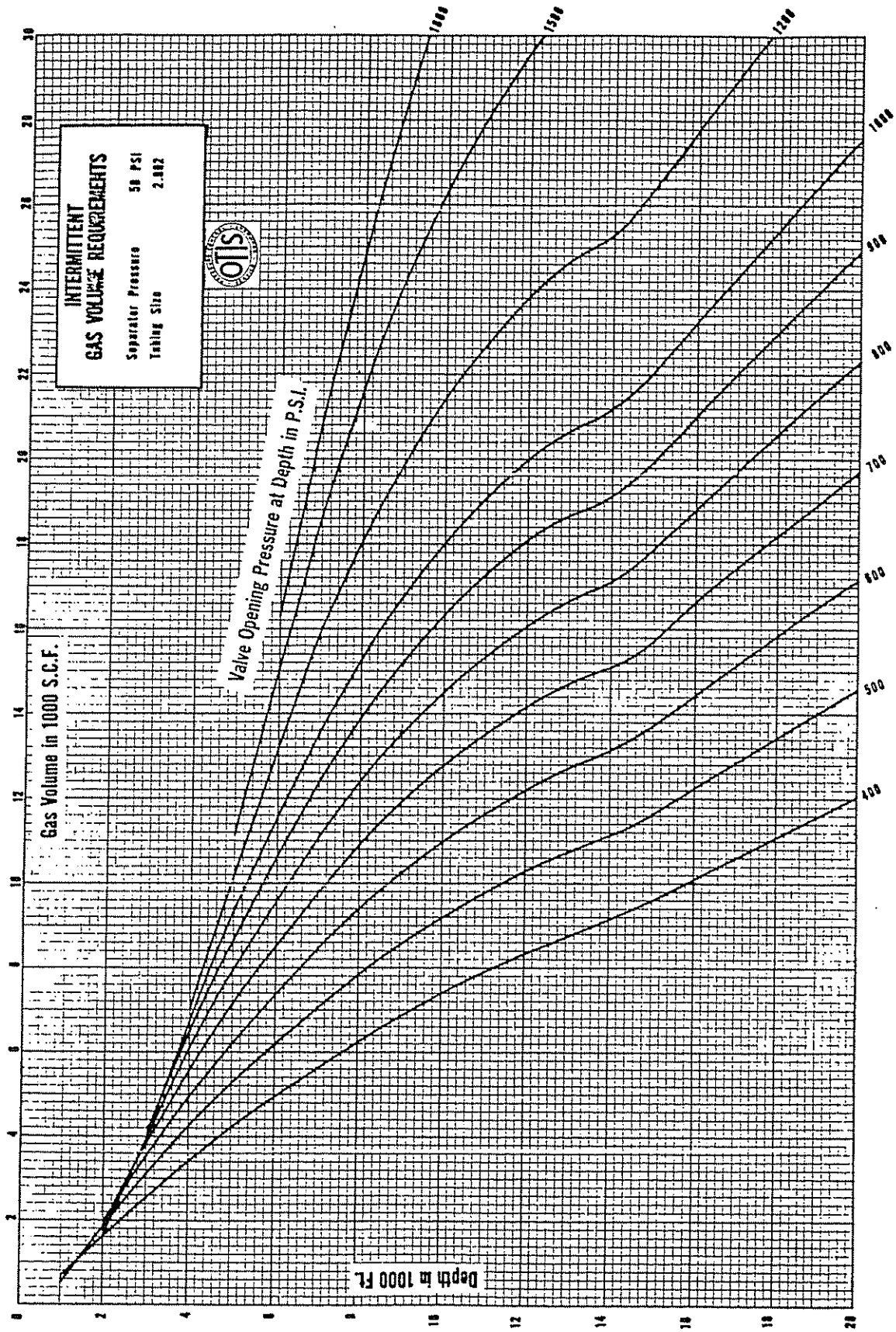


Fig. 3G-11 (Courtesy Otis Engineering Corp.)

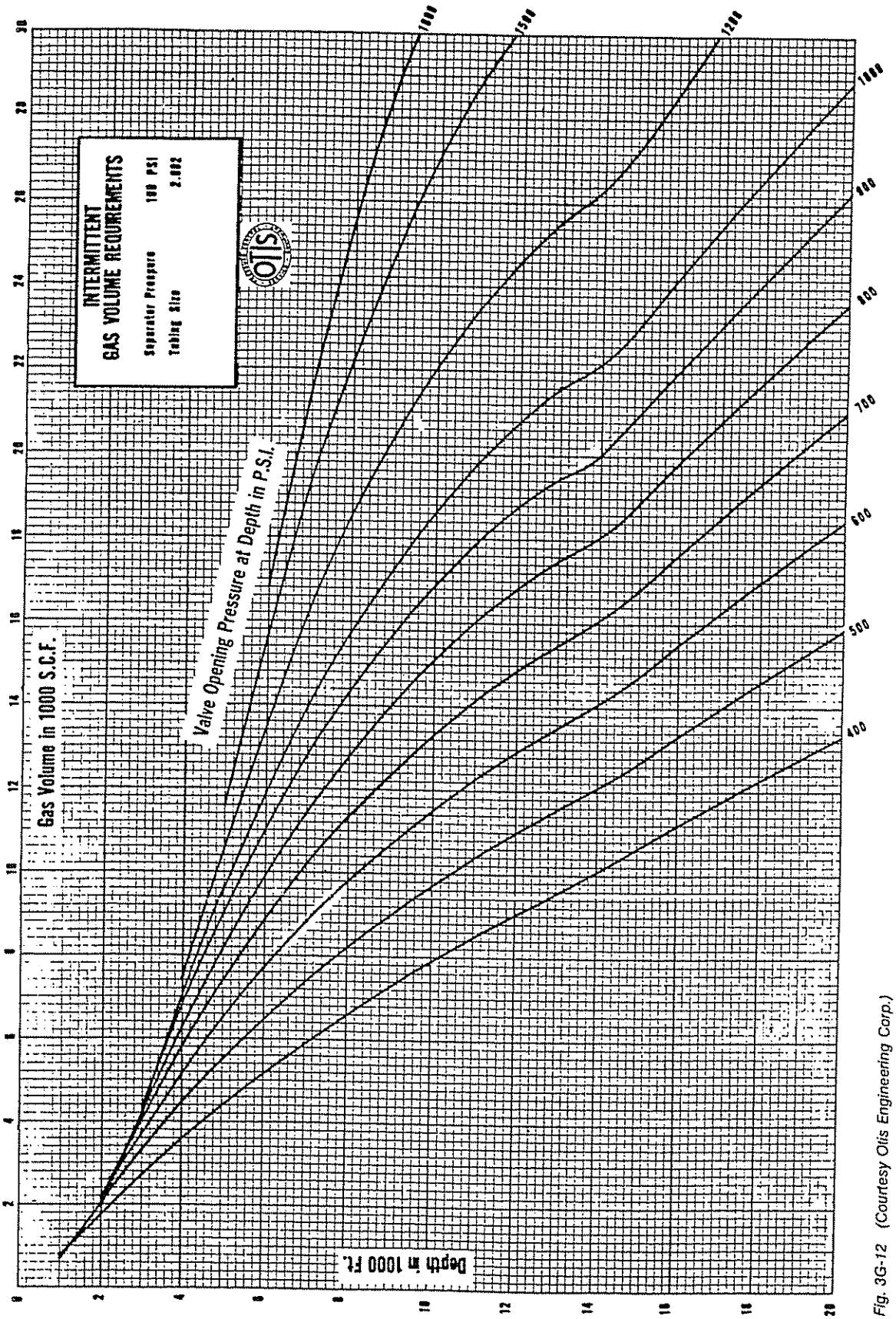


Fig. 3G-12 (Courtesy Otis Engineering Corp.)

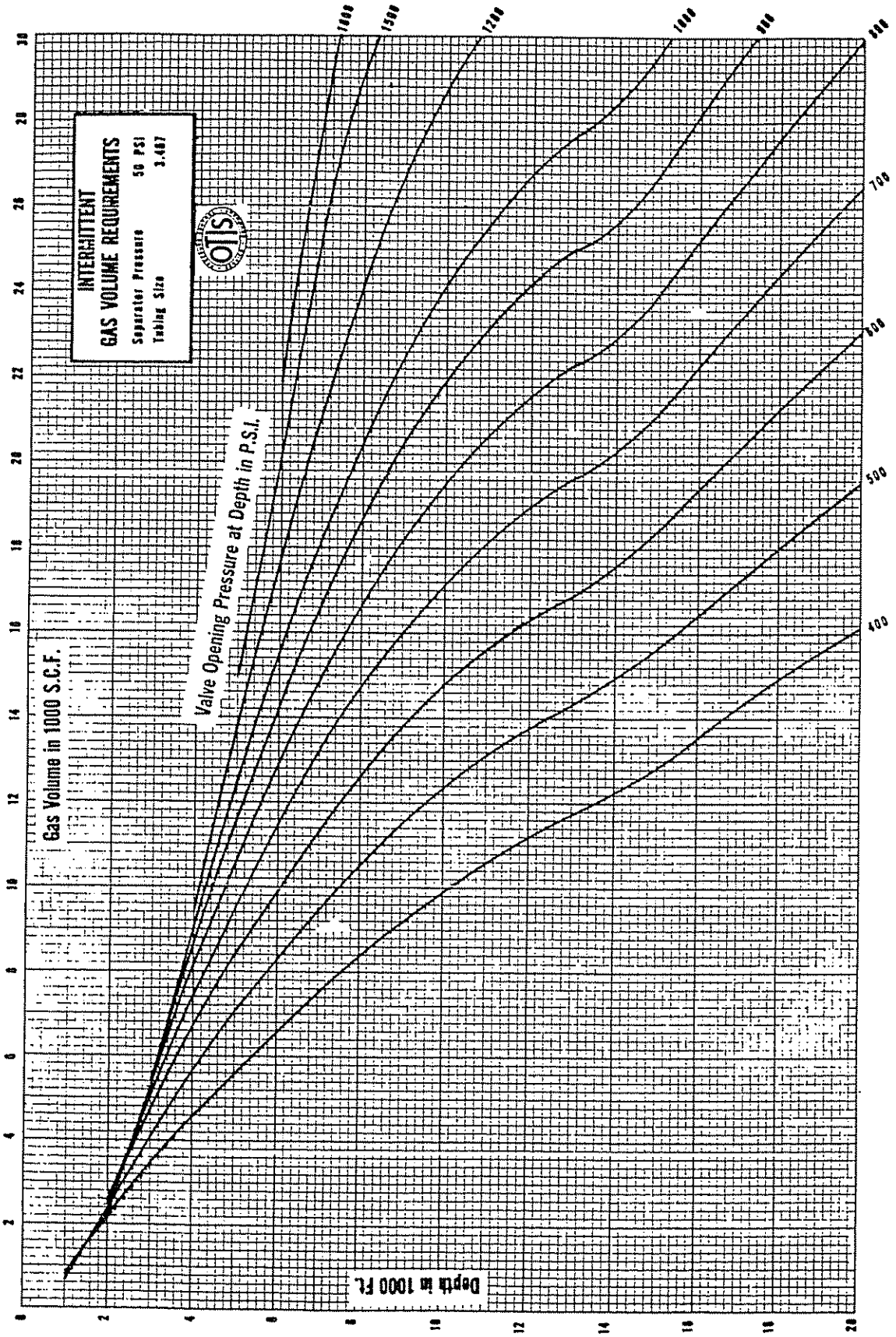


Fig. 3G-13 (Courtesy Otis Engineering Corp.)

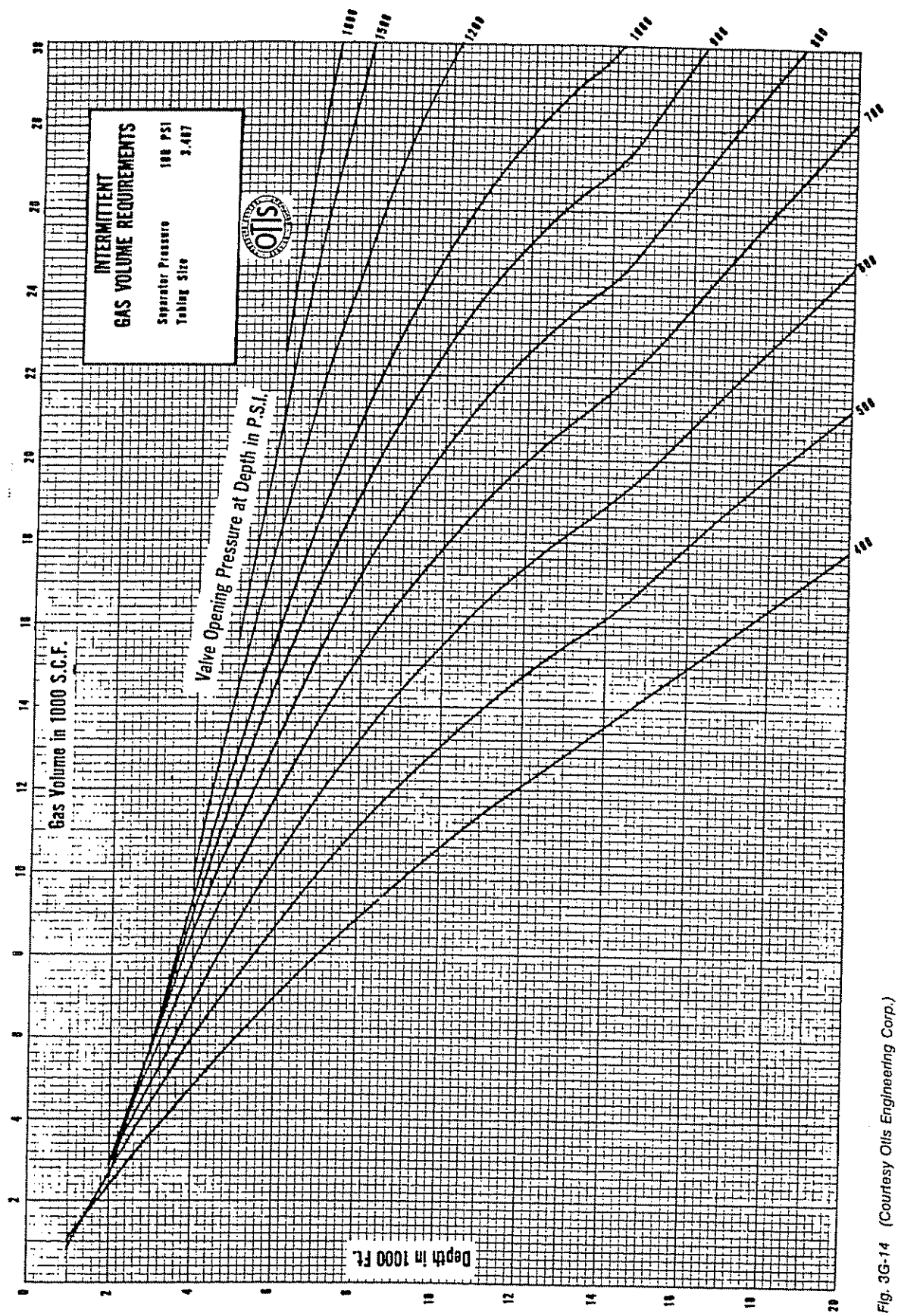


Fig. 3G-14 (Courtesy Otis Engineering Corp.)

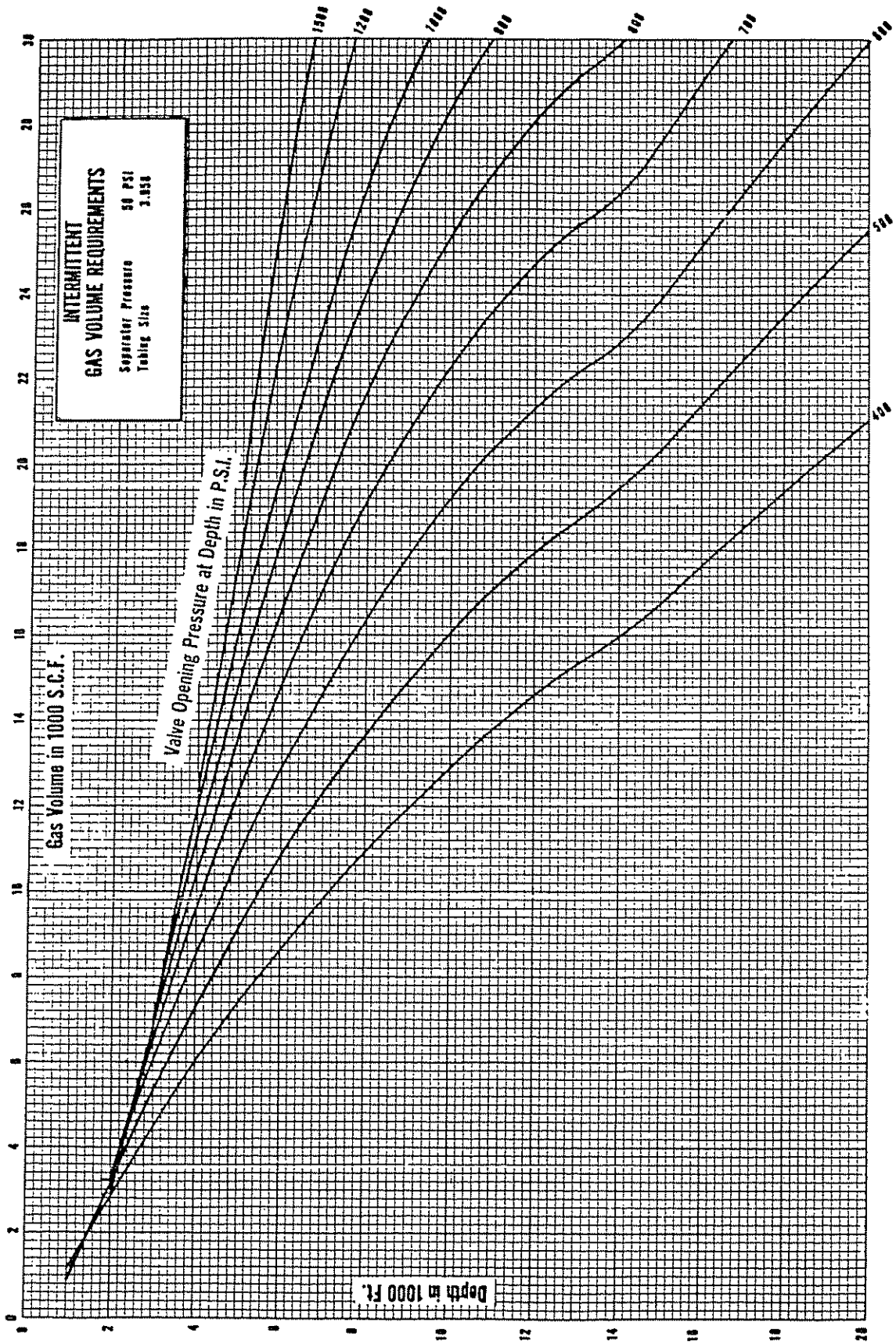


Fig. 3G-15 (Courtesy Otis Engineering Corp.)

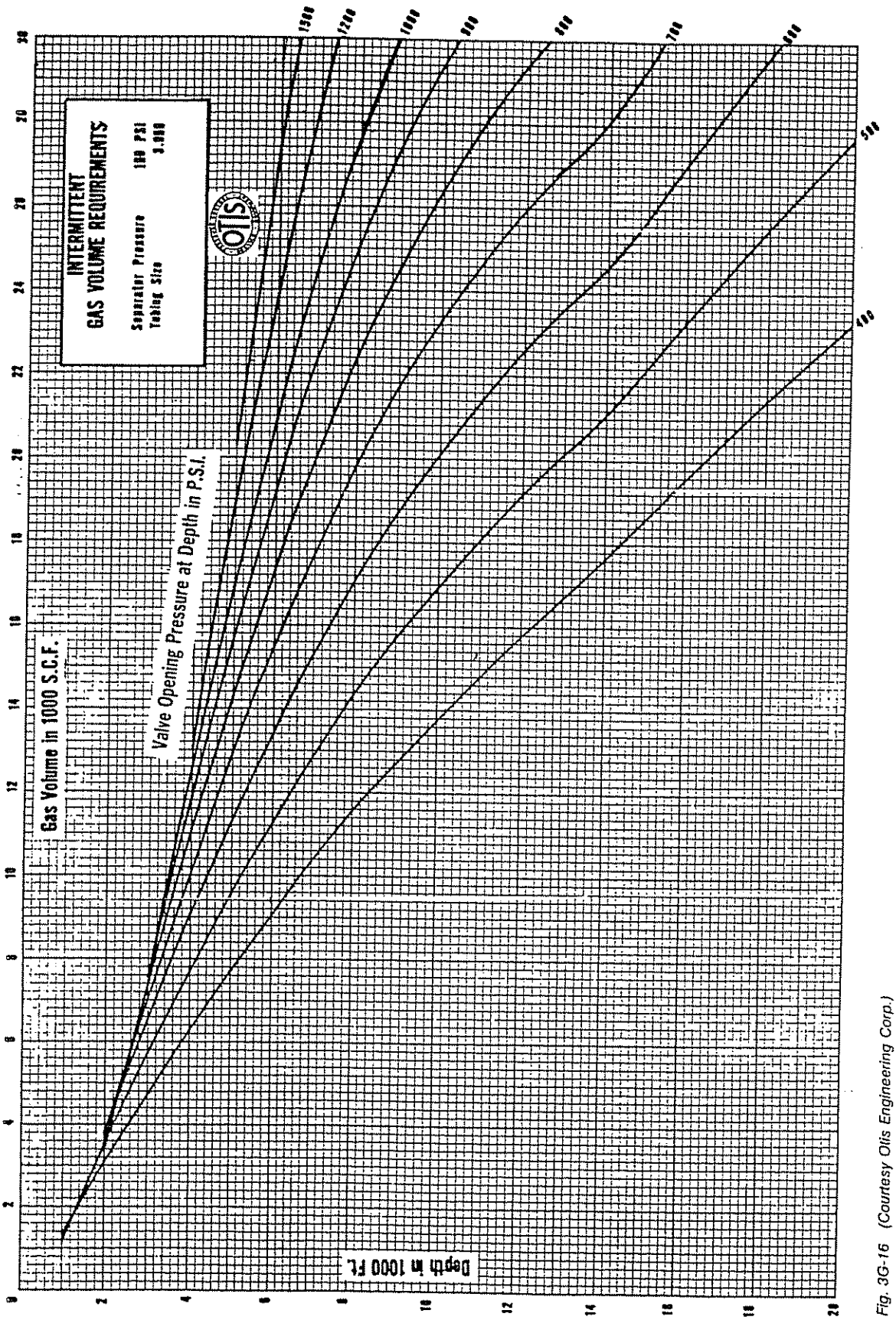


Fig. 3G-16 (Courtesy Otis Engineering Corp.)

EXPLANATION OF FIGURES ENTITLED "Annular Pressure Reduction" Figure Series 3H

These charts are prepared for ease in determining the amount of annular pressure drop that will be required to transfer a given quantity of gas into the tubing string when designing for input choke control intermitting. NOTE: A dual nomenclature such as 1½ x 1½ on a tubing size line indicates parallel dual tubing strings.

The step by step procedure to use Fig Series 3H would be as follows:

(1) Determine by calculation or by use of Fig Series 3G the total quantity of gas per cycle required to lift the liquid head.
(2) Calculate the necessary SCF/1000' by dividing the total amount of gas required per cycle by the depth of the valve in thousands of feet. Example: if the total gas requirement per cycle determined in step 1, above was 4000 SCF and the depth of the valve is 5000 feet, then

$$\text{SCF/1000} = \frac{4000}{5000}(1000) = 800 \text{ SCF/1000'}$$

(3) Select the correct Figure from Series 3G in accordance with the casing size.
(4) Proceed vertically from the SCF/1000' calculation in step 2 to the correct tubing size. Then from this point read the annular pressure reduction necessary horizontally. This value is useful for selection of correct spread characteristics under operating conditions for unbalanced valves when designing for intermitting lift using only a choke on the surface for control of the input gas.

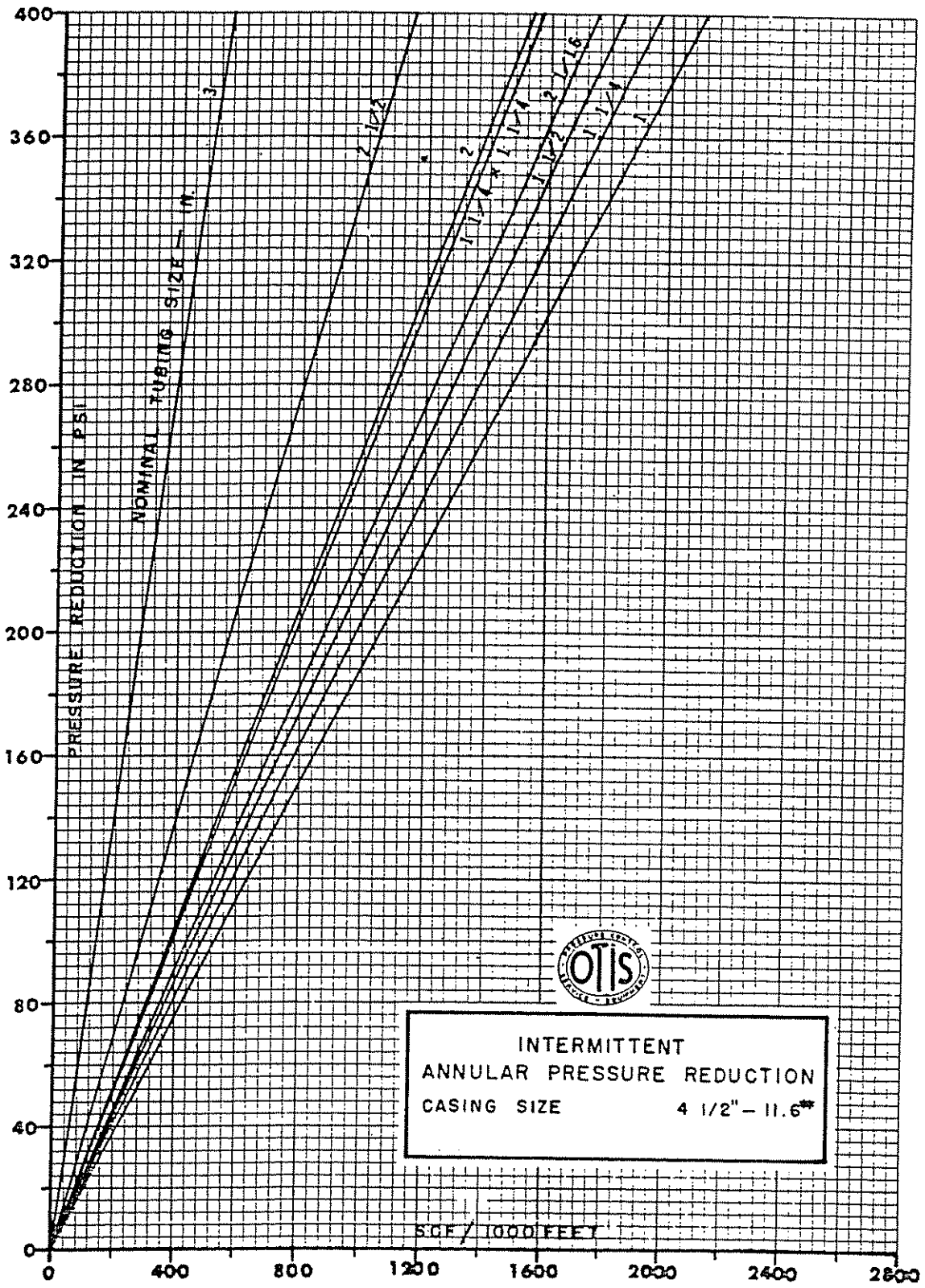


Fig. 3H-1 (Courtesy Otis Engineering Corp.)

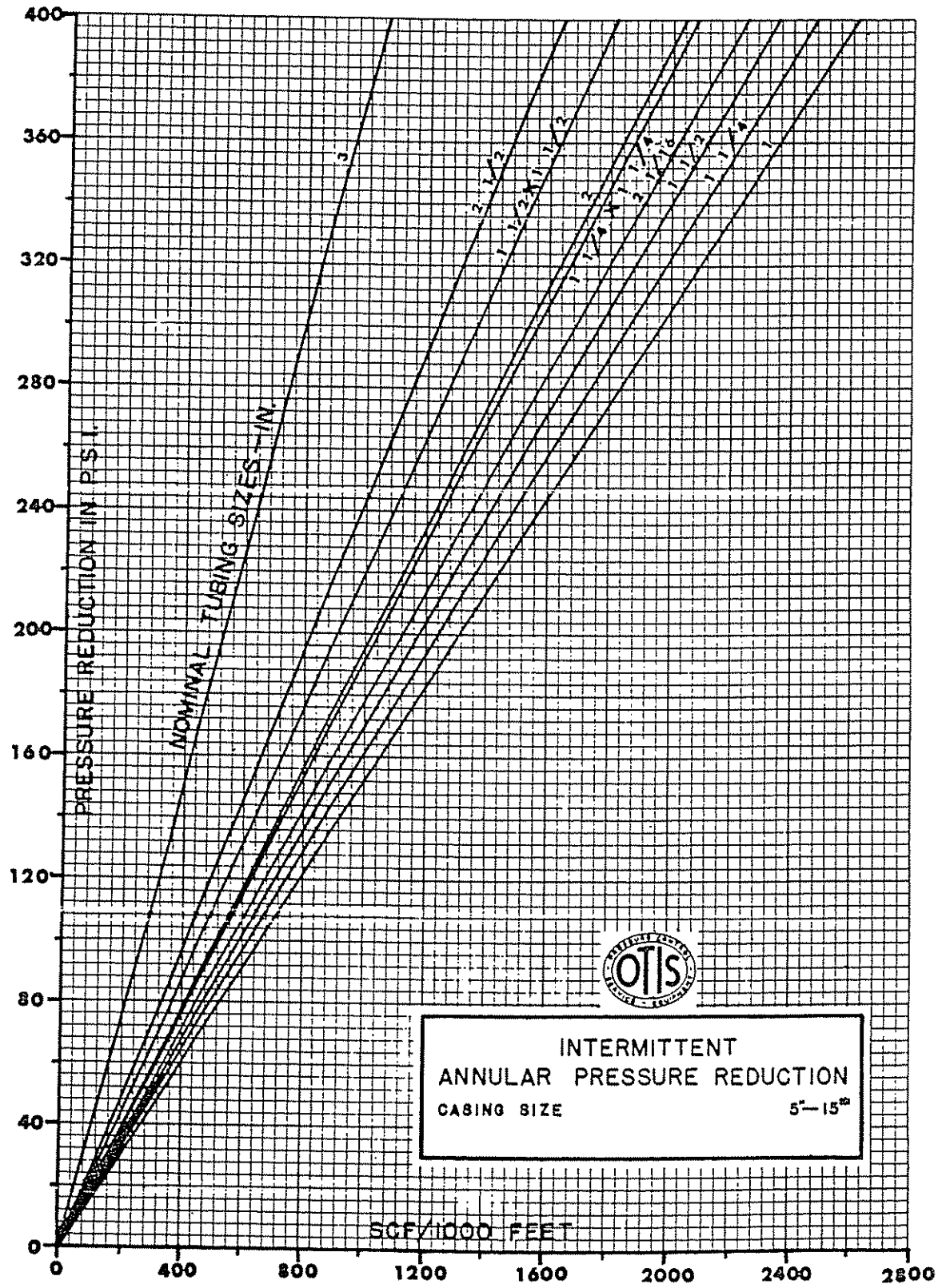


Fig. 3H-2 (Courtesy Otis Engineering Corp.)

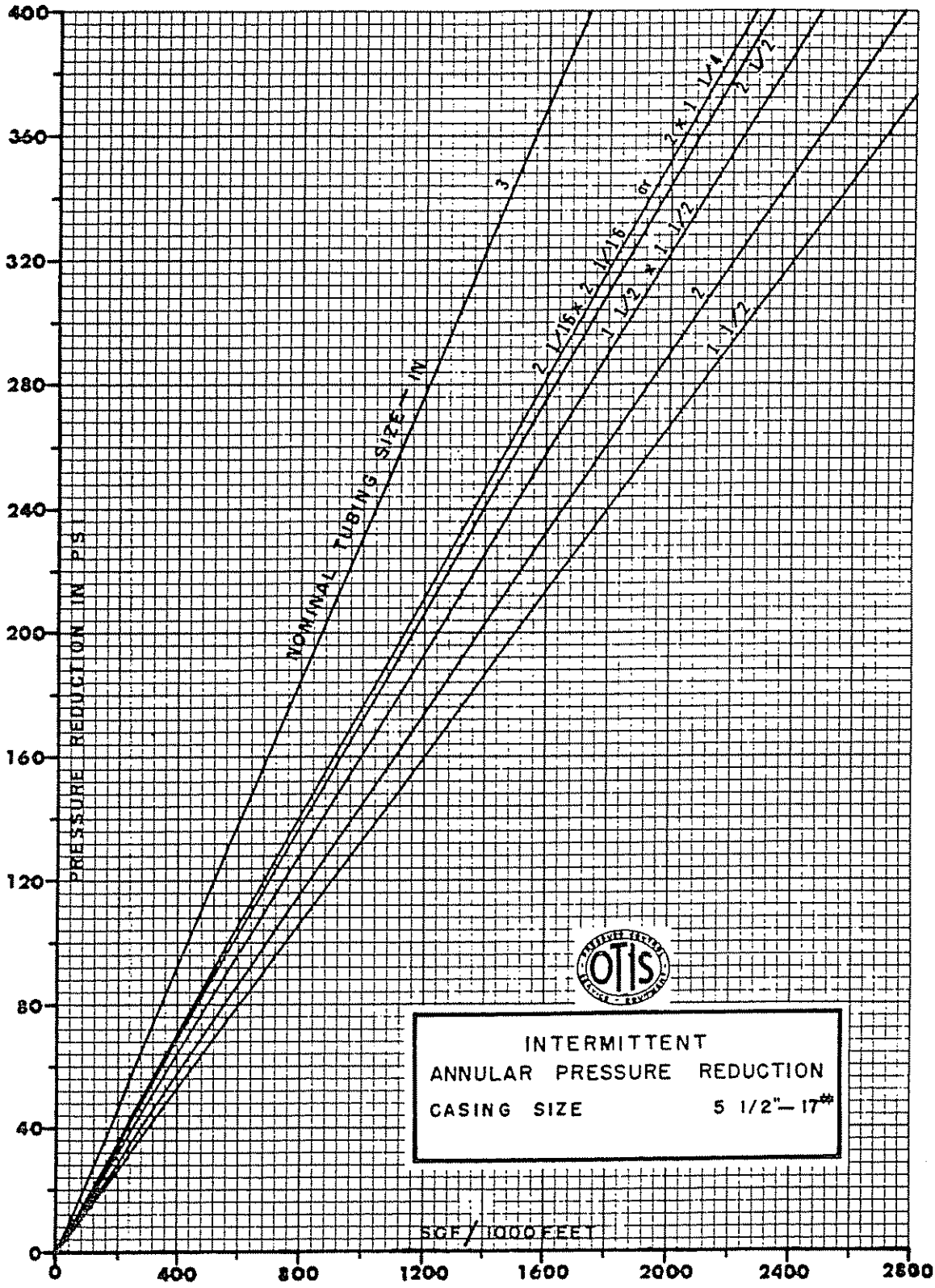


Fig. 3H-3 (Courtesy Otis Engineering Corp.)

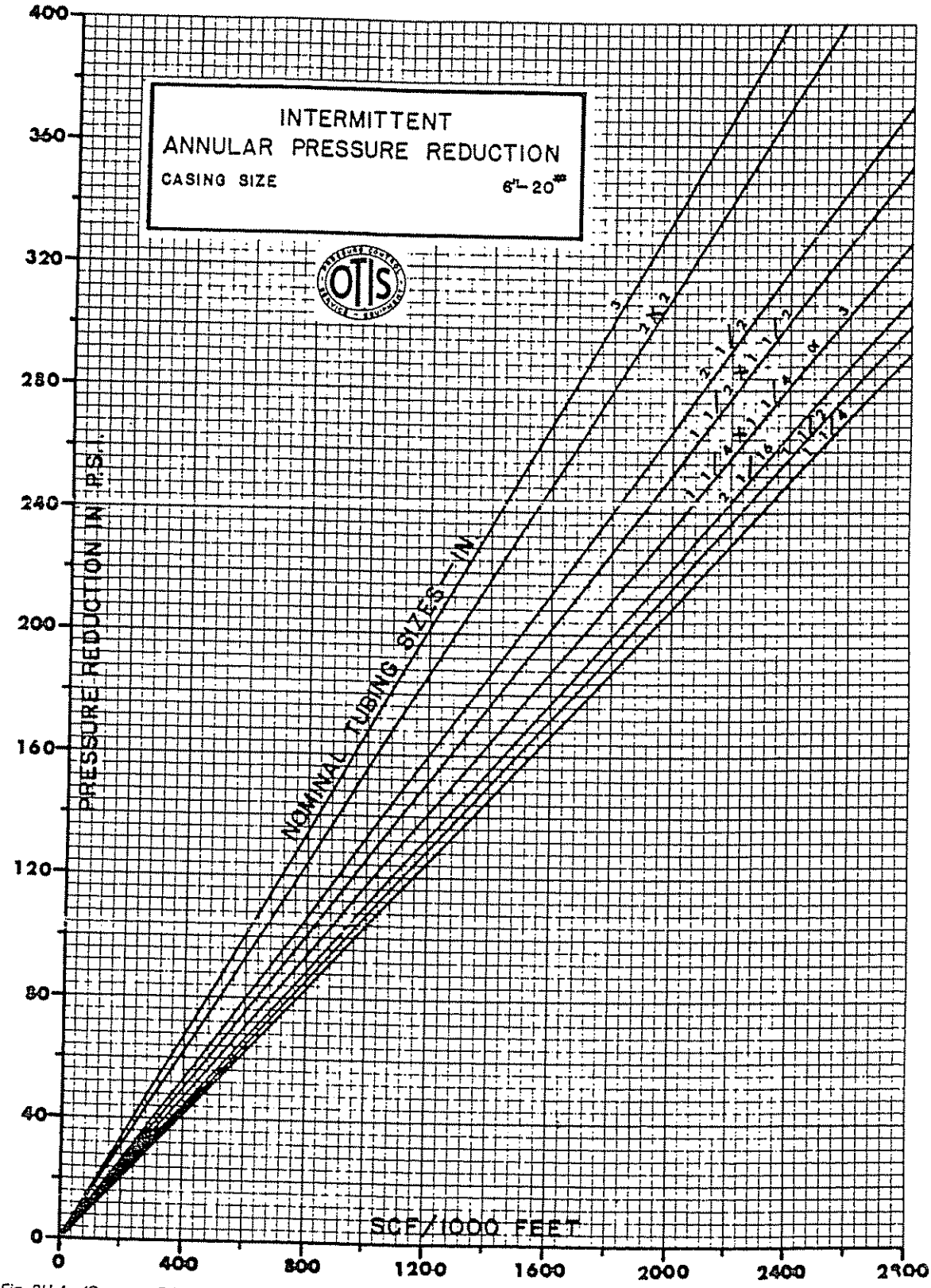


Fig. 3H-4 (Courtesy Otis Engineering Corp.)

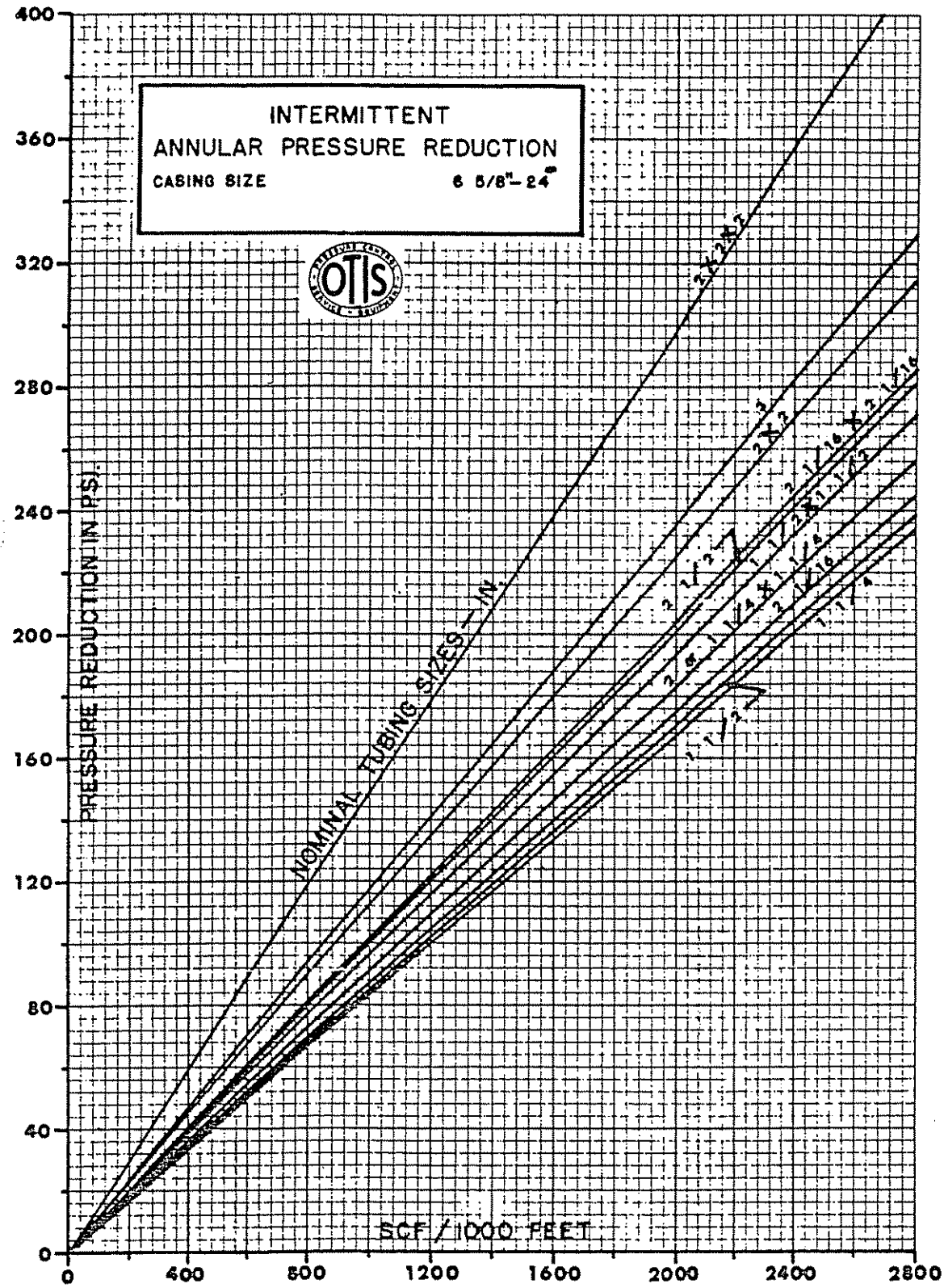


Fig. 3H-5 (Courtesy Otis Engineering Corp.)

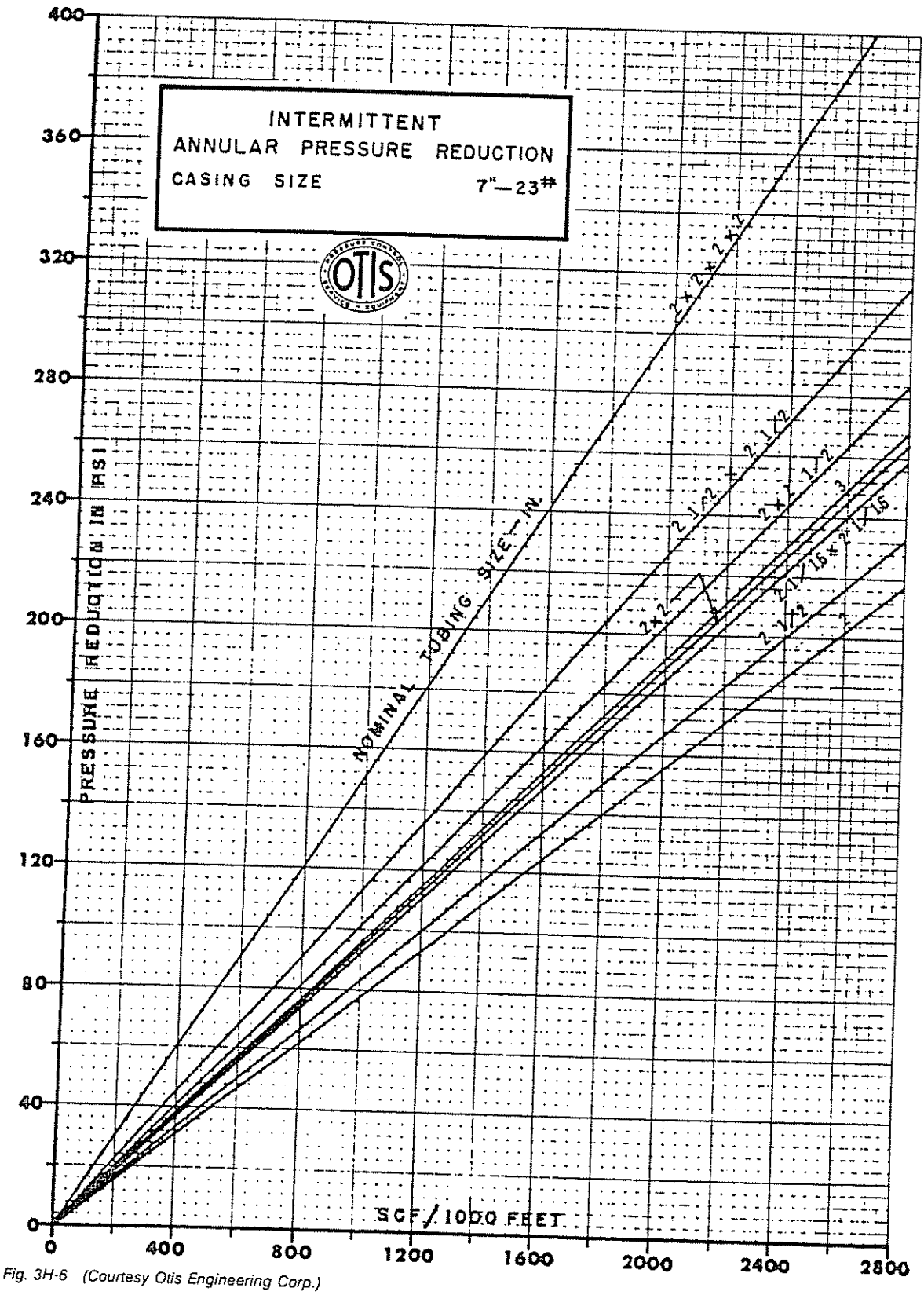


Fig. 3H-6 (Courtesy Otis Engineering Corp.)

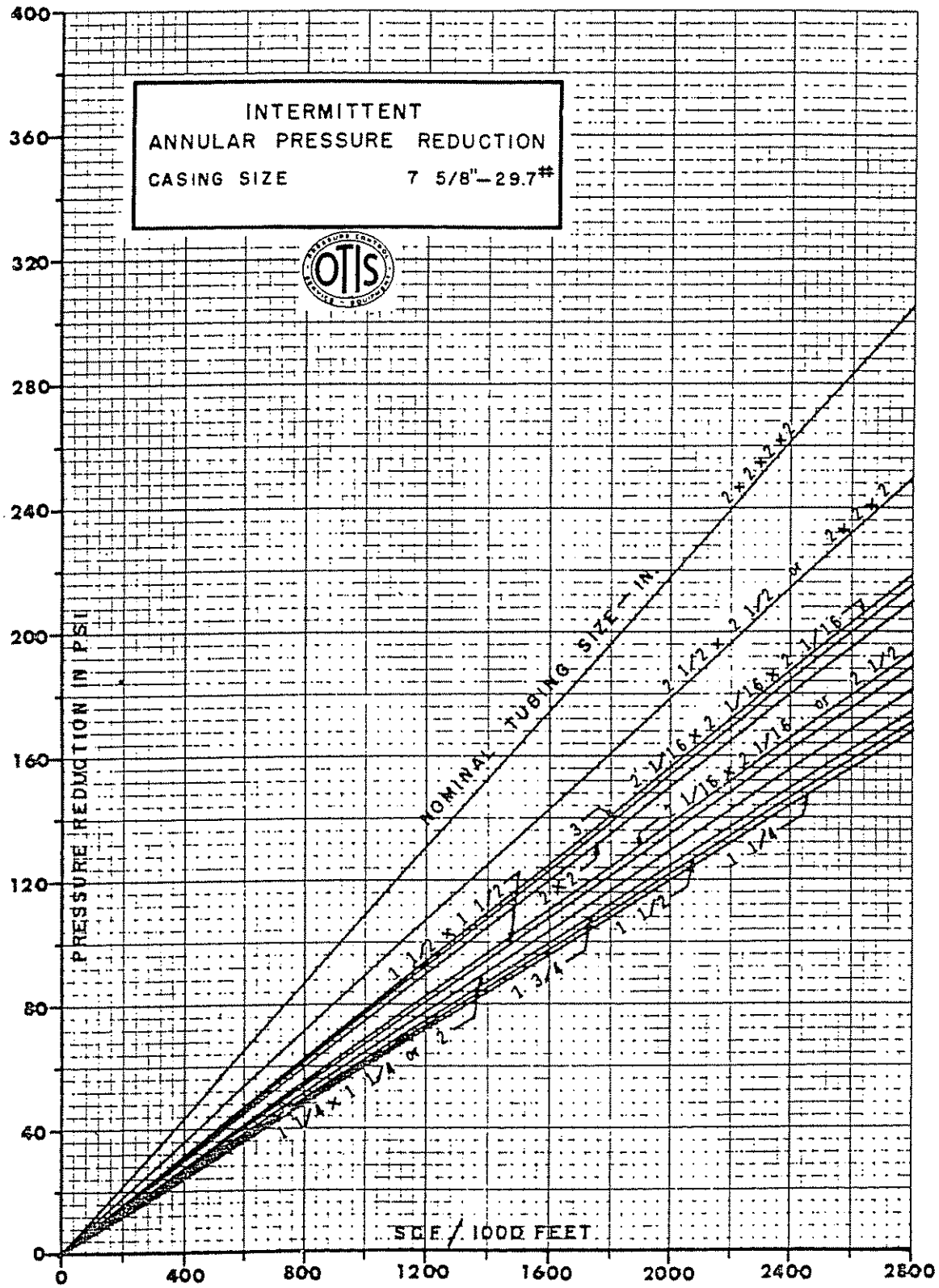


Fig. 3H-7 (Courtesy Otis Engineering Corp.)

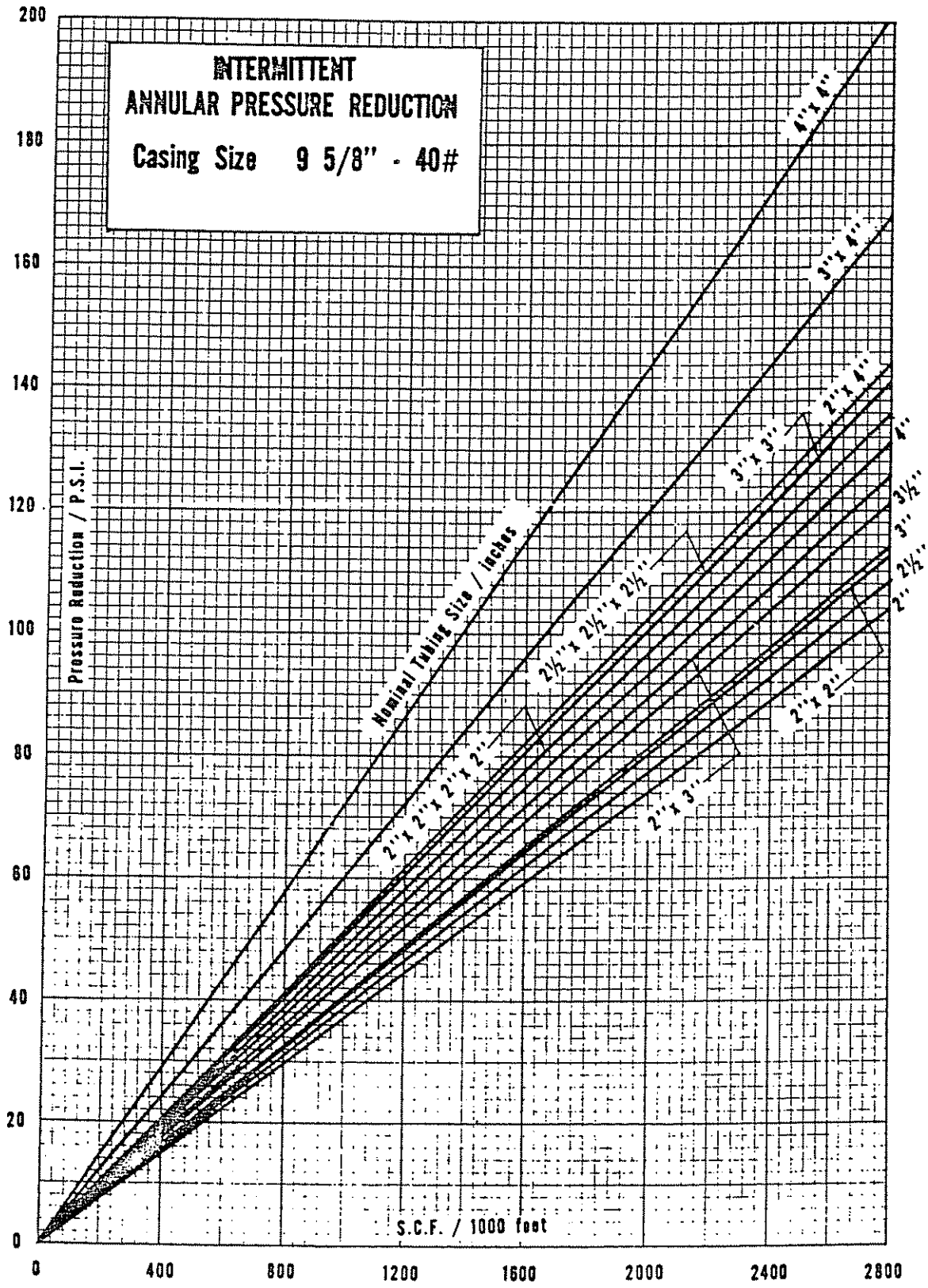


Fig. 3H-8 (Courtesy Otis Engineering Corp.)

EXPLANATION OF FIGURES 3I.1 and 3I.2 ENTITLED "R" VALUE SELECTION FOR INPUT CHOKE CONTROL — INTERMITTENT LIFT

These charts are prepared to assist in selection of the correct spread control port size for intermitting lift where the input gas is to be controlled by means of a choke rather than an intermitter.

The following procedure should be followed:

(1) Determine from Figure Series 3G the total gas volume required per cycle.
(2) Determine from Figure Series 3H the necessary pressure reduction to supply the needed gas in (1).
(3) Determine the difference between casing pressure and tubing pressure at the valve at the instant of opening (P_{vo} - P_t).
(4) Select the type valve size to be used (1" or 1½" O.D.) and from this select the correct Fig 3I.1 or 3I.2.
(5) On Fig 3I.1 or 3I.2 selected above, proceed horizontally to the right from the pressure reduction found in (2) until intersecting the correct P_{vo} - P_t line. Read the correct R Value on the horizontal scale at the bottom of the chart.
(6) Select a valve spread control port size nearest to this R value. Standard spread control port sizes are superimposed on the chart as vertical lines.

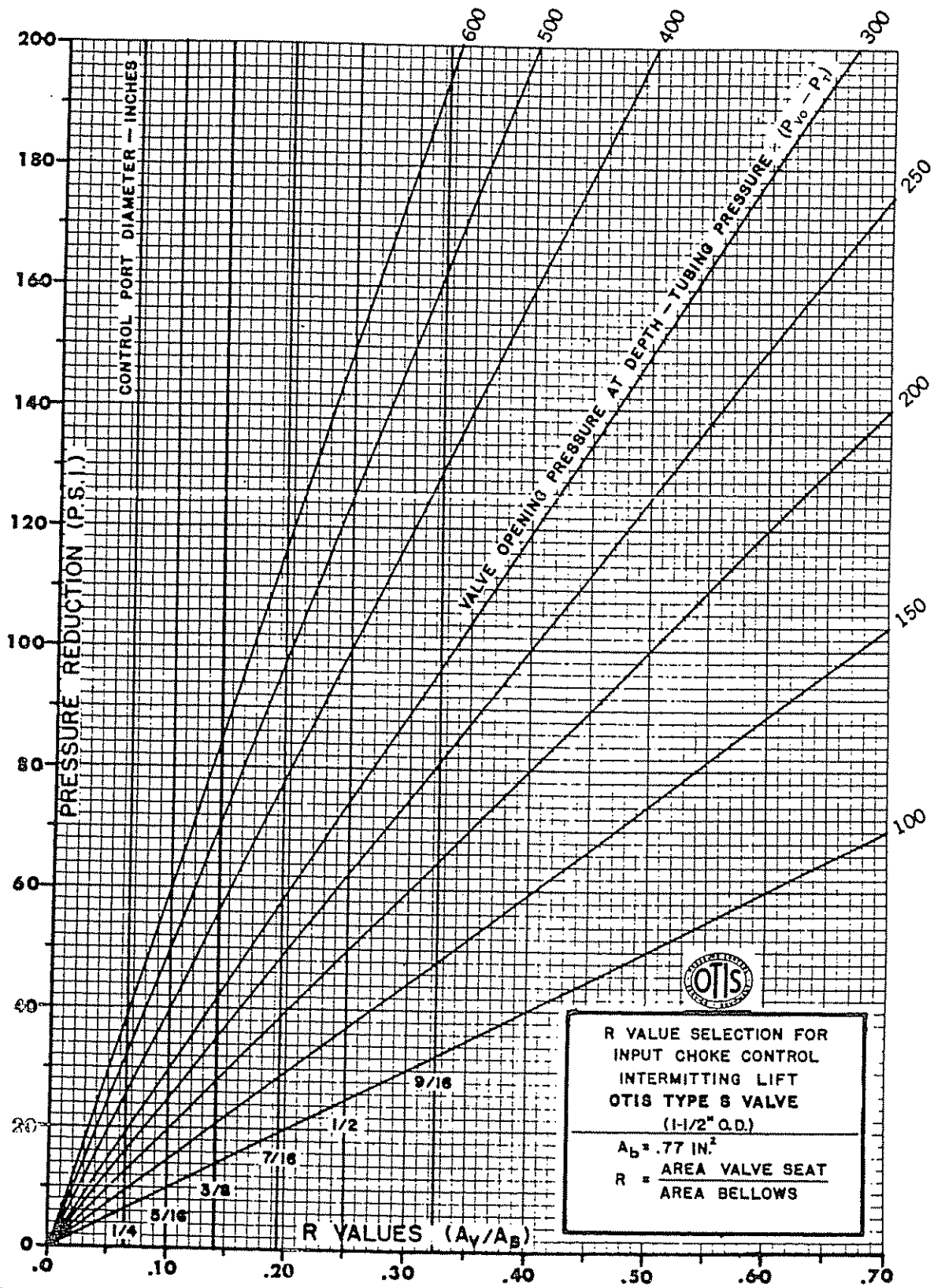


Fig. 3I-1 (Courtesy Otis Engineering Corp.)

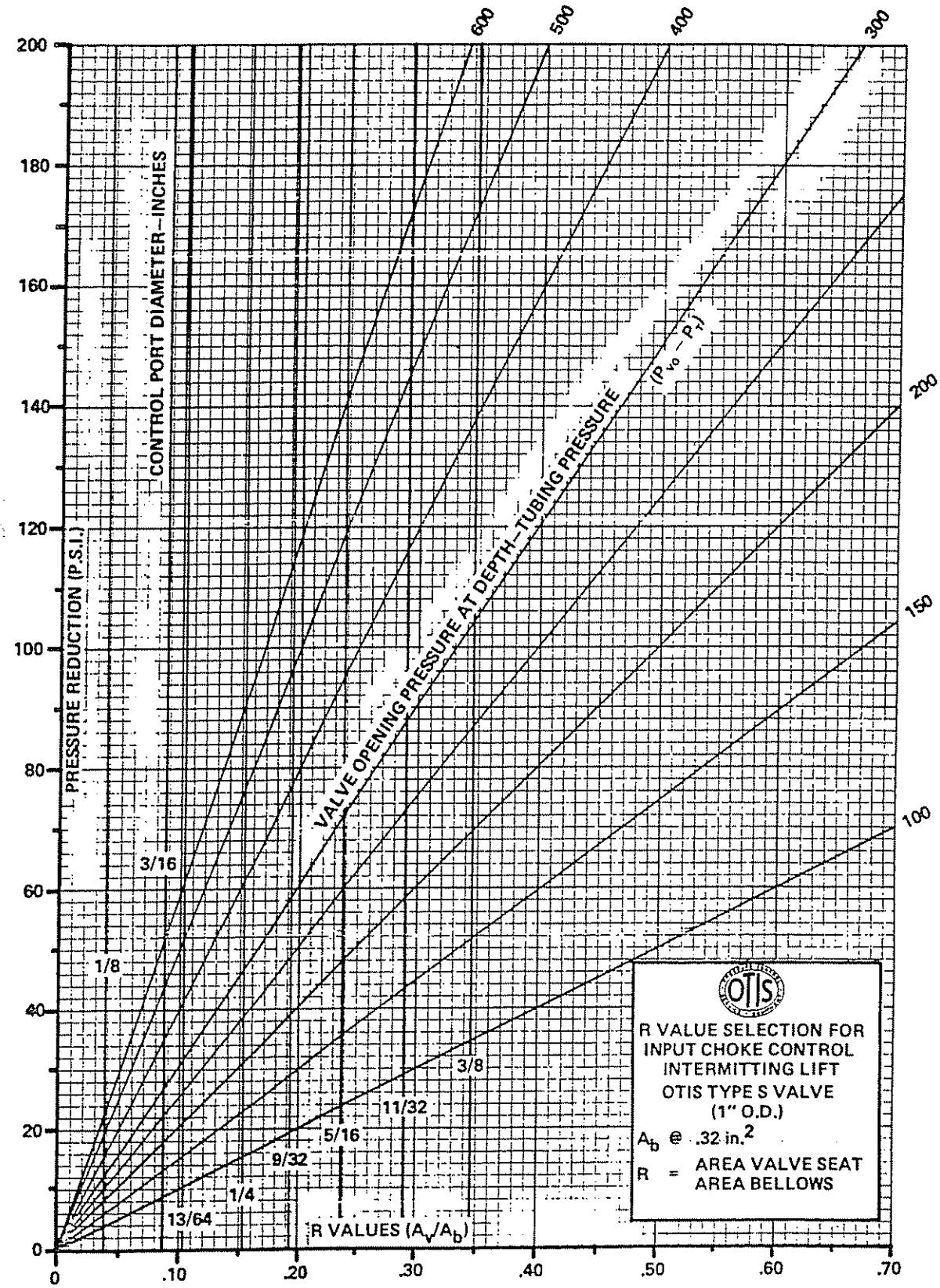


Fig. 3I-2 (Courtesy Otis Engineering Corp.)

DATA
A.P.I. TUBING

Nominal Size	Weight Per Ft.	DIAMETERS			Coupling O. D.	CROSS-SECTIONAL AREAS			CAPACITIES		
		External	Internal	Drift		External	Internal	Coupling O. D.	Lineal Feet Per Barrel	Barrels Per 100	Cubic Feet Per 100
A.P.I. TUBING — UPSET											
¾"	1.20	1.050	.824	.730	1.660	.866	.533	2.164	1510.00	.066	.37
1"	1.80	1.315	1.049	.955	1.900	1.358	.864	2.835	940.00	.107	.60
1¼"	2.40	1.660	1.380	1.286	2.200	2.16	1.50	3.80	540.54	.19	1.04
1½"	2.90	1.900	1.610	1.516	2.500	2.84	2.04	4.91	398.05	.25	1.41
2"	4.70	2.375	1.995	1.901	3.063	4.43	3.13	7.[illegible]	258.65	.39	2.17
2½"	6.50	2.875	2.441	2.347	3.668	6.49	4.68	10.57	172.76	.58	3.25
3"	9.30	3.500	2.992	2.867	4.500	9.62	7.03	15.90	114.99	.87	4.88
3½"	11.00	4.000	3.476	3.351	5.000	12.57	9.49	19.64	85.20	1.17	6.59
4"	12.75	4.500	3.958	3.833	5.563	15.90	12.30	24.31	65.71	1.52	8.54
A.P.I. TUBING — NON-UPSET											
¾"	1.14	1.050	.824	.730	1.313	.866	.533	1.354	1510.00	.066	.37
1"	1.70	1.315	1.049	.955	1.660	1.358	.864	2.164	940.00	.107	.60
1¼"	2.30	1.660	1.380	1.286	2.054	2.16	1.50	3.314	540.54	.19	1.04
1½"	2.75	1.900	1.610	1.516	2.200	2.84	2.04	3.80	398.05	.25	1.41
2"	4.60	2.375	1.995	1.901	2.875	4.43	3.13	6.49	258.65	.39	2.17
2½"	6.40	2.875	2.441	2.347	3.500	6.49	4.68	9.62	172.76	.58	3.25
3"	9.20	3.500	2.992	2.867	4.250	9.62	7.03	14.19	114.99	.87	4.88
3½"	9.50	4.000	3.548	3.423	4.750	12.57	9.89	17.72	81.78	1.22	6.87
4"	12.60	4.500	3.958	3.833	5.200	15.90	12.30	21.24	65.71	1.52	8.54
A.P.I. LINE PIPE											
¾"	1.14	1.050	.824	—	1.313	.866	.533	1.354	1510.00	.066	.37
1"	1.70	1.315	1.049	—	1.576	1.358	.864	1.951	940.00	.107	.60
1¼"	2.30	1.660	1.380	—	2.054	2.16	1.50	3.314	540.54	.19	1.04
1½"	2.75	1.900	1.610	—	2.200	2.84	2.04	3.80	398.05	.25	1.41

Fig. 3J-1 (Courtesy Otis Engineering Corp.)

ANNULUS - TUBING RATIOS

(UPSETS AND COUPLINGS DISREGARDED)

INNER STRING EU Tubing	OUTER STRING Tubing (Nominal Size)					Casing OD																		
	2"	2½"	3"	3½"	4"	5"		5½"		6"		6⅝"		7"		7"		7⅝"		8⅝"	
	Ratio	Ratio	Ratio	Ratio	Ratio	Wt.	Ratio	Wt.	Ratio	Wt.	Ratio	Wt.	Ratio	Wt.	Ratio	Wt.	Ratio	Wt.	Ratio	Wt.	Ratio
¾"	4.25	7.15	11.50	16.50	21.30	13	28.00	14	35.50	15	43.30	17	54.00	17	61.30	26	56.50	20	73.50	24	95.00
						15	27.00	17	34.00	18	41.80	24	50.00	20	60.00	29	55.00	24	72.30	28	93.20
						18	25.30	20	32.20	20	40.70	28	48.00	23	58.40	32	53.40	29.7	68.20	32	91.00
1"	2.00	3.80	6.50	9.65	12.60	13	16.80	14	21.20	15	26.20	17	32.50	17	37.20	26	34.50	20	44.60	24	58.00
						15	16.00	17	20.60	18	25.20	24	30.60	20	36.30	29	33.20	24	43.20	28	57.00
						18	15.20	20	19.20	20	24.50	28	29.00	23	35.40	32	32.30	29.7	41.50	32	55.50
1¼"		1.70	3.22	5.14	6.85	13	9.30	14	12.00	15	14.80	17	18.70	17	21.30	26	19.80	20	25.70	24	33.50
						15	8.90	17	11.40	18	14.15	24	17.20	20	20.80	29	19.00	24	25.00	28	33.00
						18	8.30	20	10.80	20	13.80	28	16.50	23	20.20	32	18.50	29.7	24.00	32	32.00
1½"			2.08	3.40	4.66	13	6.45	14	8.40	15	10.40	17	13.20	17	15.20	26	13.90	20	18.20	24	24.00
						15	6.15	17	7.95	18	10.00	24	12.20	20	14.80	29	13.40	24	17.80	28	23.60
						18	5.70	20	7.50	20	9.70	28	11.60	23	14.30	32	13.00	29.7	17.00	32	23.00
2"				1.68	2.50	13	3.68	14	4.92	15	6.26	17	8.10	17	9.40	26	8.50	20	11.40	24	15.00
						15	3.45	17	4.61	18	6.00	24	7.40	20	9.10	29	8.25	24	11.00	28	14.80
						18	3.20	20	4.36	20	5.80	28	6.90	23	8.80	32	8.00	29.7	10.40	32	14.40
2½"					1.23	13	2.00	14	2.81	15	3.75	17	4.95	17	5.80	26	5.25	20	7.15	24	9.60
						15	1.86	17	2.65	18	3.55	24	4.51	20	5.61	29	5.06	24	6.90	28	9.42
						18	1.68	20	2.45	20	3.43	28	4.25	23	5.42	32	4.88	29.7	6.60	32	9.15
3"								14	1.42	15	2.02	17	2.82	17	3.40	26	3.04	20	4.30	24	5.95
								17	1.30	18	1.90	24	2.55	20	3.30	29	2.92	24	4.15	28	5.80
								20	1.18	20	1.82	28	2.40	23	3.18	32	2.80	29.7	3.91	32	5.65
3½"										15	1.16	17	1.75	17	2.18	26	1.90	20	2.80	24	4.00
										18	1.08	24	1.54	20	2.08	29	1.80	24	2.70	28	3.90
										20	1.02	28	1.42	23	1.99	32	1.72	29.7	2.55	32	3.77
4"												17	1.12	17	1.44	26	1.24	20	1.96	24	2.90
												24	0.95	20	1.38	29	1.16	24	1.87	28	2.85
												28	0.85	23	1.30	32	1.09	29.7	1.74	32	2.73

Fig. 3J-2 (Courtesy Otis Engineering Corp.)

FLUID WEIGHT CONVERSION TABLE

Degrees A.P.I.	Specific Gravity	WEIGHT (Density)			FLUID HEAD	
		Gallon	Cubic Foot	Barrel	Pressure Per Foot	Height Per Pound
		Pounds			Lbs./Sq. In.	Feet
60.	.739	6.16	46.1	259.	.320	3.13
59.	.743	6.20	46.4	260.	.322	3.11
58.	.747	6.23	46.6	262.	.324	3.09
57.	.751	6.26	46.8	263.	.325	3.08
56	.755	6.30	47.1	265.	.327	3.06
55.	.759	6.33	47.4	266.	.329	3.04
54.	.763	6.36	47.6	267.	.330	3.03
53.	.767	6.40	47.9	269.	.332	3.01
52.	.771	6.43	48.1	270.	.334	2.99
51.	.775	6.46	48.3	271.	.336	2.98
50.	.780	6.51	48.7	273.	.338	2.96
49.	.784	6.54	48.9	275.	.340	2.94
48.	.788	6.57	49.2	276.	.341	2.93
47.	.793	6.61	49.5	278.	.343	2.92
46.	.797	6.65	49.8	279.	.345	2.90
45.	.802	6.69	50.0	281.	.348	2.87
44.	.806	6.72	50.3	282.	.349	2.87
43.	.811	6.76	50.6	284.	.351	2.85
42.	.816	6.81	50.9	286.	.354	2.82
41.	.820	6.84	51.2	287.	.355	2.82
40.	.825	6.88	51.5	289.	.357	2.80
39.	.830	6.92	51.8	291.	.359	2.79
38.	.835	6.96	52.1	292.	.362	2.76
37.	.840	7.01	52.4	294.	.364	2.75
36.	.845	7.05	52.7	296.	.366	2.73
35.	.850	7.09	53.0	298.	.368	2.72
34.	.855	7.13	53.3	299.	.370	2.70
33.	.860	7.17	53.6	301.	.372	2.69
32.	.865	7.21	53.9	303.	.375	2.67
31.	.871	7.26	54.3	305.	.377	2.65
30.	.876	7.31	54.7	307.	.380	2.63
29.	.882	7.36	55.1	309.	.382	2.62
28.	.887	7.40	55.4	311.	.384	2.60
27.	.893	7.45	55.7	313.	.387	2.58
26.	.898	7.49	56.0	315.	.389	2.57
25.	.904	7.54	56.4	317.	.392	2.55
24.	.910	7.59	56.8	319.	.394	2.54
23.	.916	7.64	57.2	321.	.397	2.52
22.	.922	7.69	57.5	323.	.399	2.51
21.	.928	7.74	57.9	325.	.402	2.49
20.	.934	7.79	58.3	327.	.405	2.47
19.	.940	7.84	58.7	329.	.407	2.46
18.	.946	7.89	59.0	331.	.410	2.44
17.	.953	7.95	59.5	334.	.413	2.42
16.	.959	8.00	59.8	336.	.416	2.40
15.	.966	8.06	60.3	339.	.419	2.39
14.	.973	8.11	60.7	341.	.421	2.38
13.	.979	8.16	61.0	343.	.424	2.36
12.	.986	8.22	61.5	345.	.427	2.34
11.	.993	8.28	61.9	348	.430	2.33
10° API or Pure Water.	1.00	8.34	62.4	350.	.433	2.31
	1.01	8.4	62.8	353	.436	2.29
	1.03	8.6	64.3	361.	.447	2.24
	1.06	8.8	65.8	370.	.457	2.19
	1.08	9.0	67.3	378	.468	2.14
Salt Water.....	1.10	9.2	68.8	386.	.478	2.09
	1.13	9.4	70.3	395.	.488	2.05
	1.15	9.6	71.8	403.	.499	2.00
	1.154	9.625	72.0	404.	.500	2.00
	1.18	9.8	73.3	412	.509	1.96

Zaba & Doherty's Practical Petroleum Engineer's Handbook

Fig. 3J-3 (Courtesy Otis Engineering Corp.)

CONVERSION FACTORS

Applying to Oil Country Calculations

Acre	=	43,560.	square feet
Acre	=	4,047.	square meters
Acre	=	160.	square rods
Acre	=	0.405 6	square varas (Texas)
Acre	=	.4047	hectares
Acre foot	=	7,758.	barrels
Atmospheres	=	33.96	feet of water
Atmospheres	=	29.92	inches of mercury
Atmospheres	=	760 0	millimeters of mercury
Atmospheres	=	14.70	pounds per square inch
Barrel	=	5.6146	cubic feet
Barrel	=	42.0	gallons
Barrel of water @ 60°F.	=	.1605	metric ton
Barrel (Oil A.P.I.)	=	.1342	metric ton
Barrel per hour	=	.0936	cubic feet per minute
Barrel per hour	=	.700	gallons per minute
Barrel per hour	=	3.968	cubic inches per second
Barrel per day	=	.02917	gallons per minute
British Thermal Unit	=	.2520	kilogram calorie
British Thermal Unit	=	.2931	watt hour
B T U per minute	=	.01334	horse power
Centimeter	=	.3937	inch
Centimeter of mercury	=	.1934	pound per square inch
Chain	=	66.00	feet
Chain	=	4.00	rods
Cubic centimeter	=	.061 023	cubic inch
Cubic foot	=	.1781	barrel
Cubic foot	=	7 4804	gallons (U.S.)
Cubic foot	=	.02523	cubic meter
Cubic foot	=	3081	cubic centimeter (m/l)
Cubic foot per minute	=	10 686	barrels per hour
Cubic foot per minute	=	59 820	cubic inches per second
Cubic foot per minute	=	7.481	gallons per minute
Cubic inch	=	16.387	cubic centimeters
Cubic meter	=	6.2897	barrels
Cubic meter	=	35 314	cubic feet
Cubic meter	=	1.308	cubic yards
Cubic yard	=	4.8089	barrels
Cubic yard	=	46,656	cubic inches
Cubic yard	=	7646	cubic meter
Fathom	=	6 00	feet
Fathom	=	.3048	meter
Fathom	=	.0600	rods (Texas)
Foot of water @ 60°F.	=	.4331	pound per square inch
Foot per second	=	.6818	mile per hour
Foot pound	=	.001 2856	British Thermal Unit
Foot pound per second	=	.001 818	horse power
Gallon (U.S.)	=	.02381	barrel
Gallon (U.S.)	=	.1337	cubic foot
Gallon (U.S.)	=	231.000	cubic inches
Gallon (U.S.)	=	3.785	liters
Gallon (U.S.)	=	.8327	gallon (Imperial)
Gallon (Imperial)	=	1.2009	gallons (U.S.)
Gallon (Imperial)	=	277 274	cubic inches
Gallon per minute	=	1 429	barrels per hour
Gallon per minute	=	.1337	cubic feet per minute
Gallon per minute	=	34 286	barrels per day
Grain (Avoirdupois)	=	.04480	grams
Grain per gallon	=	17 118	parts per million
Grain per gallon	=	142 86	pounds per million gals
Grain per gallon	=	.01714	grain per liter
Gram	=	15 432	grains
Gram	=	.035237	ounces
Gram per liter	=	58 418	grains per gallon
Hectare	=	2.471	acres
Hectare	=	.010	square kilometer
Horse-power	=	42 44	B T U's per minute
Horse-power	=	33,000	foot-pounds per minute
Horse-power	=	550.	foot-pounds per second
Horse-power	=	1 014	horse-power (metric)
Horse-power	=	7457	kilowatts
Horse-power hour	=	2,547	British Thermal Units
Inch	=	2.540	centimeters
Inch of mercury	=	1 134	feet of water
Inch of mercury	=	.4912	pound per square inch
Inch of water @ 60°F	=	.0361	pound per square inch
Kilogram	=	2.2046	pounds
Kilogram Calorie	=	3 968	British Thermal Units
Kilogram per square cm	=	14.223	pounds per square inch
Kilometer	=	3,281	feet
Kilometer	=	.6214	mile
Kilowatt	=	1 341	horse-power
Link (Surveyor's)	=	7.92	inches
Liter	=	.2642	gallon
Liter	=	1 0567	quarts
Meter	=	3 281	feet
Meter	=	39.37	inches
Mile	=	5,280	feet
Mile	=	1 609	kilometers
Mile	=	1,000 8	varas (Texas)
Mile per hour	=	1 4667	feet per second
Ounce (Avoirdupois)	=	437 5	grains
Ounce (Avoirdupois)	=	28.3495	grams
Part per million	=	.05533	grain per gallon
Part per million	=	8.345	pounds per million gals
Pound (Avoirdupois)	=	16 000	pounds
Pound	=	7,000	grains
Pound	=	0.454	kilograms
Pound per square inch	=	2.309	feet of water @ 60°F
Pound per square inch	=	2 0353	inches of mercury
Pound per square inch	=	51 697	millimeters of mercury
Pound per square inch	=	0703	kilograms per sq. cent
Pound per million gals	=	.00700	grain per gallon
Pound per million gals	=	.11983	parts per million
Quart (Liquid)	=	946	liter
Quarter (Maximum)	=	501.047	pounds
Rod	=	16 5	feet
Rod	=	25 0	links
Rock measure (m/l)	=	1 1	cubic foot
Square centimeter	=	.1550	square inch
Square foot	=	.0929	square meter
Square foot	=	1296	square varas (Texas)
Square inch	=	6.453	square centimeters
Square kilometer	=	.3861	square mile
Square meter	=	10 76	square feet
Square mile	=	2.590	square kilometers
Square vara (Texas)	=	7.716	square feet
Temp Centigrade	=	5/9 (Temp Fahr —32)	.
Temp Fahrenheit	=	9/5 Temp Cent + 32	
Temp Absolute C.	=	Temp °C + 273	
Temp Absolute F.	=	Temp °F + 460	
Ton (Long)	=	2,240	pounds
Ton (Metric)	=	2 205	pounds
Ton (Short or Net)	=	2,000	pounds
Ton (Metric)	=	1 102	tons (short or net)
Ton (Metric)	=	1,000	kilograms
Ton (Metric)	=	6.297	barrels of water @ 60°F
Ton (Metric)	=	7 454	barrels Oil A.P.I.
Ton (Short or Net)	=	.907	ton (metric)
Vara (Texas)	=	2 7778	feet
Vara (Texas)	=	33 3333	inches
Watt-hour	=	3 413	British Thermal Units
Yard	=	9144	meter

Fig. 3J-4 (Courtesy Otis Engineering Corp.)

DECIMAL EQUIVALENTS OF FRACTIONS OF AN INCH. (ADVANCING BY 64THS.)

1/64 = .015625	17/64 = .265625	33/64 = .515625	49/64 = .765625
1/32 = .03125	9/32 = .28125	17/32 = .53125	25/32 = .78125
3/64 = .046875	19/64 = .296875	35/64 = .546875	51/64 = .796875
1/16 = .0625	5/16 = .3125	9/16 = .5625	13/16 = .8125
5/64 = .078125	21/64 = .328125	37/64 = .578125	53/64 = .828125
3/32 = .09375	11/32 = .34375	19/32 = .59375	27/32 = .84375
7/64 = .109375	23/64 = .359375	39/64 = .609375	55/64 = .859375
1/8 = .125	3/8 = .375	5/8 = .625	7/8 = .875
9/64 = .140625	25/64 = .390625	41/64 = .640625	57/64 = .890625
5/32 = .15625	13/32 = .40625	21/32 = .65625	29/32 = .90625
11/64 = .171875	27/64 = .421875	43/64 = .671875	59/64 = .921875
3/16 = .1875	7/16 = .4375	11/16 = .6875	15/16 = .9375
13/64 = .203125	29/64 = .453125	45/64 = .703125	61/64 = .953125
7/32 = .21875	15/32 = .46875	23/32 = .71875	31/32 = .96875
15/64 = .234375	31/64 = .484375	47/64 = .734375	63/64 = .984375
1/4 = .25	1/2 = .50	3/4 = .75	

CASING DIMENSIONS

SIZE O. D INCHES	WEIGHT PER FOOT lbs	INSIDE DIAMETER INCHES	DRIFT DIAMETER INCHES
4-1/2	9.50	4.090	3.965
	11.60	4.000	3.875
	13.50	3.920	3.795
	15.10	3.826	3.701
5	11.50	4.560	4.435
	13.00	4.494	4.369
	15.00	4.408	4.283
	18.00	4.276	4.151
5-1/2	13.00	5.044	4.919
	14.00	5.012	4.887
	15.50	4.950	4.825
	17.00	4.892	4.767
	20.00	4.778	4.653
	23.00	4.670	4.545
6	15.00	5.524	5.399
	18.00	5.424	5.299
	20.00	5.352	5.227
	23.00	5.240	5.115
	26.00	5.132	5.007
6-5/8	17.00	6.135	6.010
	20.00	6.049	5.924
	24.00	5.921	5.796
	28.00	5.791	5.666
	32.00	5.675	5.550
7	17.00	6.538	6.413
	20.00	6.456	6.331
7 (cont)	23.00	6.366	6.241
	26.00	6.276	6.151
	29.00	6.184	6.059
	32.00	6.094	5.969
	35.00	6.004	5.879
	38.00	5.920	5.795
7-5/8	20.00	7.125	7.000
	24.00	7.025	6.900
	26.40	6.969	6.844
	29.70	6.875	6.750
	33.70	6.765	6.640
	39.00	6.625	6.500
8-5/8	24.00	8.097	7.972
	28.00	8.017	7.892
	32.00	7.921	7.796
	36.00	7.825	7.700
	40.00	7.725	7.600
	44.00	7.625	7.500
	49.00	7.511	7.386
9-5/8	29.30	9.063	8.907
	32.30	9.001	8.845
	36.00	8.921	8.765
	40.00	8.835	8.679
	43.50	8.755	8.599
	47.00	8.681	8.525
	53.50	8.535	8.379

Fig. 3J-5 (Courtesy Otis Engineering Corp.)

Areas
of Circles
Area = .7853981634 × D²

Diam.	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	Diam.
0	.....	.7854	3.1416	7.0686	12.566	19.635	28.274	38.485	50.266	63.617	78.540	95.033	113.10	132.73	153.94	176.71	201.06	226.98	0
1/64	.000192	.8101	3.1909	7.1424	12.665	19.753	28.422	38.656	50.462	63.838	78.785	95.303	113.39	133.05	154.26	177.08	201.45	227.40	1/64
1/32	.000767	.8352	3.2405	7.2166	12.763	19.881	28.570	38.829	50.659	64.060	79.031	95.574	113.69	133.37	154.63	177.45	201.85	227.82	1/32
3/64	.001726	.8607	3.2906	7.2912	12.863	20.005	28.718	39.002	50.856	64.282	79.278	95.845	113.98	133.69	154.97	177.82	202.24	228.23	3/64
1/16	.003068	.8866	3.3410	7.3662	12.962	20.129	28.866	39.175	51.054	64.504	79.525	96.116	114.23	134.01	155.32	178.19	202.64	228.65	1/16
5/64	.004794	.9129	3.3918	7.4415	13.062	20.253	29.015	39.348	51.252	64.727	79.772	96.388	114.57	134.33	155.66	178.56	203.03	229.07	5/64
3/32	.006903	.9306	3.4430	7.5173	13.162	20.378	29.165	39.522	51.450	64.950	80.019	96.660	114.87	134.65	156.01	178.93	203.43	229.49	3/32
7/64	.009396	.9666	3.4946	7.5934	13.263	20.503	29.315	39.696	51.649	65.173	80.267	96.932	115.17	134.98	156.35	179.30	203.82	229.91	7/64
1/8	.01227	.9940	3.5466	7.6699	13.364	20.629	29.465	39.871	51.849	65.397	80.516	97.205	115.47	135.30	156.70	179.67	204.22	230.33	1/8
9/64	.01553	1.0218	3.5989	7.7468	13.465	20.755	29.615	40.046	52.048	65.621	80.764	97.479	115.76	135.62	157.05	180.04	204.61	230.75	9/64
5/32	.01917	1.0500	3.6516	7.8241	13.567	20.881	29.766	40.222	52.248	65.845	81.013	97.752	116.06	135.94	157.39	180.42	205.01	231.17	5/32
11/64	.02320	1.0786	3.7048	7.9017	13.669	21.008	29.917	40.398	52.448	66.070	81.263	98.026	116.30	136.27	157.74	180.79	205.40	231.59	11/64
3/16	.02761	1.1075	3.7583	7.9798	13.772	21.135	30.069	40.574	52.649	66.296	81.513	98.301	116.66	136.59	158.09	181.16	205.80	232.01	3/16
13/64	.03240	1.1369	3.8121	8.0582	13.875	21.263	30.221	40.750	52.850	66.521	81.763	98.575	116.96	136.91	158.44	181.53	206.20	232.44	13/64
7/32	.03758	1.1666	3.8664	8.1370	13.978	21.391	30.374	40.927	53.052	66.747	82.014	98.850	117.26	137.24	158.79	181.91	206.60	232.86	7/32
15/64	.04314	1.1967	3.9211	8.2162	14.082	21.519	30.526	41.105	53.254	66.974	82.265	99.126	117.56	137.56	159.14	182.28	206.99	233.28	15/64
1/4	.04909	1.2272	3.9761	8.2958	14.186	21.648	30.680	41.282	53.456	67.201	82.516	99.402	117.86	137.89	159.48	182.65	207.39	233.71	1/4
17/64	.05541	1.2577	4.0315	8.3757	14.291	21.777	30.833	41.461	53.659	67.428	82.768	99.678	118.16	138.21	159.83	183.03	207.79	234.13	17/64
9/32	.06213	1.2893	4.0873	8.4561	14.396	21.906	30.987	41.639	53.862	67.655	83.020	99.955	118.46	138.54	160.19	183.40	208.19	234.55	9/32
19/64	.06922	1.3209	4.1435	8.5368	14.501	22.036	31.141	41.818	54.065	67.883	83.272	100.232	118.76	138.86	160.54	183.78	208.59	234.98	19/64
5/16	.07670	1.3530	4.2000	8.6179	14.607	22.166	31.296	41.997	54.269	68.112	83.525	100.509	119.06	139.19	160.89	184.15	208.99	235.40	5/16
21/64	.08456	1.3854	4.2570	8.6994	14.713	22.297	31.451	42.177	54.473	68.341	83.779	100.787	119.37	139.52	161.24	184.53	209.39	235.83	21/64
11/32	.09281	1.4182	4.3143	8.7813	14.819	22.428	31.607	42.357	54.678	68.570	84.032	101.066	119.67	139.84	161.59	184.91	209.79	236.25	11/32
23/64	.1014	1.4513	4.3720	8.8636	14.926	22.559	31.763	42.537	54.883	68.799	84.286	101.344	119.97	140.17	161.94	185.28	210.20	236.68	23/64
3/8	.1104	1.4849	4.4301	8.9462	15.033	22.691	31.919	42.718	55.088	69.029	84.541	101.623	120.28	140.50	162.30	185.66	210.60	237.10	3/8
25/64	.1198	1.5188	4.4886	9.0292	15.141	22.823	32.076	42.899	55.294	69.259	84.796	101.903	120.58	140.83	162.65	186.04	211.00	237.53	25/64
13/32	.1296	1.5532	4.5475	9.1126	15.249	22.955	32.233	43.081	55.500	69.490	85.051	102.182	120.88	141.16	163.00	186.42	211.40	237.96	13/32
27/64	.1398	1.5879	4.6067	9.1964	15.357	23.088	32.390	43.263	55.707	69.721	85.306	102.462	121.19	141.49	163.36	186.79	211.80	238.39	27/64
7/16	.1503	1.6230	4.6664	9.2806	15.466	23.221	32.548	43.445	55.914	69.953	85.562	102.743	121.49	141.82	163.71	187.17	212.21	238.81	7/16
29/64	.1613	1.6584	4.7264	9.3652	15.575	23.355	32.706	43.628	56.121	70.184	85.819	103.024	121.80	142.15	164.06	187.55	212.61	239.24	29/64
15/32	.1726	1.6943	4.7868	9.4501	15.684	23.489	32.865	43.811	56.329	70.417	86.075	103.305	122.11	142.48	164.42	187.93	213.02	239.67	15/32
31/64	.1843	1.7305	4.8476	9.5354	15.794	23.623	33.024	43.995	56.537	70.649	86.333	103.587	122.43	142.81	164.77	188.31	213.42	240.10	31/64
1/2	.1963	1.7671	4.9088	9.6212	15.904	23.758	33.183	44.179	56.745	70.882	86.590	103.869	122.72	143.14	165.13	188.69	213.82	240.53	1/2
33/64	.2088	1.8042	4.9703	9.7072	16.015	23.893	33.343	44.363	56.954	71.116	86.848	104.151	123.03	143.47	165.49	189.07	214.23	240.96	33/64
17/32	.2217	1.8415	5.0322	9.7937	16.126	24.029	33.503	44.548	57.163	71.349	87.106	104.434	123.33	143.80	165.84	189.45	214.64	241.39	17/32
35/64	.2349	1.8795	5.0946	9.8806	16.237	24.165	33.663	44.733	57.373	71.583	87.365	104.717	123.64	144.13	166.20	189.83	215.04	241.82	35/64
9/16	.2485	1.9175	5.1573	9.9678	16.349	24.301	33.824	44.918	57.583	71.818	87.624	105.001	123.95	144.47	166.56	190.22	215.45	242.25	9/16
37/64	.2625	1.9560	5.2203	10.0554	16.461	24.438	33.985	45.104	57.793	72.053	87.883	105.285	124.26	144.80	166.91	190.60	215.85	242.68	37/64
19/32	.2769	1.9949	5.2838	10.1435	16.574	24.575	34.147	45.290	58.004	72.288	88.143	105.569	124.57	145.13	167.27	190.98	216.26	243.11	19/32
39/64	.2916	2.0342	5.3477	10.2318	16.687	24.713	34.309	45.477	58.215	72.524	88.404	105.804	124.88	145.47	167.63	191.36	216.67	243.54	39/64
5/8	.3068	2.0739	5.4119	10.3206	16.800	24.850	34.472	45.664	58.426	72.760	88.664	106.139	125.19	145.80	167.99	191.75	217.08	243.98	5/8
41/64	.3223	2.1140	5.4765	10.4098	16.914	24.989	34.634	45.851	58.638	72.996	88.925	106.425	125.50	146.14	168.35	192.13	217.48	244.41	41/64
21/32	.3382	2.1545	5.5415	10.4994	17.028	25.127	34.798	46.039	58.850	73.233	89.186	106.711	125.81	146.47	168.71	192.52	217.89	244.84	21/32
43/64	.3545	2.1953	5.6069	10.5893	17.142	25.266	34.961	46.227	59.063	73.470	89.448	106.997	126.12	146.81	169.07	192.90	218.30	245.28	43/64
11/16	.3712	2.2365	5.6727	10.6796	17.257	25.406	35.125	46.415	59.276	73.708	89.710	107.284	126.43	147.14	169.43	193.28	218.71	245.71	11/16
45/64	.3883	2.2782	5.7388	10.7703	17.372	25.546	35.289	46.604	59.489	73.946	89.973	107.571	126.74	147.48	169.79	193.67	219.12	246.14	45/64
23/32	.4057	2.3201	5.8054	10.8614	17.488	25.686	35.454	46.793	59.703	74.184	90.236	107.858	127.05	147.82	170.15	194.06	219.53	246.58	23/32
47/64	.4236	2.3623	5.8723	10.9528	17.604	25.826	35.619	46.983	59.917	74.423	90.499	108.146	127.36	148.15	170.51	194.44	219.94	247.01	47/64
3/4	.4418	2.4053	5.9396	11.0447	17.721	25.967	35.785	47.173	60.132	74.662	90.763	108.434	127.68	148.49	170.87	194.83	220.35	247.45	3/4
49/64	.4604	2.4484	6.0073	11.1369	17.837	26.108	35.951	47.363	60.347	74.901	91.027	108.723	127.99	148.83	171.24	195.21	220.76	247.89	49/64
25/32	.4794	2.4929	6.0753	11.2295	17.954	26.250	36.117	47.554	60.562	75.141	91.291	109.012	128.30	149.17	171.60	195.60	221.18	248.32	25/32
51/64	.4987	2.5359	6.1438	11.3236	18.072	26.392	36.283	47.745	60.778	75.382	91.556	109.301	128.62	149.50	171.96	195.99	221.59	248.76	51/64
13/16	.5185	2.5802	6.2126	11.4159	18.190	26.535	36.450	47.937	60.994	75.622	91.821	109.591	128.93	149.84	172.32	196.38	222.00	249.20	13/16
53/64	.5386	2.6248	6.2819	11.5096	18.308	26.678	36.618	48.129	61.211	75.863	92.087	109.881	129.25	150.18	172.69	196.77	222.41	249.63	53/64
27/32	.5591	2.6699	6.3515	11.6038	18.427	26.821	36.787	48.321	61.427	76.105	92.353	110.171	129.56	150.52	173.05	197.15	222.83	250.07	27/32
55/64	.5800	2.7153	6.4215	11.6983	18.546	26.964	36.954	48.514	61.645	76.346	92.619	110.462	129.88	150.86	173.42	197.54	223.24	250.51	55/64
7/8	.6013	2.7612	6.4918	11.7933	18.665	27.109	37.122	48.707	61.862	76.589	92.886	110.753	130.19	151.20	173.78	197.93	223.65	250.95	7/8
57/64	.6230	2.8074	6.5626	11.8885	18.785	27.252	37.291	48.900	62.080	76.831	93.153	111.045	130.51	151.54	174.15	198.32	224.07	251.39	57/64
29/32	.6450	2.8540	6.6337	11.9842	18.906	27.398	37.461	49.094	62.299	77.074	93.420	111.337	130.82	151.88	174.51	198.71	224.48	251.83	29/32
59/64	.6675	2.9010	6.7052	12.0803	19.026	27.543	37.630	49.288	62.518	77.317	93.688	111.630	131.14	152.22	174.88	199.10	224.90	252.26	59/64
15/16	.6903	2.9483	6.7771	12.1768	19.147	27.688	37.800	49.483	62.737	77.561	93.956	111.922	131.46	152.57	175.25	199.49	225.31	252.70	15/16
61/64	.7135	2.9961	6.8494	12.2736	19.268	27.834	37.971	49.678	62.956	77.805	94.225	112.215	131.78	152.91	175.61	199.89	225.73	253.15	61/64
31/32	.7371	3.0442	6.9221	12.3708	19.390	27.981	38.142	49.874	63.176	78.050	94.494	112.509	132.09	153.25	175.98	200.28	226.15	253.59	31/32
63/64	.7610	3.0927	6.9952	12.4684	19.512	28.127	38.313	50.069	63.396	78.295	94.763	112.803	132.41	153.59	176.35	200.67	226.56	254.03	63/64

Fig. 3J-6 (Courtesy Otis Engineering Corp.)

API TUBING, API COUPLING, AND API INTEGRAL JOINT DATA



TUBING TABLES

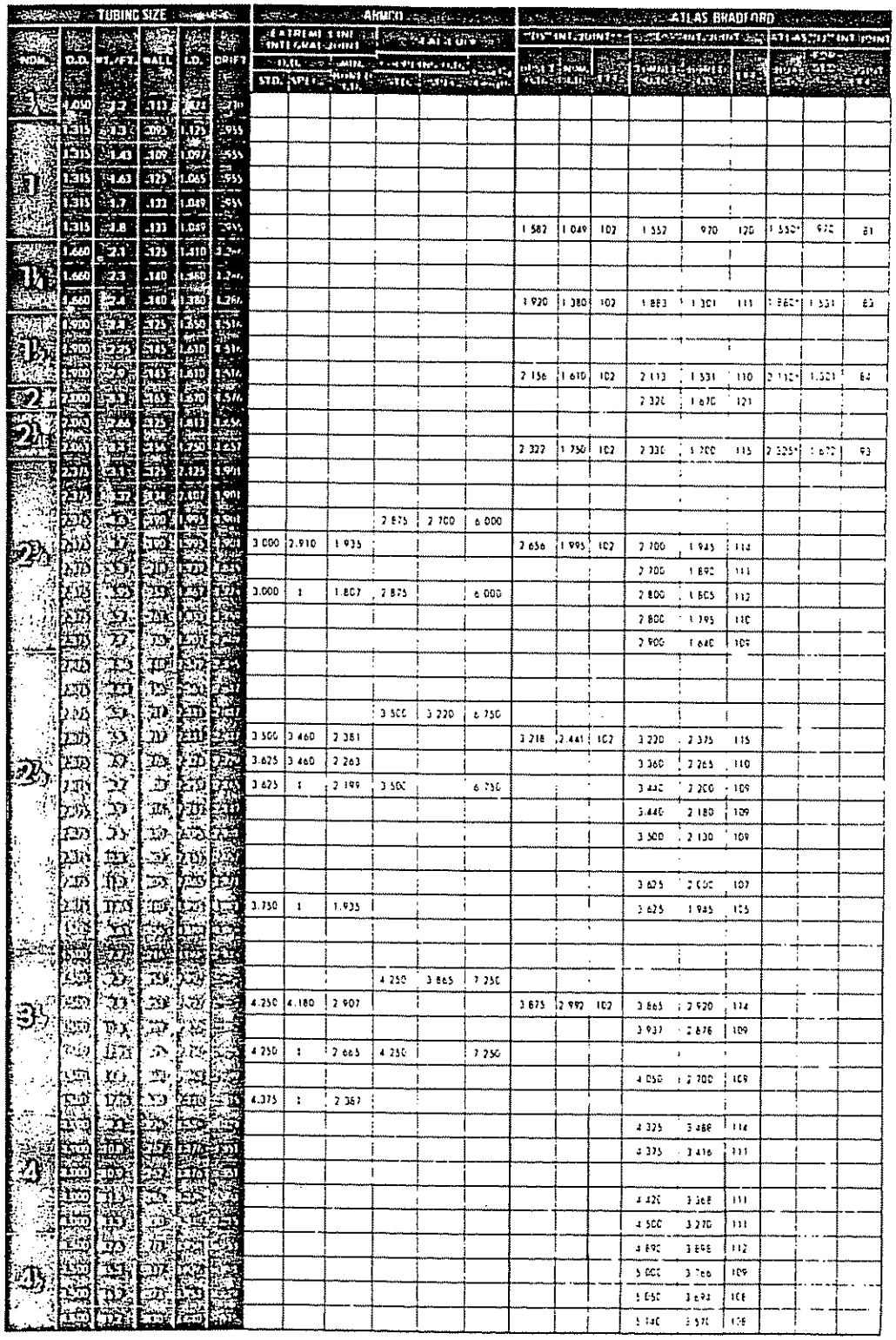
Information Revised to Include Performance Properties Adopted for Inclusion in API Bul. 5C2 (8th Edition), March, 1964

TUBING SIZE		NOMINAL WEIGHT			GRADE	WALL THICKNESS	INSIDE DIAM.	THREADED AND COUPLED			INTEGRAL JOINT		COLLAPSE RESISTANCE	INTERNAL YIELD PRESSURE	JOINT YIELD STRENGTH			FREE POINT CONSTANT
		T&C NON-UP	T&C UPSET	INT. JT.				DRIFT DIAM.	COUP. OUTSIDE DIAM.	UPSET OUTSIDE DIAM.	DRIFT DIAM.	BOX OD			T&C NON-UP	T&C UPSET	INT. JT.	
		lb/ft	lb/ft	lb/ft		in.	in.	in.	in.	in.	in.	in.	psi	psi	lb	lb	lb	
1.050	1.14	1.20		H-40	.113	.824	.730	1.313	1.660				7,200	7,530	6,360	13,300		.080
1.050	1.14	1.20		J-55	.113	.824	.730	1.313	1.660				9,370	10,360	8,740	18,290		.080
1.050	1.14	1.20		C-75	.113	.824	.730	1.313	1.660				12,250	14,120	11,920	24,940		.080
1.050	1.14	1.20		N-80	.113	.824	.730	1.313	1.660				12,970	15,370	12,710	26,610		.080
1.315	1.70	1.80	1.72	H-40	.133	1.049	.955	1.660	1.900		.955	1.550	5,320	7,080	10,960	19,760	15,970	.107
1.315	1.70	1.80	1.72	J-55	.133	1.049	.955	1.660	1.900		.955	1.550	8,260	9,730	15,060	27,160	21,960	.107
1.315	1.70	1.80	1.72	C-75	.133	1.049	.955	1.660	1.900		.955	1.550	11,570	13,270	20,540	37,040	29,940	.107
1.315	1.70	1.80	1.72	N-80	.133	1.049	.955	1.660	1.900		.955	1.550	12,270	14,160	21,910	39,510	31,940	.107
1.660			2.10	H-40	.125	1.410					1.286	1.880	5,220	5,270			22,180	.205
1.660	2.30	2.40	2.33	H-40	.140	1.380	1.286	2.054	2.200		1.286	1.880	5,790	5,900	15,530	26,740	22,180	.185
1.660			2.10	J-55	.125	1.410					1.286	1.880	6,790	7,250			30,500	.205
1.660	2.30	2.40	2.33	J-55	.140	1.380	1.286	2.054	2.200		1.286	1.880	7,530	8,120	21,360	36,770	30,500	.185
1.660	2.30	2.40	2.33	C-75	.140	1.380	1.286	2.054	2.200		1.286	1.880	9,860	11,070	29,120	50,140	41,600	.185
1.660	2.30	2.40	2.33	N-80	.140	1.380	1.286	2.054	2.200		1.286	1.880	10,420	11,810	31,060	53,480	44,370	.185
1.900			2.40	H-40	.125	1.650					1.516	2.110	4,450	4,610			26,890	.264
1.900	2.75	2.90	2.76	H-40	.145	1.610	1.516	2.200	2.500		1.516	2.110	5,290	5,340	19,090	31,980	26,890	.252
1.900			2.40	J-55	.125	1.650					1.516	2.110	5,790	6,330			36,970	.264
1.900	2.75	2.90	2.76	J-55	.145	1.610	1.516	2.200	2.500		1.516	2.110	6,870	7,350	26,250	43,970	36,970	.252
1.900	2.75	2.90	2.76	C-75	.145	1.610	1.516	2.200	2.500		1.516	2.110	8,990	10,020	35,800	59,960	50,420	.252
1.900	2.75	2.90	2.76	N-80	.145	1.610	1.516	2.200	2.500		1.516	2.110	9,520	10,680	38,180	63,960	53,780	.252
2.063			3.25	H-40	.156	1.751					1.657	2.325	5,240	5,290			31,850	.298
2.063			3.25	J-55	.156	1.751					1.657	2.325	6,920	7,280			43,800	.298
2.063			3.25	C-75	.156	1.751					1.657	2.325	8,910	9,920			59,720	.298
2.063			3.25	N-80	.156	1.751					1.657	2.325	9,440	10,590			63,700	.298
2.375	4.00			H-40	.167	2.041	1.947	2.875					4,380	4,920	30,130			.429
2.375	4.60	4.70		H-40	.190	1.995	1.901	2.875	3.063	2.910			5,520	5,660	35,960	52,170		.387
2.375	4.00			J-55	.167	2.041	1.947	2.875					6,340	6,770	41,430			.429
2.375	4.60	4.70		J-55	.190	1.995	1.901	2.875	3.063	2.910			7,180	7,700	49,450	71,730		.387
2.375	4.00			C-75	.167	2.041	1.947	2.875					8,150	9,230	56,500			.429
2.375	4.60	4.70		C-75	.190	1.995	1.901	2.875	3.063	2.910			9,380	10,500	67,430	97,820		.387
2.375	5.80	5.95		C-75	.254	1.867	1.773	2.875	3.063	2.910			12,180	14,040	96,560	126,940		.338
2.375	4.00			N-80	.167	2.041	1.947	2.875					8,660	9,840	60,260			.429
2.375	4.60	4.70		N-80	.190	1.995	1.901	2.875	3.063	2.910			9,940	11,200	71,930	104,340		.387
2.375	5.80	5.95		N-80	.254	1.867	1.773	2.875	3.063	2.910			12,890	14,970	102,990	135,400		.338
2.375	4.60	4.70		P-105	.190	1.995	1.901	2.875	3.063	2.910			13,250	14,720	94,410	136,940		.387
2.375	5.80	5.95		P-105	.254	1.867	1.773	2.875	3.063	2.910			17,170	19,550	135,180	177,710		.338
2.875	6.40	6.50		H-40	.217	2.441	2.347	3.500	3.668	3.460			5,230	5,280	52,780	72,480		.517
2.875	6.40	6.50		J-55	.217	2.441	2.347	3.500	3.668	3.460			6,800	7,260	72,580	99,660		.517
2.875	6.40	6.50		C-75	.217	2.441	2.347	3.500	3.668	3.460			8,900	9,910	98,970	135,900		.517
2.875	8.60	8.70		C-75	.308	2.259	2.165	3.500	3.668	3.460			12,200	14,260	149,360	185,290		.496
2.875	6.40	6.50		N-80	.217	2.441	2.347	3.500	3.668	3.460			9,420	10,570	105,570	144,960		.517
2.875	8.60	8.70		N-80	.308	2.259	2.165	3.500	3.668	3.460			12,920	15,000	159,310	198,710		.496
2.875	6.40	6.50		P-105	.217	2.441	2.347	3.500	3.668	3.460			12,560	13,870	138,560	190,260		.517
2.875	8.60	8.70		P-105	.308	2.259	2.165	3.500	3.668	3.460			17,220	19,690	209,100	260,810		.496
3.500	7.70			H-40	.216	3.068	2.943	4.250					4,370	4,320	65,070			.999
3.500	9.20	9.30		H-40	.254	2.992	2.867	4.250	4.500	4.180			5,050	5,080	79,540	103,610		.869
3.500	10.20			H-40	.289	2.922	2.797	4.250					5,580	5,780	92,550			.829
3.500	7.70			J-55	.216	3.068	2.943	4.250					5,290	5,940	89,470			.999
3.500	9.20	9.30		J-55	.254	2.992	2.867	4.250	4.500	4.180			6,560	6,980	109,370	142,460		.869
3.500	10.20			J-55	.289	2.922	2.797	4.250					7,390	7,950	127,250			.829
3.500	7.70			C-75	.216	3.068	2.943	4.250					6,690	8,100	122,010			.999
3.500	9.20	9.30		C-75	.254	2.992	2.867	4.250	4.500	4.180			8,530	9,520	149,140	194,260		.869
3.500	10.20			C-75	.289	2.922	2.797	4.250					9,660	10,340	173,530			.829
3.500	12.70	12.95		C-75	.375	2.750	2.625	4.250	4.500	4.180			12,260	14,060	230,990	276,120		.729
3.500	7.70			N-80	.216	3.068	2.943	4.250					7,080	8,640	130,140			.999
3.500	9.20	9.30		N-80	.254	2.992	2.867	4.250	4.500	4.180			9,580	10,150	159,090	207,220		.869
3.500	10.20			N-80	.289	2.922	2.797	4.250					10,230	11,560	185,100			.829
3.500	12.70	12.95		N-80	.375	2.750	2.625	4.250	4.500	4.180			12,920	15,000	246,390	294,530		.729
3.500	9.20	9.30		P-105	.254	2.992	2.867	4.250	4.500	4.180			12,110	13,330	208,800	271,970		.869
3.500	12.70	12.95		P-105	.375	2.750	2.625	4.250	4.500	4.180			17,220	19,690	323,390	386,570		.729
4.000	9.50			H-40	.226	3.548	3.423	4.750					3,580	3,960	72,000			1.222
4.000		11.00		H-40	.262	3.476	3.351		5.000				4,420	4,580		123,370		1.173
4.000	9.50			J-55	.226	3.548	3.423	4.750					4,650	5,440	99,010			1.222
4.000		11.00		J-55	.262	3.476	3.351		5.000				5,750	6,300		169,220		1.173
4.000	9.50			C-75	.226	3.548	3.423	4.750					5,860	7,420	135,010			1.222
4.000		11.00		C-75	.262	3.476	3.351		5.000				7,330	8,600		230,750		1.173
4.000	9.50			N-80	.226	3.548	3.423	4.750					6,120	7,910	144,010			1.222
4.000		11.00		N-80	.262	3.476	3.351		5.000				7,780	9,170		245,140		1.173
4.500	12.60	12.75		H-40	.271	3.958	3.833	5.200	5.563				3,930	4,220	104,360	144,920		1.521
4.500	12.60	12.75		J-55	.271	3.958	3.833	5.200	5.563				5,120	5,800	143,500	198,030		1.521
4.500	12.60	12.75		C-75	.271	3.958	3.833	5.200	5.563				6,430	7,920	195,680	270,040		1.521
4.500	12.60	12.75		N-80	.271	3.958	3.833	5.200	5.563				6,810	8,430	208,730	288,040		1.521

TUBING TABLES REPRINTED FROM PETROLEUM EQUIPMENT

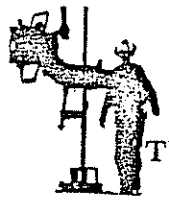
Fig. 3J-7 (Courtesy Otis Engineering Corp.)

THE SPECIAL TUBING JOINTS



†Special clearance couplings—for certain applications it is an accepted practice to turn 2⅜" couplings to 2.910 dia. and 2⅞" couplings to 3.450 dia.
††See U. S. Steel column for complete buttress thread data.
†Diameters, other than those shown and where symbol appears, are subject to inquiry.
*Meets API integral joint standards.
** "DS" Int. Joint OD +0 to −1/16"

Fig. 3J-8 (Courtesy Otis Engineering Corp.)



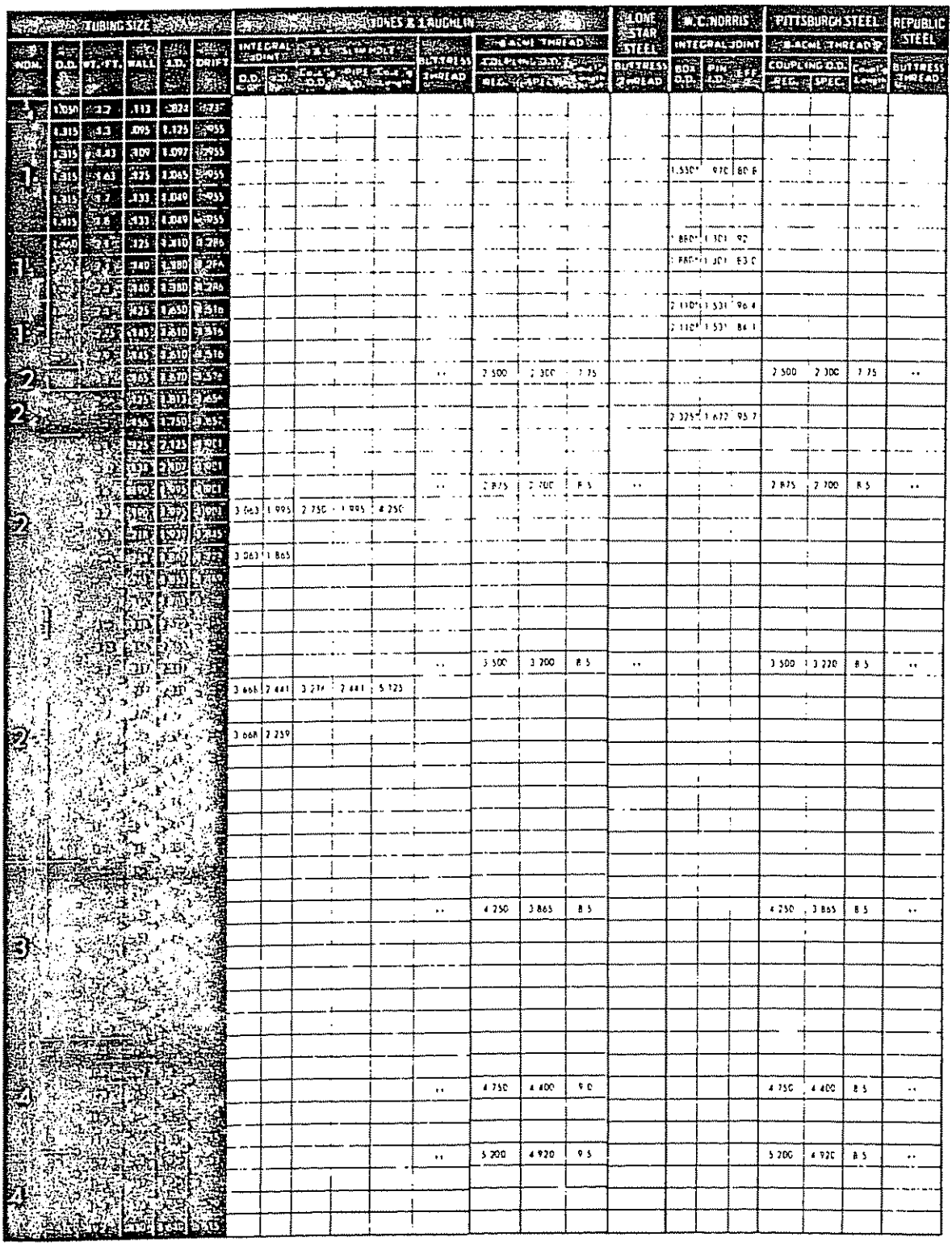
TUBING TABLES

			1 310	824	100				1 327	1 300	11/16	140									
1 552	970	1" EUE																			
			1 550	1 049	100				1 552	1 525	970	116									
1 883	1 300	1¼" NU*																			
			1 880	1 380	130	1 358	1 300	95	1 883	1 858	1 300	114									
2 313	1 530	1½" NU																			
			2 310	1 610	130	2 094	1 530	95	2 313	2 094	1 530	110									
2 330	1 690	1½" EUE	2 300	1 750	100	2 285	1 700	95	2 330	2 300	1 700	109									
2 700	1 925	2" NU																			
			2 700	1 995	100	2 610	1 945	95	2 700	2 630	1 945	108									
									2 700		1 890	105									
													2⅜"	1.605	110						
													2½"	1 795	130						
													3½	1 645	107						
3 220	2 441	2" NU																			
						3 130	2 375	95	3 220	3 155	2 375	102									
													3½	2 265	108						
													3½	2.200	108						
													3½	2 130	108						
													3½	2 000	100						
						3 780	2 920	95	3 865	3 805	2 920	105									
									3 865		2 878	101									
													4½"	2.687	119						
													4½"	2 700	120						
						4 290	3 395	95	4 343	4 315	3 395	105									
													4½	3.275	108						
						4 790	3 865	95	4 855	4 825	3 865	105									
													5½	3.765	108						
													5½"	3.560	108						

ALL TUBING AND JOINT DIMENSIONS IN INCHES.
Efficiency percentage based on average metal areas of pin, unless otherwise noted.
Weight per foot in lb based on average of coupled and threaded 20 ft joint.

Fig. 3J-8 (Cont'd) (Courtesy Otis Engineering Corp.)

THE SPECIAL TUBING JOINTS

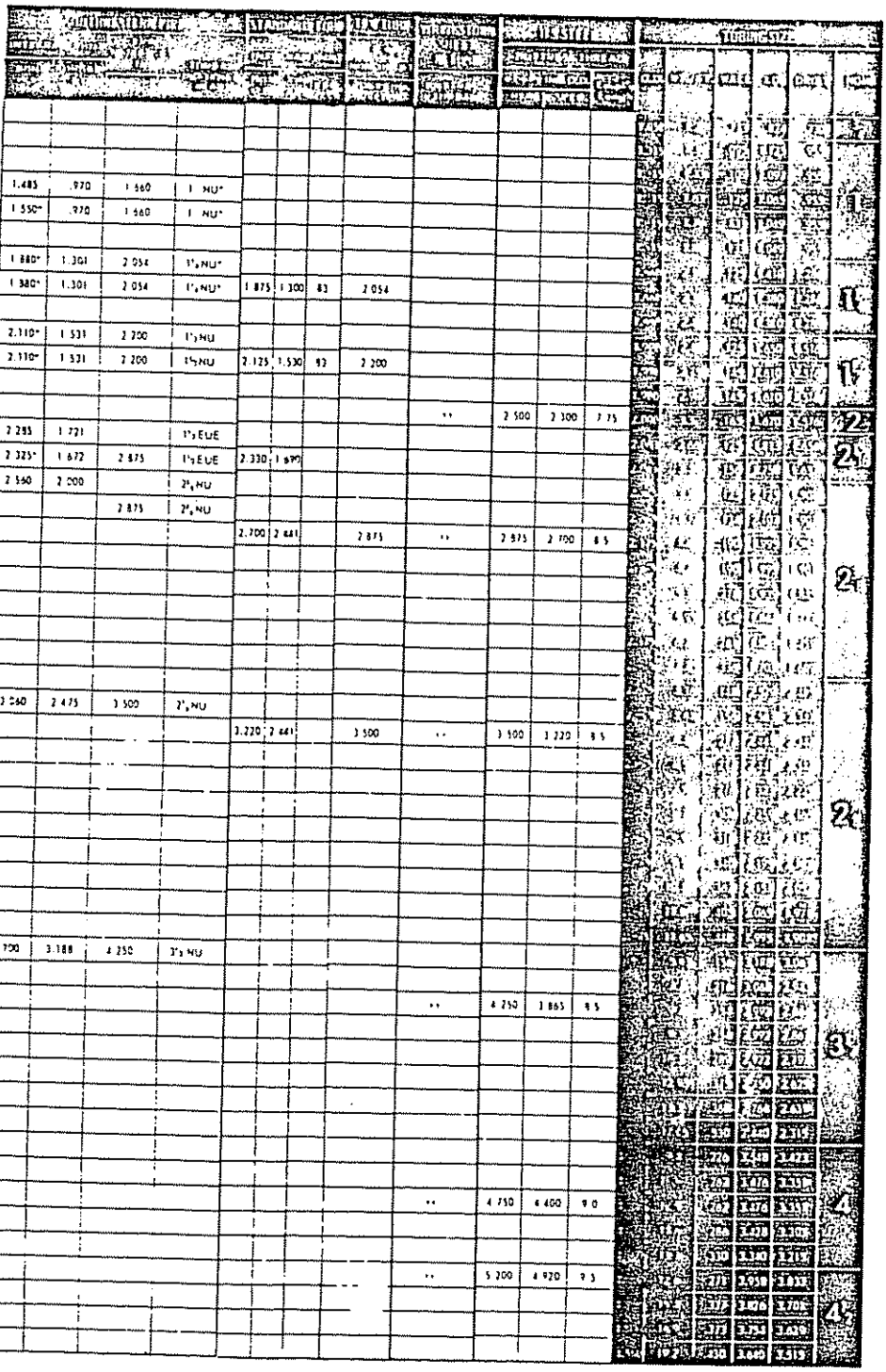


†Special clearance couplings—for certain applications it is an accepted practice to turn 2⅜" couplings to 2.910 dia. and 2⅞" couplings to 3.450 dia.
*Meets API integral joint standards.
†Diameters, other than those shown and where symbol appears, are subject to inquiry.
††See U. S. Steel column for complete buttress thread data.

Fig. 3J-8 (Cont'd) (Courtesy Otis Engineering Corp.)



TUBING TABLES



1.485	.970	1.560	1 NU*								
1.550*	.970	1.560	1 NU*								
1.880*	1.301	2.054	1¼NU*								
1.880*	1.301	2.054	1¼NU*	1.875	1.300	83	2.054				
2.110*	1.531	2.200	1½NU								
2.110*	1.531	2.200	1½NU	2.125	1.530	93	2.200				
								"	2.500	2.300	7.75
2.285	1.721		1½EUE								
2.325*	1.672	2.875	1½EUE	2.330	1.690						
2.560	2.200		2½NU								
		2.875	2½NU								
				2.700	2.441		2.875	"	2.875	2.700	8.5
3.060	2.475	3.500	2½NU								
				3.220	2.441		3.500	"	3.500	3.220	8.5
3.790	3.188	4.250	3½NU								
								"	4.250	3.865	9.5
								"	4.750	4.400	9.0
								"	5.200	4.920	9.3

ALL TUBING AND JOINT DIMENSIONS IN INCHES.
Efficiency percentage based on average metal areas of pin, unless otherwise noted.
Weight per foot in lb based on average of coupled and threaded 20 ft joint.

Fig. 3J-8 (Cont'd) (Courtesy Otis Engineering Corp.)

TUBING MAKE-UP TORQUE GUIDE

THREADED-AND-COUPLED JOINTS

JOINT NAME	TUBING GRADE	¾ 1.050" OD			1 1.315" OD			1¼ 1.660" OD			1½ 1.900" OD			2 2.000" OD		
		MIN.	OPT.	MAX.	MIN.	OPT.	MAX.	MIN.	OPT.	MAX.	MIN.	OPT.	MAX.	MIN.	OPT.	MAX.
[illegible]	J-55 N-80		• •			• •			• •			• •				
[illegible]	J-55 N-80		• •			• •			• •			• •				
[illegible]																
[illegible]																
[illegible]	J-55 N-80															
[illegible]					Power-tight position indicated by make-up to base of J-L stamped on tubing.											
[illegible]					Power-tight position indicated by make-up to base of (P) stamped on tubing;											
[illegible]					Joint has very low make-up torque requirement; may be easily overtorqued.											
[illegible]									•			•				
[illegible]					Power make-up should be to point of diamond-stamped on tubing; joint has low											

INTEGRAL JOINTS

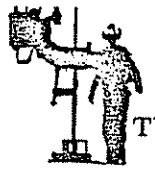
JOINT NAME	TUBING GRADE	¾ 1.050" OD			1 1.315" OD			1¼ 1.660" OD			1½ 1.900" OD			2 2.000" OD		
		MIN.	OPT.	MAX.	MIN.	OPT.	MAX.	MIN.	OPT.	MAX.	MIN.	OPT.	MAX.	MIN.	OPT.	MAX.
[illegible]						•			•			•				
[illegible]	J-55 N-80 P-105															
[illegible]		150		250	150		300	200		400	400		550			
[illegible]					150		300	200		400	400		550	450		600
[illegible]																
[illegible]						350			375			425				
[illegible]																
[illegible]		450	500	550	450	500	550	450	500	550	450	500	550			
[illegible]	J-55 N-80								400 600			600 800				
[illegible]	J-55 N-80 P-105		200 300 300			300 400 400			400 600 600			600 800 800				
[illegible]	J-55 N-80 P-105															
[illegible]					Recommended torque for this integral joint is the same as API threaded-and-coupled											
[illegible]						•			•			•				
[illegible]					Joint has very low make-up torque requirement; may be easily overtorqued.											
[illegible]									375			425				

* Ft lb torque values not established by the manufacturer for these joints.

[a] Torque for 2⅜-in. DSS joints in tubing weights of 4.7 and 5.3 lb per ft.

[b] Torque for 2⅜-in. DSS joints in tubing weights of 5.95, 6.2, and 7.7 lb per ft.

Fig. 3J-8 (Cont'd) (Courtesy Otis Engineering Corp.)



TUBING TABLES

Information on the make-up requirements of tubing joints and couplings on these pages are the recommendations of the manufacturers, or in the case of API joints, the suggestions of the American Petroleum Institute's Committee on Standardization of Tubular Goods.

The tables should be used only as a guide to proper make-up, since many factors affect the torque requirements necessary for proper joint assembly.

Higher torque requirements may be necessary if these conditions exist:

1. Joint threads are worn, burred or improperly cleaned.
2. Threads are plated.
3. Sealing rings are used in couplings or joints.

Less torque may be needed for proper make-up if friction-reducing resilient thread compounds are used. One compound has been known to reduce make-up friction as much as 50 percent.

If power tongs are used, proper make-up necessarily depends on proper calibration, proper rig-up, proper tool speed, and similar factors.

It should be noted that some manufacturers give recommended optimum ft-lb torque figures while others give minimum (below which an effective seal is not made) and maximum (above which may destroy threads or joint) figures.

All figures below are in ft-lb of torque.

2 1/16" 2.063" OD			2 3/8" 2.375" OD			2 7/8" 2.875" OD			3 1/2" 3.500" OD			4" 4.000" OD			4 1/2" 4.500" OD			TUBING GRADE	JOINT NAME
MIN.	OPT.	MAX.	MIN.	OPT.	MAX.	MIN.	OPT.	MAX.	MIN.	OPT.	MAX.	MIN.	OPT.	MAX.	MIN.	OPT.	MAX.		
			1000 1300	1200 1500	1700 2000	1300 1600	1500 1800	2000 2300		•			•			•		J-55 N-80	API Non-Upset
			1000 1300	1200 1500	1700 2000	1300 1600	1500 1800	2000 2300		•			•			•		J-55 N-80	API External Upset
			800	1000	1200	1000	1200	1400	1200	1400	1600								Armco Spaced Joint
																			Atlas Bradford Integral Joint
			700 700		900 900	900 900		1000 1000										J-55 N-80	Jones & Laughlin Non-Upset
																			Jones & Laughlin Hy-Ten
ft lb torque ratings in API range																			Lone Star Steel Co.
																			Republic Steel Corp.
				•			•			•									T.R.W. Non-Upset
make-up torque requirement, joint may be easily overtorqued.																			Tex-Tube Co.
																			U.S. Steel Corp. Tubing

2 1/16" 2.063" OD			2 3/8" 2.375" OD			2 7/8" 2.875" OD			3 1/2" 3.500" OD			4" 4.000" OD			4 1/2" 4.500" OD			TUBING GRADE	JOINT NAME
MIN.	OPT.	MAX.	MIN.	OPT.	MAX.	MIN.	OPT.	MAX.	MIN.	OPT.	MAX.	MIN.	OPT.	MAX.	MIN.	OPT.	MAX.		
	•																		API Integral Joint
				1700 1700 1700	2300 2800 3300		2100 2100 2100	2600 3100 3600		2400 2400 2400	3000 3500 4000							J-55 N-80 P-105	Armco Extreme Line
500		650	750		900	950		1200	1100		1300								Atlas Bradford DS
550		900	750[c] 800[b]		1000[c] 1100[b]	950[c] 1000[d]		1200[c] 1300[d]	1100		1400	1200		1600	1200		1600		Atlas Bradford DSS
																			Atlas Bradford Atlas LJ
	450			525			625												Aztec LJ
				1800			2500			3000									Cigy Tool Type B Grades
450	500	550	450	500	550														Gulf States Tube GST Streamline
	700 900			1100 1500			1500 2100			2500 3000								J-55 N-80	Hydril A-25
	700 900 900			1100 1500 1500			1500 2100 2100			2500 3000 3000			3000 3500 3500			3500 3500 3500		J-55 N-80 P-105	Hydril CS
				1600 2200 2200			2200 3000 3000			3500 4000 4000			3500 4000 4000			3500 4000 4000		J-55 N-80 P-105	Hydril PH-6
joints of the same size and grade																			Jones & Laughlin LJ
	•																		W. C. Norris LJ
																			Southwestern Pipe LJ
	450			525			625												Standard Tube LJ

[c] Torque for 2 7/8-in. DSS joints in tubing weight of 6.5 lb per ft.
[d] Torque for 2 7/8-in. DSS joints in tubing weights of 7.9 lb per ft and heavier.
• Joint not now being manufactured, but still found in the field.

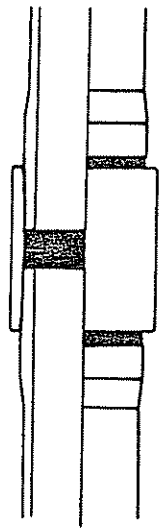
Fig. 3J-8 (Cont'd) (Courtesy Otis Engineering Corp.)

TUBING JOINT IDENTIFICATION CHARTS

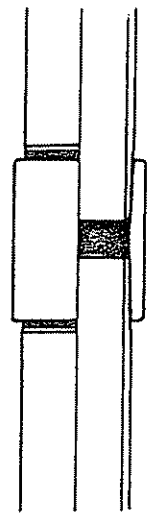
The addition of descriptive data on various API and special tubing joints and couplings has been suggested for sometime. The material presented on these pages marks a "first" for compilation of this oft-needed information.

Schematic drawings appearing here are designed for identification purposes only, and are not drawn to scale. Certain features on several joints have been enlarged or accentuated as an aid to field identification.

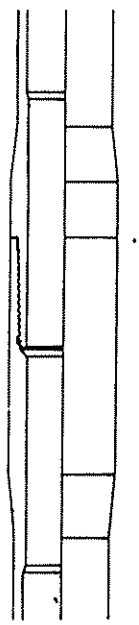
Descriptive data accompanying the special joint diagrams have been compiled by their respective manufacturers to reflect individual recommendations for service, applications and utility.



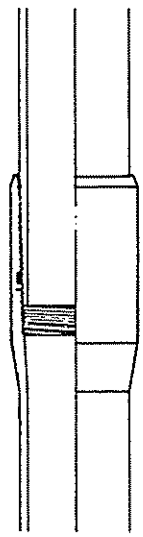
API EXTERNAL UPSET coupling is furnished by many manufacturers in both regular and special clearance sizes. The special clearance coupling has a smaller OD. Short and long thread lengths also available. Ten round threads per inch are turned on external upset tubing 1.900-in. OD and smaller; 8 round threads per inch are used on 2⅜-in. OD external upset tubing and larger. Special clearance couplings are identified by a black band painted around the center of the usual color band used to indicate steel grade.



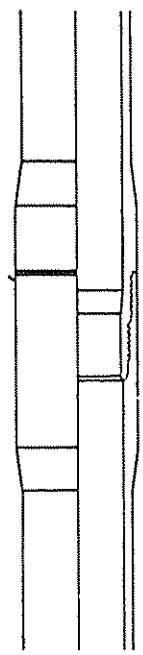
API NON-UPSET threaded-and-coupled (T&C) joint utilizes 10 round threads per inch in tubing from 1.050-in. OD up to and including 3½-in. OD sizes; the 4 and 4½-in. OD tubing has 8 round threads per inch. Length of the coupling for each size non-upset tubing is different from any other, ranging from 3 3/16-in. for the coupling on 1.050-in. OD tubing to 6½-in. in length for the coupling used on 4½-in. OD tubing. See "who makes tubing" section for makers of API monogrammed tubing and couplings.



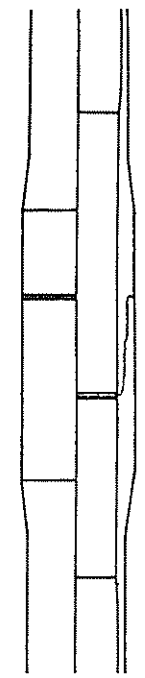
ATLAS BRADFORD DSS (Dual Seal Taper) integral joints are designed for minimum clearance applications, utilizes a Teflon seal to insure a leak-proof connection. Joint efficiencies are over 100%. The modified buttress thread assures quick stabbing and fast make-up. The DSS uses a heavier upset than the DS and is applied to jobs requiring higher efficiencies and ruggedness. It is designed for maximum castability.



GST STREAMLINE is a full opening joint made by Gulf States Tube Corp. The pin member is non-upset, having modified truncated buttress threads extending to full run-out. The box member is upset and machine-turned for optimum clearance and included in the box is a teflon ring which assures a positive seal even at highest pressures. The box OD is beveled to prevent hang-up when running multiple completions. Reduced carbide diameters afford greater clearance in "tight hole" completions. The ID, having full opening, provides maximum clearance for running tools. Makeup requires relatively few turns to power-tight position.

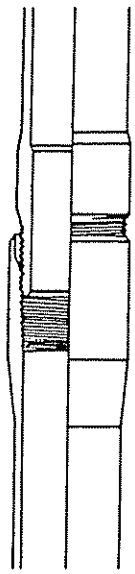


HYDRIL "A-95": This is an economy form of the type "CS" (interchangeable with "CS"). The tubing upset is like the API tubing upset, produced as standard upsets in the mills. "A-95" is principally used in economy multiple completions where mechanical strength, and running and pulling streamlining determine the choice of tubing joints.

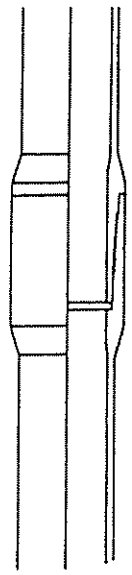


HYDRIL "CS" is an "all purpose" tubing connection — with integral joints on upset tubing. It is particularly used on premium grade steels, and in high pressure service, and in especially tricky completions —and also, in multiple completions. "CS" is a "slim" type joint—yet a full strength joint.

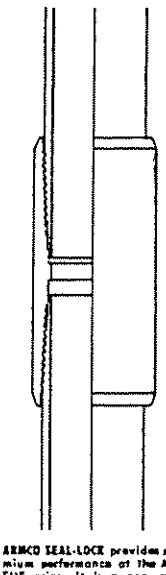
Fig. 3J-8 (Cont'd) (Courtesy Otis Engineering Corp.)



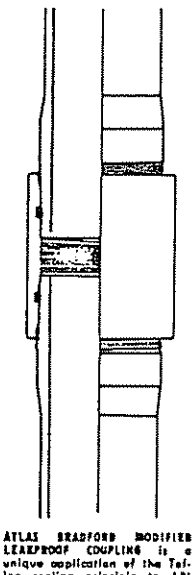
API INTEGRAL JOINT, recently adopted, 10 round threads per inch on all four approved sizes. Illustration has exaggerated the slight internal upset that exists on the pin end of the joint, a feature designed to prevent pullout and joint collapse. Like other API tubing joints, hand-tight standoff is two complete thread turns. When power-tight make-up is achieved, the box end of the joint covers the last thread on the pin end.



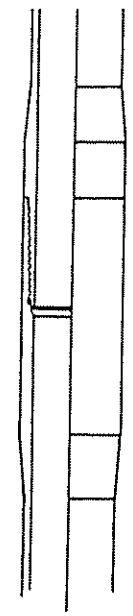
ARMCO EXTREME LINE is a high strength integral joint, specifically developed to meet the requirements of deeper production. This joint has external upset ends; with a shoulder, threads and seal machined on each end. Male and female members make up into a sealed joint that is shoulder tight. The joint is streamlined internally and externally, with a minimum outside diameter. The unique seal resists leakage, even under extreme conditions.



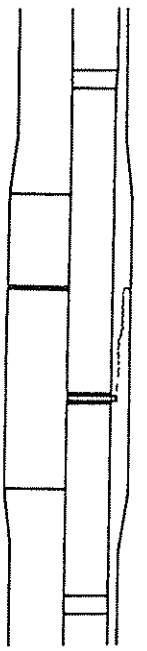
ARMCO SEAL-LOCK provides premium performance at the API-EUE price. It is a non-upset, threaded and coupled joint that permits fast stabbing and make and break while minimizing danger of cross-threading. Thread and seal resist damage despite repeated make-up. Strength of joint is essentially equal to that of the pipe body, and an internal shoulder lessens danger of overtorque. This gas-tight connection with metal-to-metal seal also offers simplicity of design that permits machining of accessory stock in local machine shops.



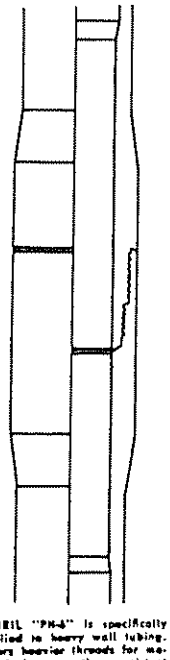
ATLAS BRADFORD MODIFIED LEAKPROOF COUPLING is a unique application of the Teflon sealing principle to API and non-API tubing or casing couplings. This design converts standard couplings to high pressure connections. Completions have been made with this coupling as high as 13,000 psi, and is used in helium storage wells. A complete magnetic particle inspection of each coupling is included in the modification process. Strength of couplings in tension, burst and collapse are not effected by the modification process.



ATLAS BRADFORD DS (Dual Seal) integral joints are designed for minimum clearance applications. A Teflon seal is utilized to insure their leakproof character. Joint efficiencies are over 100%. The rugged modified buttress thread assures quick stabbing and fast make-up. It is designed for maximum castability.

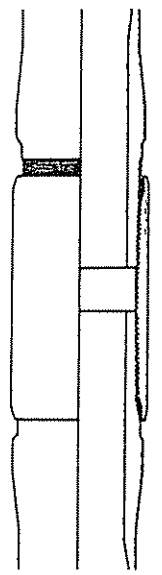


HYBRIL "CS-CB" is especially intended for internal plastic coating — for total protection against corrosion attack at the connections. It features a "corrosion barrier" ring assembled in the box to act as a corrodent sealing gasket between the plastic coated end faces of pin and box. All running and performance characteristics of "CS-CB" are same as standard "CS".

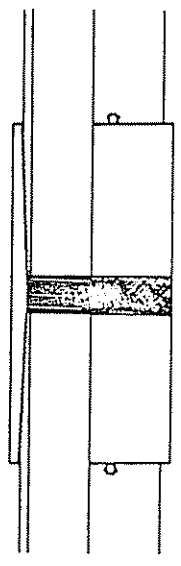


HYBRIL "PH-6" is specifically applied to heavy wall tubing. Offers heavier threads for mechanical strength consistent with heavier pipes. Popular for high strength joints on "C-55" and "C-75" tubing steels (in heavy walls) for potentially embrittling environment.

"PH-6" is also widely used on high strength standard steels (N-80 and P-105) in extreme depth wells with extremely high production pressure.

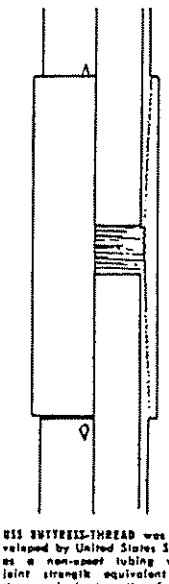


J&L SLIM HOLE is designed with maximum joint clearance and 100% joint efficiency in tension to facilitate use in single or multiple completion wells. The T&C joint is furnished with a special pitch diameter, API 10-round threads cut on API external upset tubing. The joint diameter is smaller than API regular non-upset tubing. The bearing face of the coupling is protected by the unturned portion of the upset, thus providing a streamlined exterior to prevent any tendency to hang-up. The joint is not interchangeable with other connections.



PITTSBURGH STEEL'S 8-ACME THREAD is a non-upset long coupling, high strength joint for maximum clearance and exceptional leak resistance in deep wells, multiple completions, and high pressures. It offers more perfect threads in an 8-pitch design. Its broad-crest threads allow faster (and recorded) make-ups and breakouts with less thread damage. Coupling threads are lubricoated for a better seal.

J&L Steel Corp. is now a producer of this joint, under Pittsburgh Steel license. A "J&L" dia mark replaces Pittsburgh's keystone on the tubing there above.

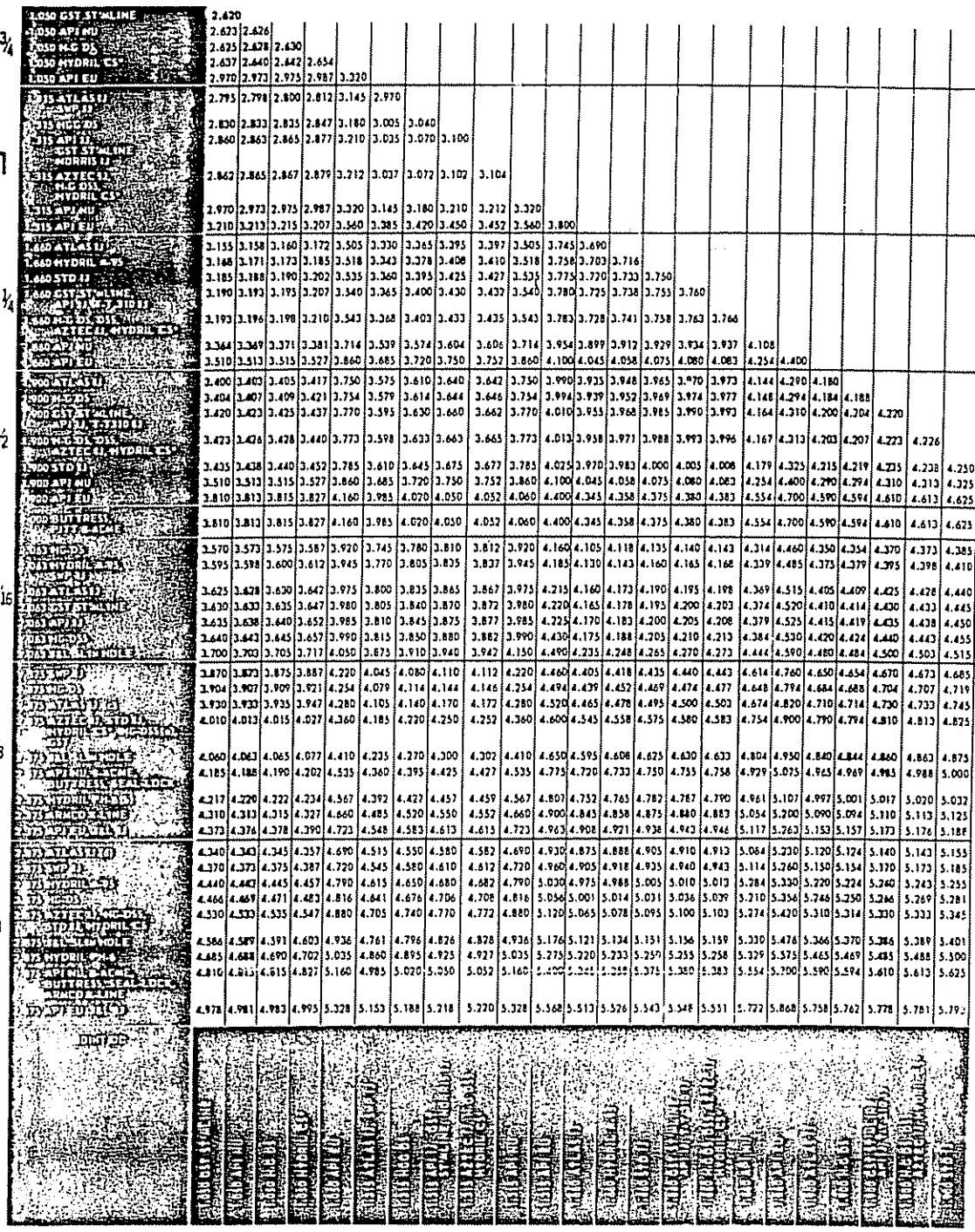


USS UNIVERSAL-THREAD was developed by United States Steel as a non-upset tubing with joint strength equivalent to the nominal strength of the tubing body. This joint is designed for maximum clearance; high tensile and compression strength; gas-lock tightness at 15,000 psi through 100 temperature cycles between 60 and 200 F; ability to withstand at least 12 repeated make-ups and breaks without thread damage or impairment to joint performance, and accurate threads and better finish due to carbide threading.

Several manufacturers are licensed to make this tubing, as indicated on the Special Tubing Joint tables on previous pages.

Fig. 3J-8 (Cont'd) (Courtesy Otis Engineering Corp.)

PARALLEL COUPLING OR JOINT DIAMETER



LEGEND:

API IJ—Integral Joint; EU—External Upset T&C; NU—Non-Upset T&C;

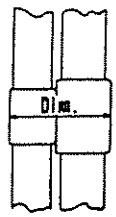
ATLAS IJ—Atlas Bradford Co. Integral Joint; AZTEC IJ—Aztec Mfg. Co. Integral Joint; ARMCO X-LINE—Armco Steel Corp. Extreme Line Integral Joint

BUTTRESS—U. S. Steel Corp. Buttress Thread T&C; V-ACME—Pittsburg Steel V-Acme Thread T&C; GST ST'MLINE—Gulf States Tube Corp. Streamline Integral Joint;

Fig. 3J-8 (Cont'd) (Courtesy Otis Engineering Corp.)

SAVES DUAL COMPLETION CALCULATIONS

(COMBINED DIAMETERS JOINT-TO-JOINT OR COUPLING-TO-COUPLING)



The Dimensions listed in this Chart are the exact minimum combined OD as shown in the illustration. It is recommended that a diametrical clearance of 3/16" to 5/16" be subtracted from the theoretical ID of the casing.

Clamped Tubing Strings—
The type and size clamp determines the center-to-center spacing of the tubing and thus in large measure determines the OD of the assembly. For maximum accuracy it is common practice to draw top view of tubing assembly to full scale. Since couplings or tubing connections are staggered, the OD of only one of the joints is used. This is normally the tubing string having the largest coupling or tubing joint diameter.
Where more than two strings of tubing are to be run, the most universally used method of determining clearance is to draw a top view of tubing within the casing to full scale. Operators generally accept combinations giving a clearance of .010 or better.

Three, Four or More Strings of Tubing—Multiple Tubing Installations Using One Size—
Where all strings used in the installation are to be the same size and are equipped with joints of the same OD, the following simplified formula may be used to determine the smallest internal casing diameters in which strings may be run.

A. Two Strings—
Minimum Running Diameter = 2 (OD of Tubing Joint).

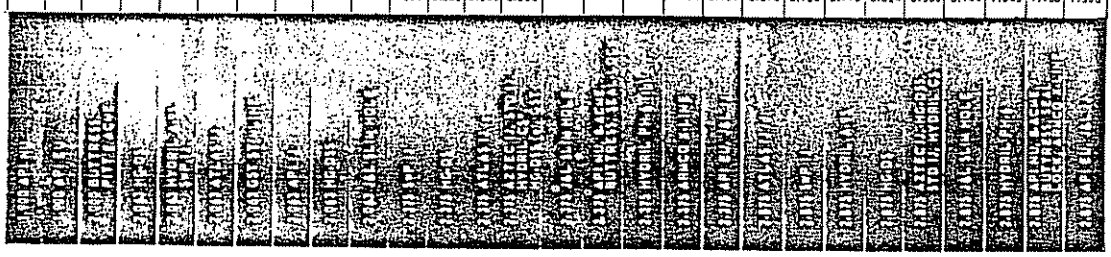
B. Three Strings—
Minimum Running Diameter = 2.1547 (OD of Tubing Joint).

C. Four Strings—
Minimum Running Diameter = 2.414 (OD of Tubing Joint).

NOTE: Tolerances on joint ODs will vary from +.015" to +1%, depending upon manufacturer or API specs. *Calculated for regular turned OD; special OD available, see Tubing Tables. (a) Calculated on .190 or .218-in. wall tubing; heavier wall tubing has larger joint OD. (b) Calculated on .254-in. wall tubing; heavier wall tubing avail. has larger joint OD. (c) Calculated on .190-in. wall tubing; .134-in. wall tubing has smaller joint OD. (d) Calculated on .148-in. wall tubing; heavier wall tubing avail. has larger joint OD.

—Furnished by Atlas Bradford Company

4.400																														
4.700	5.000																													
4.700	5.000	5.000																												
4.460	4.760	4.760	4.520																											
4.485	4.785	4.785	4.545	4.570																										
4.515	4.815	4.815	4.575	4.600	4.630																									
4.520	4.820	4.820	4.580	4.605	4.635	4.640																								
4.525	4.825	4.825	4.585	4.610	4.640	4.645	4.650																							
4.530	4.830	4.830	4.590	4.615	4.645	4.650	4.655	4.660																						
4.590	4.890	4.890	4.650	4.675	4.705	4.710	4.715	4.720	4.780																					
4.760	5.060	5.060	4.820	4.845	4.875	4.880	4.885	4.890	4.950	5.120																				
4.794	5.094	5.094	4.854	4.879	4.909	4.914	4.919	4.924	4.984	5.154	5.188																			
4.820	5.120	5.120	4.880	4.905	4.935	4.940	4.945	4.950	5.010	5.130	5.214	5.240																		
4.900	5.200	5.200	4.960	4.985	5.015	5.020	5.025	5.030	5.090	5.260	5.294	5.320	5.400																	
4.950	5.250	5.250	5.010	5.035	5.065	5.070	5.075	5.080	5.140	5.310	5.344	5.370	5.450	5.500																
5.075	5.375	5.375	5.135	5.160	5.190	5.195	5.200	5.205	5.265	5.435	5.469	5.495	5.575	5.625	5.750															
5.107	5.407	5.407	5.167	5.192	5.222	5.227	5.232	5.237	5.297	5.467	5.501	5.527	5.607	5.657	5.782	5.814														
5.200	5.500	5.500	5.260	5.285	5.315	5.320	5.325	5.330	5.390	5.560	5.594	5.620	5.700	5.750	5.875	5.907	6.000													
5.263	5.563	5.563	5.323	5.348	5.378	5.383	5.388	5.393	5.453	5.623	5.654	5.683	5.763	5.813	5.938	5.970	6.063	6.126												
5.230	5.530	5.530	5.290	5.315	5.345	5.350	5.355	5.360	5.420	5.590	5.624	5.650	5.730	5.780	5.905	5.937	6.030	6.093	6.060											
5.260	5.560	5.560	5.320	5.345	5.375	5.380	5.385	5.390	5.450	5.620	5.651	5.680	5.760	5.810	5.935	5.967	6.060	6.123	6.090	6.120										
5.330	5.630	5.630	5.390	5.415	5.445	5.450	5.455	5.460	5.520	5.690	5.724	5.750	5.830	5.880	6.005	6.037	6.130	6.193	6.160	6.190	6.260									
5.356	5.656	5.656	5.416	5.441	5.471	5.476	5.481	5.486	5.546	5.716	5.750	5.776	5.856	5.906	6.031	6.063	6.156	6.219	6.186	6.126	6.286	6.317								
5.420	5.720	5.720	5.480	5.505	5.535	5.540	5.545	5.550	5.610	5.780	5.814	5.840	5.910	5.970	6.095	6.127	6.220	6.283	6.250	6.280	6.350	6.376	6.440							
5.476	5.776	5.776	5.536	5.561	5.591	5.596	5.601	5.606	5.666	5.836	5.870	5.896	5.976	6.026	6.151	6.183	6.276	6.339	6.306	6.336	6.406	6.437	6.496	6.552						
5.575	5.875	5.875	5.635	5.660	5.690	5.695	5.700	5.705	5.765	5.935	5.969	5.995	6.075	6.125	6.250	6.282	6.375	6.438	6.405	6.435	6.505	6.531	6.595	6.651	6.750					
5.700	6.000	6.000	5.760	5.785	5.815	5.820	5.825	5.830	5.890	6.060	6.094	6.120	6.200	6.250	6.375	6.407	6.500	6.563	6.530	6.560	6.630	6.656	6.720	6.776	6.875	7.000				
5.868	6.168	6.168	5.728	5.593	5.983	5.988	5.993	5.998	6.058	6.228	6.262	6.288	6.368	6.418	6.543	6.575	6.668	6.731	6.698	6.728	6.798	6.824	6.938	6.944	7.043	7.168	7.336			



HYDRIL A-95—Hydril Co. A-95 Integral Joint;
HYDRIL CS—Hydril Co. Integral Joint;
HG-DS—Atlas Bradford Co. HG-DS Integral Joint;
HG-DSS—Atlas Bradford Co. HG-DSS Integral Joint;
JEL SLIM-HOLE—Jones & Laughlin Steel Corp. Slim-Hole;
MORRIS II—W. C. Morris Div. Dresser Corp. Integral Joint;
SEAL-LOCK—Armco Steel Corp. Seal-Lock TEC;
STANDARD II—Standard Tube Co. Integral Joint;
SWP II—Southwestern Pipe, Inc., Integral Joint;
TT-30—Tex-Tube, Inc. Tex-Trol Integral Joint;
TT 320—Tex-Tube, Inc. Series 320 Integral Joint.

Fig. 3J-8 (Cont'd) (Courtesy Otis Engineering Corp.)

4½" OD

Casing Dimensions

Wt/ft (Lbs)		Casing Body				Areas (Sq. In.)	
Nominal T&C	Plain End	Wall (In.)	ID (In.)	Drift (In.)	D/T Ratio	PE and Buttress	Last Perfect Thread (8Rd)
9.50	9.40	.205	4.090	3.965	21.95	2.766	1.775
10.50	10.23	.224	4.052	3.927	20.09	3.009	2.018
11.60	11.35	.250	4.000	3.875	18.00	3.338	2.347
13.50	13.04	.290	3.920	3.795	15.52	3.836	2.844
15.10	14.98	.337	3.826	3.701	13.35	4.407	3.416
[1]16.60	16.52	.375	3.750	3.625	12.00	4.860	3.868
[1]19.10	18.96	.437	3.626	3.501	10.30	5.578	4.587

Coupling Dimensions

OD REG.= 5.000 INCHES
OD SCC= 4.875 INCHES
LENGTH SHORT= 6.250 INCHES
LENGTH LONG= 7.000 INCHES
LENGTH BUTT.= 8.875 INCHES
D1 SHORT= 4.358 INCHES
D1 LONG= 4.334 INCHES
D1 BUTT.= 4.388 INCHES
AREA SHORT= 4.719 SQ.INCHES
AREA LONG= 4.882 SQ.INCHES
AREA BUTTRESS= 4.512 SQ.INCHES
AREA BUTT.SCC= 3.543 SQ.INCHES

Wt/ft	Grade	Collapse		[3]Joint Strength												Plain End Strength	
		[1,2] Feet @ 1.125	Psi	Round				Buttress									
				Short		Long		Regular				Special Clearance				Yield (Min.)	[3]Ultimate (Avg.)
								Same Grade		Higher Grade		Same Grade		Higher Grade			
				[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	Kips	Kips
9.50	H-40	4920	2770	4500	77											111	207
9.50	K-55	5880	3310	6550	112											152	296
10.50	K-55	7130	4010	7720	146			13170	249	13170	249	13170	249	13170	249	166	322
11.60	K-55	8820	4960	8140	170	8620	180	13270	277	13270	277	13270	277	13270	277	184	357
11.60	C-75	10900	6130			10150	212	13790	288	13790	288	13790	288	13790	288	250	367
13.50	C-75	14520	8170			10580	257	13620	331	13620	331	13170	320	13620	331	288	422
11.60	N-80	11290	6350			10680	223	14560	304	14560	304	14560	304	14560	304	267	384
13.50	N-80	15180	8540			11110	270	14360	349	14360	349	13870	337	14360	349	307	441
11.60	L-80	11290	6350			10150	212	13940	291	13940	291	13940	291	13940	291	267	367
13.50	L-80	15180	8540			10580	257	13740	334	13740	334	13170	320	13740	334	307	422
11.60	C-95	12460	7010			11210	234	15570	325	—	—	15570	325	—	—	317	414
13.50	C-95	17160	9650			11690	284	15390	374	—	—	14530	353	—	—	364	476
[1]15.10	C-95	22670	12750			12550	341	15780	429	—	—	12990	353	—	—	419	546
[1]16.60	C-95	25800	14510			12920	386	15060	450	—	—	11810	353	—	—	462	603
[1]19.10	C-95	29620	16660			13320	458	13090	450	—	—	10270	353	—	—	530	692
11.60	P-110	13440	7560			13360	279	18440	385	18440	385	18440	385	18440	385	367	464
13.50	P-110	18970	10670			13910	338	18230	443	18230	443	17330	421	18230	443	422	533
15.10	P-110	25460	14320			14940	406	18730	509	18730	509	15490	421	18730	509	485	613
[1]16.60	P-110	29880	16810			15360	459	17940	536	18780	561	14090	421	18040	539†	535	676
[1]19.10	P-110	34290	19290			15850	545	15590	536	18730	644	12250	421	15680	539†	614	775
[1]15.10	V-150	32200	18110			19090	519	24210	658	—	—	19830	539	—	—	661	749
[1]16.60	V-150	39700	22330			19680	588	22960	686	—	—	18040	539	—	—	729	826
[1]19.10	V-150	46760	26300			20270	697	19950	686	—	—	15680	539	—	—	837	943

Fig. 3J-9 (1A) (Courtesy U.S. Steel)

USS Seamless Casing

Maximum Coupling Performance Properties

Grade	Tension (Kips)			Int. Yield Press. (Psi)		
	Round	Buttress		Round	Buttress	
	Long	Regular	SCC	Long	Regular	SCC
K-55	441	407	320	7330	6730	5490
C-75	441	407	320	9990	9180	7490
N-80	464	429	337	10660	9790	7990
L-80	441	407	320	10660	9790	7990
C-95	487	450	353	12650	11630	9490
P-110	580	536	421	14650	13460	10990
V-150	742	686	539	19980	18360	14980

— Not Available

* Same as Round T&C Pipe

† Next Higher Grade Insufficient for Maximum Pressure of Pipe

Wt/ft	Grade	Minimum Internal Yield Pressure						Mill Test Pressure						80% Minimum Yield Strength Also Maximum Formation Fracturing Pressure					
		Pipe Body	Round	Buttress				PE	Round	Buttress				PE	Round	Buttress			
				Regular		Special Clearance				Regular		Special Clearance				Regular		Special Clearance	
				Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade
9.50	H-40	3190	*					2900	2900					2900	*				
9.50	K-55	4380	*					3000	3000					4000	*				
10.50	K-55	4790	*	*	*	*	*	3000	3000	3000	4400	4400	4400	4400	*	*	*	*	*
11.60	K-55	5350	*	*	*	*	*	3000	3000	3000	4900	4400	4900	4900	*	*	*	4400	*
11.60	C-75	7290	*	*	*	*	*	3000	6700	6700	6700	6000	6700	6700	*	*	*	6000	*
13.50	C-75	8460	*	*	*	7490	*	3000	7700	7300	7700	6000	7600	7700	*	7300	*	6000	7600†
11.60	N-80	7780	*	*	*	*	*	3000	7100	7100	7100	6400	7100	7100	*	*	*	6400	*
13.50	N-80	9020	*	*	*	7990	*	3000	8200	7800	8200	6400	8200	8200	*	7800	*	6400	*
11.60	L-80	7780	*	*	*	*	*	3000	7100	7100	7100	6400	7100	7100	*	*	*	6400	*
13.50	L-80	9020	*	*	*	7990	*	3000	8200	7800	8200	6400	7600	8200	*	7800	*	6400	7600†
11.60	C-95	9240	*	*	—	*	—	3000	8400	8400	—	7600	—	8400	*	*	—	7600	—
13.50	C-95	10710	*	*	—	9490	—	3000	9800	9300	—	7600	—	9800	*	9300	—	7600	—
15.10	C-95	12450	*	11630	—	9490	—	3000	10000	9300	—	7600	—	11400	10100	9300	—	7600	—
16.60	C-95	13850	12650	11630	—	9490	—	3000	10000	9300	—	7600	—	12700	10100	9300	—	7600	—
19.10	C-95	16140	12650	11630	—	9490	—	3000	10000	9300	—	7600	—	14800	10100	9300	—	7600	—
11.60	P-110	10690	*	*	*	*	*	3000	9800	9800	9800	8800	9800	9800	*	*	*	8800	*
13.50	P-110	12410	*	*	*	10990	*	3000	10000	10000	11300	8800	11300	11300	*	10800	*	8800	*
15.10	P-110	14420	*	13460	*	10990	*	3000	10000	10000	11700	8800	*	13200	11700	10800	*	8800	12000†
16.60	P-110	16040	14650	13460	*	10990	14980	3000	10000	10000	11700	8800	*	14700	11700	10800	*	8800	12000†
19.10	P-110	18690	14650	13460	18360	10990	14980	3000	10000	10000	11700	8800	*	17100	11700	10800	14700	8800	12000†
15.10	V-150	19660	*	18360	—	14980	—	3000	12000	12000	—	12000	—	18000	16000	14700	—	12000	—
16.60	V-150	21880	19980	18360	—	14980	—	3000	12000	12000	—	12000	—	20000	16000	14700	—	12000	—
19.10	V-150	25490	19980	18360	—	14980	—	3000	12000	12000	—	12000	—	23300	16000	14700	—	12000	—

Fig. 3J-9 (:B) (Courtesy U.S. Steel)

5" OD

Casing Dimensions

Wt/ft (Lbs)		Casing Body				Areas (Sq. in.)	
Nominal T&C	Plain End	Wall (in.)	ID (in.)	Drift (in.)	D/T Ratio	PE and Buttress	Last Perfect Thread (8Rd)
11.50	11.23	.220	4.560	4.435	22.73	3.304	2.200
13.00	12.83	.253	4.494	4.369	19.76	3.773	2.670
15.00	14.87	.296	4.408	4.283	16.89	4.374	3.271
18.00	17.93	.362	4.276	4.151	13.81	5.275	4.171
[1]21.40	21.30	.437	4.126	4.001	11.44	6.264	5.161
[1]24.10	24.03	.500	4.000	3.875	10.00	7.069	5.965
[1]26.70	26.64	.562	3.876	3.751	8.90	7.836	6.732

Coupling Dimensions

OD REG.= 5.563 INCHES
OD SCC= 5.375 INCHES
LENGTH SHORT= 6.500 INCHES
LENGTH LONG= 7.750 INCHES
LENGTH BUTT.= 9.125 INCHES
D1 SHORT= 4.850 INCHES
D1 LONG= 4.811 INCHES
D1 BUTT.= 4.874 INCHES
AREA SHORT= 5.831 SQ.INCHES
AREA LONG= 6.127 SQ.INCHES
AREA BUTTRESS= 5.648 SQ.INCHES
AREA BUTT.SCC= 4.033 SQ.INCHES

Wt/ft	Grade	Collapse		[3]Joint Strength												Plain End Strength	
		[1,2] Feet @ 1.125	Psi	Round													
				Short		Long		Regular				Special Clearance				Yield (Min.)	[1]Ultimate (Avg.)
								Same Grade		Higher Grade		Same Grade		Higher Grade			
				[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	Kips	Kips
11.50	K-55	5440	3060	7100	147											182	354
13.00	K-55	7360	4140	7950	186	8590	201	13210	309	13210	309	13210	309	13210	309	208	404
15.00	K-55	9870	5550	8440	228	9110	246	13300	359	13300	359	13300	359	13300	359	241	468
15.00	C-75	12390	6970			10930	295	13890	375	13890	375	13480	364	13890	375	328	481
18.00	C-75	17780	10000			11600	376	13950	452	13950	452	11230	364	12410	402†	396	580
15.00	N-80	12890	7250			11520	311	14670	396	14670	396	14190	383	14670	396	350	503
18.00	N-80	18650	10490			12220	396	14720	477	14720	477	11820	383	14720	477	422	607
15.00	L-80	12890	7250			10930	295	14040	379	14040	379	13480	364	14040	379	350	481
18.00	L-80	18650	10490			11600	376	14100	457	14100	457	11230	364	12410	402†	422	580
15.00	C-95	14380	8090			12070	326	15700	424	—	—	14890	402	—	—	416	542
18.00	C-95	21350	12010			12840	416	15800	512	—	—	12410	402	—	—	501	654
[1]21.40	C-95	26930	15150			13370	515	14620	563	—	—	10440	402	—	—	595	777
[1]24.10	C-95	30400	17100			13720	595	12980	563	—	—	9270	402	—	—	672	877
[1]26.70	C-95	33710	18960			13980	672	11710	563	—	—	8360	402	—	—	744	972
15.00	P-110	15700	8830			14370	388	18630	503	18630	503	17740	479	18630	503	481	608
18.00	P-110	23910	13450			15280	495	18700	606	18700	606	14780	479	18700	606	580	733
[1]21.40	P-110	31200	17550			15910	613	17420	671	18690	720	12440	479	15910	613†	689	871
[1]24.10	P-110	35200	19800			16320	708	15470	671	18720	812	11040	479	14130	613†	778	983
[1]26.70	P-110	39020	21950			16630	799	13960	671	17850	858	9970	479	12750	613†	862	1089
[1]15.00	V-150	18240	10260			18410	497	24110	651	—	—	22700	613	—	—	656	744
[1]18.00	V-150	29970	16860			19570	634	24230	785	—	—	18920	613	—	—	791	897
[1]21.40	V-150	42540	23930			20350	784	22270	858	—	—	15910	613	—	—	940	1065
[1]24.10	V-150	48000	27000			20910	907	19780	858	—	—	14130	613	—	—	1060	1202
[1]26.70	V-150	53210	29930			21290	1023	17850	858	—	—	12750	613	—	—	1175	1332

Fig. 3J-9 (2A) (Courtesy U S. Steel)

USS Seamless Casing

Maximum Coupling Performance Properties

Grade	Tension (Kips)			Int. Yield Press. (Psi)		
	Round	Buttress		Round	Buttress	
	Long	Regular	SCC	Long	Regular	SCC
K-55	553	510	364	7430	6810	5130
C-75	553	510	364	10140	9290	6990
N-80	582	537	383	10810	9910	7460
L-80	553	510	364	10810	9910	7460
C-95	611	563	402	12840	11770	8850
P-110	728	671	479	14870	13620	10250
V-150	931	858	613	20280	18580	13980

— Not Available

* Same as Round T&C Pipe

† Next Higher Grade Insufficient for Maximum Pressure of Pipe

Wt/ft	Grade	Minimum Internal Yield Pressure						Mill Test Pressure						80% Minimum Yield Strength Also Maximum Formation Fracturing Pressure					
		Pipe Body	Round	Buttress				PE	Round	Buttress				PE	Round	Buttress			
				Regular		Special Clearance				Regular		Special Clearance				Regular		Special Clearance	
				Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade
11.50	K-55	4240	*					3000	3000					3900	*				
13.00	K-55	4870	*	*	*	*	*	3000	3000	3000	4500	4100	4500	4500	*	*	*	4100	*
15.00	K-55	5700	*	*	*	5130	*	3000	3000	3000	5200	4100	5200	5200	*	*	*	4100	*
15.00	C-75	7770	*	*	*	6990	*	3000	7100	7100	7100	5600	7100	7100	*	*	*	5600	*
18.00	C-75	9500	*	9290	*	6990	8850	3000	8100	7400	8100	5600	7100	8700	8100	7400	*	5600	7100†
15.00	N-80	8290	*	*	*	7460	*	3000	7600	7600	7600	6000	7600	7600	*	*	*	6000	*
18.00	N-80	10140	*	9910	*	7460	*	3000	8600	7900	8600	6000	8200	9300	8600	7900	*	6000	8200†
15.00	L-80	8290	*	*	*	7460	*	3000	7600	7600	7600	6000	7100	7600	*	*	*	6000	7100†
18.00	L-80	10140	*	9910	*	7460	8850	3000	8600	7900	8600	6000	7100	9300	8600	7900	*	6000	7100†
15.00	C-95	9840	*	*	—	8850	—	3000	9000	9000	—	7100	—	9000	*	*	—	7100	—
18.00	C-95	12040	*	11770	—	8850	—	3000	10000	9400	—	7100	—	11000	10300	9400	—	7100	—
21.40	C-95	14530	12840	11770	—	8850	—	3000	10000	9400	—	7100	—	13300	10300	9400	—	7100	—
24.10	C-95	16630	12840	11770	—	8850	—	3000	10000	9400	—	7100	—	15200	10300	9400	—	7100	—
26.70	C-95	18690	12840	11770	—	8850	—	3000	10000	9400	—	7100	—	17100	10300	9400	—	7100	—
15.00	P-110	11400	*	*	*	10250	*	3000	10000	10000	10400	8200	10400	10400	*	*	*	8200	*
18.00	P-110	13940	*	13620	*	10250	*	3000	10000	10000	11900	8200	11200	12700	11900	10900	*	8200	11200†
21.40	P-110	16820	14870	13620	*	10250	13980	3000	10000	10000	11900	8200	11200	15400	11900	10900	14900	8200	11200†
24.10	P-110	19250	14870	13620	18580	10250	13980	3000	10000	10000	11900	8200	11200	17600	11900	10900	14900	8200	11200†
26.70	P-110	21640	14870	13620	18580	10250	13980	3000	10000	10000	11900	8200	11200	19800	11900	10900	14900	8200	11200†
15.00	V-150	15540	*	*	—	13980	—	3000	12000	12000	—	11200	—	14200	*	*	—	11200	—
18.00	V-150	19010	*	18580	—	13980	—	3000	12000	12000	—	11200	—	17400	16200	14900	—	11200	—
21.40	V-150	22940	20280	18580	—	13980	—	3000	12000	12000	—	11200	—	21000	16200	14900	—	11200	—
24.10	V-150	26250	20280	18580	—	13980	—	3000	12000	12000	—	11200	—	24000	16200	14900	—	11200	—
26.70	V-150	29510	20280	18580	—	13980	—	3000	12000	12000	—	11200	—	27000	16200	14900	—	11200	—

Fig. 3J-9 (2B) (Courtesy U.S Steel)

5½" OD

Casing Dimensions

Wt/ft (Lbs)		Casing Body				Areas (Sq. In.)	
Nominal T&C	Plain End	Wall (In.)	ID (In.)	Drift (In.)	D/T Ratio	PE and Buttress	Last Perfect Thread (8Rd)
14.00	13.70	.244	5.012	4.887	22.54	4.029	2.814
15.50	15.35	.275	4.950	4.825	20.00	4.514	3.299
17.00	16.87	.304	4.892	4.767	18.09	4.962	3.747
20.00	19.81	.361	4.778	4.653	15.24	5.828	4.613
23.00	22.54	.415	4.670	4.545	13.25	6.630	5.414
‡26.80	26.70	.500	4.500	4.375	11.00	7.854	6.639
‡29.70	29.64	.562	4.376	4.251	9.79	8.718	7.503

Coupling Dimensions

OD REG.= 6.050 INCHES
OD SCC= 5.875 INCHES
LENGTH SHORT= 6.750 INCHES
LENGTH LONG= 8.000 INCHES
LENGTH BUTT.= 9.250 INCHES
D1 SHORT= 5.342 INCHES
D1 LONG= 5.303 INCHES
D1 BUTT.= 5.369 INCHES
AREA SHORT= 6.335 SQ.INCHES
AREA LONG= 6.661 SQ.INCHES
AREA BUTTRESS= 6.108 SQ.INCHES
AREA BUTT.SCC= 4.469 SQ.INCHES

Wt/ft	Grade	Collapse		[3]Joint Strength												Plain End Strength	
		[1,2] Feet @ 1.125	Psi	Round				Buttress								Yield (Min.)	‡Ultimate (Avg.)
				Short		Long		Regular				Special Clearance					
								Same Grade		Higher Grade		Same Grade		Higher Grade			
				‡Ft @ 1.8	Kips	‡Ft @ 1.8	Kips	‡Ft @ 1.8	Kips	‡Ft @ 1.8	Kips	‡Ft @ 1.8	Kips	‡Ft @ 1.8	Kips	Kips	Kips
14.00	H-40	4680	2630	5160	130											161	302
14.00	K-55	5550	3120	7500	189			12980	327	12980	327	12980	327	12980	327	222	431
15.50	K-55	7180	4040	7960	222	8570	239	13120	366	13120	366	13120	366	13120	366	248	483
17.00	K-55	8730	4910	8240	252	8890	272	13140	402	13140	402	13140	402	13140	402	273	531
17.00	C-75	10790	6070			10690	327	13820	423	13820	423	13170	403	13820	423	372	546
20.00	C-75	15000	8440			11190	403	13810	497	13810	497	11190	403	12390	445†	437	641
23.00	C-75	18600	10460			11430	473	13310	551	13650	565	9730	403	10770	446†	497	729
17.00	N-80	11160	6280			11370	348	14580	446	14580	446	13890	425	14580	446	397	571
20.00	N-80	15700	8830			11890	428	14560	524	14560	524	11810	425	14560	524	466	670
23.00	N-80	19840	11150			12130	502	14010	580	14400	596	10270	425	12830	531†	530	762
17.00	L-80	11160	6280			11050	338	13990	428	13990	428	13170	403	13990	428	397	546
20.00	L-80	15700	8830			11560	416	13970	503	13970	503	11190	403	12390	446†	466	641
23.00	L-80	19840	11160			11810	489	13310	551	13820	572	9730	403	10770	446†	530	729
17.00	C-95	12320	6930			12220	374	15690	480	—	—	14580	446	—	—	471	615
20.00	C-95	17780	10000			12780	460	15640	563	—	—	12390	446	—	—	554	723
23.00	C-95	22970	12920			13040	540	14710	609	—	—	10770	446	—	—	630	822
‡26.80	C-95	27910	15700			13720	662	12620	609	—	—	9250	446	—	—	746	974
‡29.70	C-95	30990	17430			13990	748	11390	609	—	—	8340	446	—	—	828	1081
17.00	P-110	13260	7460			14540	445	18560	568	18560	568	17350	531	18560	568	546	690
20.00	P-110	19700	11080			15220	548	18530	667	18530	667	14750	531	18530	667	641	810
23.00	P-110	25810	14520			15530	643	17510	725	18330	759	12830	531	16400	679†	729	922
‡26.80	P-110	32320	18180			16330	788	15030	725	18640	899	11010	531	14080	679†	864	1092
‡29.70	P-110	35880	20180			16670	891	13560	725	17360	928	9930	531	12700	679†	959	1212
‡20.00	V-150	23960	13480			19470	701	24030	865	—	—	18860	679	—	—	874	991
‡23.00	V-150	32690	18390			19880	823	22420	928	—	—	16400	679	—	—	995	1127
‡26.80	V-150	44070	24790			20920	1009	19240	928	—	—	14080	679	—	—	1178	1335
‡29.70	V-150	48920	27520			21320	1140	17360	928	—	—	12700	679	—	—	1308	1482

Fig. 3J-9 (3A) (Courtesy U.S. S eel)

USS Seamless Casing

Maximum Coupling Performance Properties

Grade	Tension (Kips)			Int. Yield Press. (Psi)		
	Round	Buttress		Round	Buttress	
	Long	Regular	SCC	Long	Regular	SCC
K-55	601	551	403	6790	6190	4740
C-75	601	551	403	9260	8440	6460
N-80	633	580	425	9880	9000	6890
L-80	601	551	403	9880	9000	6890
C-95	664	609	446	11730	10690	8180
P-110	791	725	531	13580	12380	9470
† V-150	1012	928	679	18520	16880	12920

— Not Available

* Same as Round T&C Pipe

† Next Higher Grade Insufficient for Maximum Pressure of Pipe

Wt/ft	Grade	Minimum Internal Yield Pressure						Mill Test Pressure						80% Minimum Yield Strength Also Maximum Formation Fracturing Pressure					
		Pipe Body	Round	Buttress				PE	Round	Buttress				PE	Round	Buttress			
				Regular		Special Clearance				Regular		Special Clearance				Regular		Special Clearance	
				Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade
14.00	H-40	3110	*					2800	2800					2800	*				
14.00	K-55	4270	*	*	*	*	*	3000	3000	3000	3900	3800	3900	3900	*	*	*	3800	*
15.50	K-55	4810	*	*	*	4740	*	3000	3000	3000	4400	3800	4400	4400	*	*	*	3800	*
17.00	K-55	5320	*	*	*	4740	*	3000	3000	3000	4900	3800	4900	4900	*	*	*	3800	*
17.00	C-75	7250	*	*	*	6460	*	3000	6600	6600	6600	5200	6500	6600	*	*	*	5200	65001
20.00	C-75	8610	*	8440	*	6460	8180	3000	7400	6800	7400	5200	6500	7900	7400	6800	*	5200	65001
23.00	C-75	9900	9260	8440	*	6460	8180	3000	7400	6800	7400	5200	6500	9100	7400	6800	8600	5200	65001
17.00	N-80	7740	*	*	*	6890	*	3000	7100	7100	7100	5500	7100	7100	*	*	*	5500	*
20.00	N-80	9190	*	9000	*	6890	*	3000	7900	7200	7900	5500	7600	8400	7900	7200	*	5500	76001
23.00	N-80	10560	9880	9000	*	6890	9470	3000	7900	7200	7900	5500	7600	9700	7900	7200	*	5500	76001
17.00	L-80	7740	*	*	*	6890	*	3000	7100	7100	7100	5500	6500	7100	*	*	*	5500	65001
20.00	L-80	9190	*	9000	*	6890	8180	3000	7900	7200	7900	5500	6500	8400	7900	7200	*	5500	65001
23.00	L-80	10560	9880	9000	*	6890	8180	3000	7900	7200	7900	5500	6500	9700	7900	7200	8600	5500	65001
17.00	C-95	9190	*	*	—	8180	—	3000	8400	8400	—	6500	—	8400	*	*	—	6500	—
20.00	C-95	10910	*	10690	—	8180	—	3000	9400	8600	—	6500	—	10000	9400	8600	—	6500	—
23.00	C-95	12540	11730	10690	—	8180	—	3000	9400	8600	—	6500	—	11500	9400	8600	—	6500	—
26.80	C-95	15110	11730	10690	—	8180	—	3000	9400	8600	—	6500	—	13800	9400	8600	—	6500	—
29.70	C-95	16990	11730	10690	—	8180	—	3000	9400	8600	—	6500	—	15500	9400	8600	—	6500	—
17.00	P-110	10640	*	*	*	9470	*	3000	9700	9700	9700	7600	9700	9700	*	*	*	7600	*
20.00	P-110	12640	*	12380	*	9470	*	3000	10000	9900	10900	7600	10300	11600	10900	9900	*	7600	103001
23.00	P-110	14530	13580	12380	*	9470	12920	3000	10000	9900	10900	7600	10300	13300	10900	9900	*	7600	103001
26.80	P-110	17500	13580	12380	16880	9470	12920	3000	10000	9900	10900	7600	10300	16000	10900	9900	13500	7600	103001
29.70	P-110	19670	13580	12380	16880	9470	12920	3000	10000	9900	10900	7600	10300	18000	10900	9900	13500	7600	103001
20.00	V-150	17230	*	16880	—	12920	—	3000	12000	12000	—	10300	—	15800	14800	13500	—	10300	—
23.00	V-150	19810	18520	16880	—	12920	—	3000	12000	12000	—	10300	—	18100	14800	13500	—	10300	—
26.80	V-150	23860	18520	16880	—	12920	—	3000	12000	12000	—	10300	—	21800	14800	13500	—	10300	—
29.70	V-150	26820	18520	16880	—	12920	—	3000	12000	12000	—	10300	—	24500	14800	13500	—	10300	—

Fig. 3J-9 (3B) (Courtesy U.S. Steel)

6⅝" OD

Casing Dimensions

Wt/ft (Lbs)		Casing Body				Areas (Sq. In.)	
Nominal T&C	Plain End	Wall (In.)	ID (In.)	Drift (In.)	D/T Ratio	PE and Buttress	Last Perfect Thread (8Rd)
20.00	19.49	.288	6.049	5.924	23.00	5.734	4.267
24.00	23.58	.352	5.921	5.796	18.82	6.937	5.470
28.00	27.65	.417	5.791	5.666	15.89	8.133	6.666
32.00	31.20	.475	5.675	5.550	13.95	9.177	7.710

Coupling Dimensions

OD REG.= 7.390 INCHES
OD SCC= 7.000 INCHES
LENGTH SHORT= 7.250 INCHES
LENGTH LONG= 8.750 INCHES
LENGTH BUTT.= 9.625 INCHES
D1 SHORT= 6.451 INCHES
D1 LONG= 6.404 INCHES
D1 BUTT.= 6.483 INCHES
AREA SHORT=10.208 SQ.INCHES
AREA LONG=10.682 SQ.INCHES
AREA BUTTRESS= 9.883 SQ.INCHES
AREA BUTT.SCC= 5.475 SQ.INCHES

Wt/ft	Grade	Collapse		[3]Joint Strength												Plain End Strength	
		[1,2] Feet @ 1.125	Psi	Round				Buttress								Yield (Min.)	[4]Ultimate (Avg.)
				Short		Long		Regular				Special Clearance					
								Same Grade		Higher Grade		Same Grade		Higher Grade			
				[4]Ft @ 1.8	Kips	[4]Ft @ 1.8	Kips	[4]Ft @ 1.8	Kips	[4]Ft @ 1.8	Kips	[4]Ft @ 1.8	Kips	[4]Ft @ 1.8	Kips	Kips	Kips
20.00	H-40	4480	2520	5110	184											229	430
20.00	K-55	5280	2970	7420	267	8060	290	12580	453	12580	453	12580	453	12580	453	315	614
24.00	K-55	8110	4560	7920	342	8610	372	12690	548	12690	548	11440	494	12040	520†	382	742
24.00	C-75	9900	5570			10490	453	13500	583	13500	583	11440	494	12640	546†	520	763
28.00	C-75	13920	7830			10950	552	13550	683	13550	683	9800	494	10830	546†	610	895
32.00	C-75	17480	9830			11080	638	13390	771	13390	771	8580	494	9480	546†	688	1009
24.00	N-80	10240	5760			11130	481	14240	615	14240	615	12040	520	14240	615	555	798
28.00	N-80	14520	8170			11630	586	14310	721	14310	721	10320	520	12900	650†	651	935
32.00	N-80	18350	10320			11750	677	14130	814	14130	814	9030	520	11280	650†	734	1055
24.00	L-80	10240	5750			10950	473	13700	592	13700	592	11440	494	12640	546†	555	763
28.00	L-80	14520	8170			11430	576	13750	693	13750	693	9800	494	10830	546†	651	895
32.00	L-80	18350	10320			11560	666	13590	783	13590	783	8580	494	9480	546†	734	1009
24.00	C-95	11180	6290			12640	546	15390	665	—	—	12640	546	—	—	659	860
28.00	C-95	16360	9200			13190	665	15480	780	—	—	10830	546	—	—	773	1009
32.00	C-95	20980	11800			13350	769	15280	880	—	—	9480	546	—	—	872	1138
24.00	P-110	11930	6710			14840	641	18190	786	18190	786	15050	650	18190	786	763	964
28.00	P-110	18030	10140			15500	781	18290	922	18290	922	12900	650	16510	832†	895	1130
32.00	P-110	23480	13210			15690	904	18060	1040	18060	1040	11280	650	14440	832†	1009	1276

Fig. 3J-9 (4A) (Courtesy U.S. Steel)

USS Seamless Casing

Maximum Coupling Performance Properties

Grade	Tension (Kips)			Int. Yield Press. (Psi)		
	Round	Buttress		Round	Buttress	
	Long	Regular	SCC	Long	Regular	SCC
K-55	964	892	494	7340	6750	4060
C-75	964	892	494	10010	9210	5540
N-80	1015	939	520	10670	9820	5910
L-80	964	892	494	10670	9820	5910
C-95	1066	986	546	12680	11660	7020
P-110	1268	1174	650	14680	13500	8120
† V-150	1624	1502	832	20010	18410	11080

— Not Available

* Same as Round T&C Pipe

† Next Higher Grade Insufficient for Maximum Pressure of Pipe

Wt/ft	Grade	Minimum Internal Yield Pressure						Mill Test Pressure						80% Minimum Yield Strength Also Maximum Formation Fracturing Pressure					
		Pipe Body	Round	Buttress				PE	Round	Buttress				PE	Round	Buttress			
				Regular		Special Clearance				Regular		Special Clearance				Regular		Special Clearance	
				Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade
20.00	H-40	3040	*					2800	2800					2800	*				
20.00	K-55	4180	*	*	*	4060	*	3000	3000	3000	3800	3200	3800	3800	*	*	*	3200	*
24.00	K-55	5110	*	*	*	4060	*	3000	3000	3000	4700	3200	4700	4700	*	*	*	3200	*
24.00	C-75	6970	*	*	*	5540	*	3000	6400	6400	6400	4400	5600	6400	*	*	*	4400	5600†
28.00	C-75	8260	*	*	*	5540	7020	3000	7600	7400	7600	4400	5600	7600	*	7400	*	4400	5600†
32.00	C-75	9410	*	9210	*	5540	7020	3000	8000	7400	8000	4400	5600	8600	8000	7400	*	4400	5600†
24.00	N-80	7440	*	*	*	5910	*	3000	6800	6800	6800	4700	6500	6800	*	*	*	4700	6500†
28.00	N-80	8810	*	*	*	5910	8120	3000	8100	7900	8100	4700	6500	8100	*	7900	*	4700	6500†
32.00	N-80	10040	*	9820	*	5910	8120	3000	8500	7900	8500	4700	6500	9200	8500	7900	*	4700	6500†
24.00	L-80	7440	*	*	*	5910	7020	3000	6800	6800	6800	4700	5600	6800	*	*	*	4700	5600†
28.00	L-80	8810	*	*	*	5910	7020	3000	8100	7900	8100	4700	5600	8100	*	7900	*	4700	5600†
32.00	L-80	10040	*	9820	*	5910	7020	3000	8500	7900	8500	4700	5600	9200	8500	7900	*	4700	5600†
24.00	C-95	8830	*	*	—	7020	—	3000	8100	8100	—	5600	—	8100	*	*	—	5600	—
28.00	C-95	10460	*	*	—	7020	—	3000	9600	9300	—	5600	—	9600	*	9300	—	5600	—
32.00	C-95	11920	*	11660	—	7020	—	3000	10000	9300	—	5600	—	10900	10100	9300	—	5600	—
24.00	P-110	10230	*	*	*	8120	*	3000	9400	9400	9400	6500	8900	9400	*	*	*	6500	8900†
28.00	P-110	12120	*	*	*	8120	11080	3000	10000	10000	11100	6500	8900	11100	*	10800	*	6500	8900†
32.00	P-110	13800	*	13500	*	8120	11080	3000	10000	10000	11700	6500	8900	12600	11700	10800	*	6500	8900†

Fig. 3J-9 (4B) (Courtesy U.S. Steel)

7" OD See pages 18 and 19 for additional 7" OD grades.

Casing Dimensions

Wt/ft (Lbs)		Casing Body				Areas (Sq. In.)	
Nominal T&C	Plain End	Wall (In.)	ID (In.)	Drift (In.)	D/T Ratio	PE and Buttress	Last Perfect Thread (8Rd)
17.00	16.70	.231	6.538	6.413	30.30	4.912	3.361
20.00	19.54	.272	6.456	6.331	25.74	5.749	4.198
23.00	22.63	.317	6.366	6.241	22.08	6.656	5.105
26.00	25.66	.362	6.276	6.151	19.34	7.549	5.998
29.00	28.72	.408	6.184	6.059	17.16	8.449	6.899
32.00	31.68	.453	6.094	5.969	15.45	9.317	7.766
35.00	34.58	.498	6.004	5.879	14.06	10.172	8.622
38.00	37.26	.540	5.920	5.795	12.96	10.959	9.408
[1]42.70	42.55	.625	5.750	5.625	11.20	12.517	10.966

Coupling Dimensions

OD REG.= 7.656 INCHES
OD SCC= 7.375 INCHES
LENGTH SHORT= 7.250 INCHES
LENGTH LONG= 9.000 INCHES
LENGTH BUTT.= 9.625 INCHES
D1 SHORT= 6.826 INCHES
D1 LONG= 6.772 INCHES
D1 BUTT.= 6.846 INCHES
AREA SHORT= 9.441 SQ.INCHES
AREA LONG=10.017 SQ.INCHES
AREA BUTTRESS= 9.226 SQ.INCHES
AREA BUTT.SCC= 5.908 SQ.INCHES

Wt/ft	Grade	Collapse		[3]Joint Strength												Plain End Strength	
		[1,2] Feet @ 1.125	Psi	Round				Buttress								Yield (Min.)	[1]Ultimate (Avg.)
				Short		Long		Regular				Special Clearance					
								Same Grade		Higher Grade		Same Grade		Higher Grade			
				[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	Kips	Kips
17.00	H-40	2580	1450	3990	122											196	363
20.00	H-40	3520	1980	4890	176											230	431
20.00	K-55	4040	2270	7060	254	7810	281	12530	451	12530	451	12530	451.	12530	451	316	615
23.00	K-55	5810	3270	7460	309	8240	341	12610	522	12610	522	12610	522	12610	522	366	712
26.00	K-55	7680	4320	7780	364	8570	401	12650	592	12650	592	11390	533	11990	561†	415	803
23.00	N-80	6810	3830			10680	442	14200	588	14200	588	13550	561	14200	588	532	765
26.00	N-80	9620	5410			11090	519	14250	667	14250	667	11990	561	14250	667	604	868
29.00	N-80	12480	7020			11440	597	14290	746	14290	746	10750	561	13450	702†	676	972
32.00	N-80	15290	8600			11670	672	14290	823	14290	823	9740	561	12190	702†	745	1071
35.00	N-80	18100	10130			11840	746	13900	876	14250	898	8900	561	11140	702†	814	1170
38.00	N-80	20250	11390			11900	814	12810	876	14150	968	8200	561	10260	702†	877	1260
[1]23.00	USS95	10040	5650			12370	512	15920	659	15920	659	14900	617	15920	659	632	845
[1]26.00	USS95	13870	7800			12860	602	15960	747	15960	747	13180	617	15000	702†	717	959
[1]29.00	USS95	16360	9200			13260	692	16020	836	16020	836	11820	617	13450	702†	803	1073
26.00	P-110	11040	6210			14810	693	18230	853	18230	853	15000	702	18230	853	830	1049
29.00	P-110	15130	8510			15270	797	18300	955	18300	955	13450	702	17200	898†	929	1174
32.00	P-110	19130	10760			15570	897	18280	1053	18280	1053	12190	702	15590	898†	1025	1295
35.00	P-110	23130	13010			15810	996	17400	1096	18250	1150	11140	702	14250	898†	1119	1414
38.00	P-110	26860	15110			15890	1087	16020	1096	18110	1239	10260	702	13130	898†	1205	1523
[1]42.70	P-110	31800	17890			16480	1267	14260	1096	18240	1402	9130	702	11680	898†	1377	1740
[1]29.00	V-150	17420	9800			20100	1049	23810	1243	—	—	17200	898	—	—	1267	1435
[1]32.00	V-150	23150	13020			20490	1180	23780	1370	—	—	15590	898	—	—	1398	1584
[1]35.00	V-150	28850	16230			20810	1311	22250	1402	—	—	14250	898	—	—	1526	1729
[1]38.00	V-150	34200	19240			20910	1430	20500	1402	—	—	13130	898	—	—	1644	1863
[1]42.70	V-150	43360	24390			21690	1667	18240	1402	—	—	11680	898	—	—	1878	2128

Fig. 3J-9 (5A) (Courtesy U.S. Steel)

USS Seamless Casing

Maximum Coupling Performance Properties

Grade	Tension (Kips)			Int. Yield Press. (Psi)		
	Round	Buttress		Round	Buttress	
	Long	Regular	SCC	Long	Regular	SCC
K-55	904	833	533	6350	5820	3950
C-75	904	833	533	8660	7930	5380
N-80	952	876	561	9240	8460	5740
L-80	904	833	533	9240	8460	5740
C-95	999	920	589	10970	10050	6810
USS95	1047	964	617	10970	10050	6810
P-110	1190	1096	702	12700	11640	7890
V-150	1523	1402	898	17320	15870	10760

— Not Available

• Same as Round T&C Pipe

† Next Higher Grade Insufficient for Maximum Pressure of Pipe

Wt/ft	Grade	Minimum Internal Yield Pressure						Mill Test Pressure						80% Minimum Yield Strength Also Maximum Formation Fracturing Pressure					
		Pipe Body	Round	Buttress				PE	Round	Buttress				PE	Round	Buttress			
				Regular		Special Clearance				Regular		Special Clearance				Regular		Special Clearance	
				Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade
17.00	H-40	2310	•					2100	2100					2100	•				
20.00	H-40	2720	•					2500	2500					2500	•				
20.00	K-55	3740	•	•	•	•	•	3000	3000	3000	3400	3200	3400	3400	•	•	•	3200	•
23.00	K-55	4360	•	•	•	3950	•	3000	3000	3000	4000	3200	4000	4000	•	•	•	3200	•
26.00	K-55	4980	•	•	•	3950	•	3000	3000	3000	4600	3200	4600	4600	•	•	•	3200	•
23.00	N-80	6340	•	•	•	5740	•	3000	5800	5800	5800	4600	5800	5800	•	•	•	4600	•
26.00	N-80	7240	•	•	•	5740	•	3000	6600	6600	6600	4600	6300	6600	•	•	•	4600	6300†
29.00	N-80	8160	•	•	•	5740	7890	3000	7400	6800	7400	4600	6300	7500	7400	6800	•	4600	6300†
32.00	N-80	9060	•	8460	•	5740	7890	3000	7400	6800	7400	4600	6300	8300	7400	6800	•	4600	6300†
35.00	N-80	9960	9240	8460	•	5740	7890	3000	7400	6800	7400	4600	6300	9100	7400	6800	•	4600	6300†
38.00	N-80	10800	9240	8460	•	5740	7890	3000	7400	6800	7400	4600	6300	9900	7400	6800	9300	4600	6300†
23.00	USS95	7530	•	•	•	6810	•	3000	6900	6900	6900	5500	6300	6900	•	•	•	5500	6300†
26.00	USS95	8600	•	•	•	6810	7890	3000	7900	7900	7900	5500	6300	7900	•	•	•	5500	6300†
29.00	USS95	9690	•	•	•	6810	7890	3000	8800	8000	8800	5500	6300	8900	8800	8000	•	5500	6300†
26.00	P-110	9960	•	•	•	7890	•	3000	9100	9100	9100	6300	8600	9100	•	•	•	6300	8600†
29.00	P-110	11220	•	•	•	7890	10760	3000	10000	9300	10200	6300	8600	10300	10200	9300	•	6300	8600†
32.00	P-110	12460	•	11640	•	7890	10760	3000	10000	9300	10200	6300	8600	11400	10200	9300	•	6300	8600†
35.00	P-110	13700	12700	11640	•	7890	10760	3000	10000	9300	10200	6300	8600	12500	10200	9300	•	6300	8600†
38.00	P-110	14850	12700	11640	•	7890	10760	3000	10000	9300	10200	6300	8600	13600	10200	9300	12700	6300	8600†
42.70	P-110	17190	12700	11640	15870	7890	10760	3000	10000	9300	10200	6300	8600	15700	10200	9300	12700	6300	8600†
29.00	V-150	15300	•	•	—	10760	—	3000	12000	12000	—	8600	—	14000	13900	12700	—	8600	—
32.00	V-150	16990	•	15870	—	10760	—	3000	12000	12000	—	8600	—	15500	13900	12700	—	8600	—
35.00	V-150	18680	17320	15870	—	10760	—	3000	12000	12000	—	8600	—	17100	13900	12700	—	8600	—
38.00	V-150	20250	17320	15870	—	10760	—	3000	12000	12000	—	8600	—	18500	13900	12700	—	8600	—
42.70	V-150	23440	17320	15870	—	10760	—	3000	12000	12000	—	8600	—	21400	13900	12700	—	8600	—

Fig. 3J-9 (5B) (Courtesy U.S. Steel)

7" OD See pages 16 and 17 for additional 7" OD grades.

Casing Dimensions

Wt/ft (Lbs)		Casing Body				Areas (Sq. In.)	
Nominal T&C	Plain End	Wall (In.)	ID (In.)	Drift (In.)	D/T Ratio	PE and Buttress	Last Perfect Thread (8Rd)
17.00	16.70	.231	6.538	6.413	30.30	4.912	3.361
20.00	19.54	.272	6.456	6.331	25.74	5.749	4.198
23.00	22.63	.317	6.366	6.241	22.08	6.656	5.105
26.00	25.66	.362	6.276	6.151	19.34	7.549	5.998
29.00	28.72	.408	6.184	6.059	17.16	8.449	6.899
32.00	31.68	.453	6.094	5.969	15.45	9.317	7.766
35.00	34.58	.498	6.004	5.879	14.06	10.172	8.622
38.00	37.26	.540	5.920	5.795	12.96	10.959	9.408
[1]42.70	42.55	.625	5.750	5.625	11.20	12.517	10.966

Coupling Dimensions

OD REG.= 7.656 INCHES
OD SCC= 7.375 INCHES
LENGTH SHORT= 7.250 INCHES
LENGTH LONG= 9.000 INCHES
LENGTH BUTT.=10.000 INCHES
D1 SHORT= 6.826 INCHES
D1 LONG= 6.772 INCHES
D1 BUTT.= 6.846 INCHES
AREA SHORT= 9.441 SQ.INCHES
AREA LONG=10.017 SQ.INCHES
AREA BUTTRESS= 9.226 SQ.INCHES
AREA BUTT.SCC= 5.908 SQ.INCHES

Wt/ft	Grade	Collapse		[3]Joint Strength												Plain End Strength	
		[1,2] Feet @ 1.125	Psi	Round				Buttress								Yield (Min.)	[1]Ultimate (Avg.)
				Short		Long		Regular				Special Clearance					
								Same Grade		Higher Grade		Same Grade		Higher Grade			
				[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	Kips	Kips
17.00	H-40	2580	1450	3990	122											196	368
20.00	H-40	3520	1980	4890	176											230	431
20.00	K-55	4040	2270	7060	254	7810	281	12530	451	12530	451	12530	451	12530	451	316	615
23.00	K-55	5810	3270	7460	309	8240	341	12610	522	12610	522	12610	522	12610	522	366	712
26.00	K-55	7680	4320	7780	364	8570	401	12650	592	12650	592	11390	533	11990	561†	415	808
23.00	C-75	6700	3770			10050	416	13450	557	13450	557	12870	533	13450	557	499	732
26.00	C-75	9330	5250			10450	489	13480	631	13480	631	11390	533	12590	589†	566	830
29.00	C-75	12020	6760			10770	562	13540	707	13540	707	10210	533	11280	589†	634	929
32.00	C-75	14630	8230			10990	633	13520	779	13520	779	9250	533	10230	589†	699	1025
35.00	C-75	17260	9710			11160	703	13220	833	13510	851	8460	533	9350	589†	763	1119
38.00	C-75	18990	10680			11210	767	12180	833	13390	916	7790	533	8610	589†	822	1205
23.00	L-80	6810	3830			10510	435	13650	565	13650	565	12870	533	13650	565	532	732
26.00	L-80	9620	5410			10920	511	13700	641	13700	641	11390	533	12590	589†	604	830
29.00	L-80	12480	7020			11250	587	13750	718	13750	718	10210	533	11280	589†	676	929
32.00	L-80	15290	8600			11480	661	13730	791	13730	791	9250	533	10230	589†	745	1025
35.00	L-80	18100	10180			11650	734	13220	833	13710	864	8460	533	9350	589†	814	1119
38.00	L-80	20250	11390			11710	801	12180	833	13450	920	7790	533	8610	589†	877	1205
23.00	C-95	7380	4150			12200	505	15360	636	—	—	14230	589	—	—	632	825
26.00	C-95	10440	5870			12670	593	15430	722	—	—	12590	589	—	—	717	936
29.00	C-95	13900	7820			13080	683	15480	808	—	—	11280	589	—	—	803	1048
32.00	C-95	17300	9730			13330	768	15470	891	—	—	10230	589	—	—	885	1155
35.00	C-95	20690	11640			13540	853	14600	920	—	—	9350	589	—	—	966	1261
38.00	C-95	23860	13420			13610	931	13450	920	—	—	8610	589	—	—	1041	1359
[1]42.70	C-95	27470	15450			14120	1085	11970	920	—	—	7660	589	—	—	1189	1552

Fig. 3J-9 (6A) (Courtesy U.S. Steel)

USS Seamless Casing

Maximum Coupling Performance Properties

Grade	Tension (Kips)			Int. Yield Press. (Psi)		
	Round	Buttress		Round	Buttress	
	Long	Regular	SCC	Long	Regular	SCC
K-55	904	833	533	6350	5820	3950
C-75	904	833	533	8660	7930	5380
N-80	952	876	561	9240	8460	5740
L-80	904	833	533	9240	8460	5740
C-95	999	920	589	10970	10050	6810
‡USS95	1047	964	617	10970	10050	6810
P-110	1190	1096	702	12700	11640	7890
‡ V-150	1523	1402	898	17320	15870	10760

— Not Available

* Same as Round T&C Pipe

‡ Next Higher Grade Insufficient for Maximum Pressure of Pipe

Wt/ft	Grade	Minimum Internal Yield Pressure						Mill Test Pressure						80% Minimum Yield Strength Also Maximum Formation Fracturing Pressure					
		Pipe Body	Round	Buttress				PE	Round	Buttress				PE	Round	Buttress			
				Regular		Special Clearance				Regular		Special Clearance				Regular		Special Clearance	
				Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade
17.00	H-40	2310	*					2100	2100					2100	*				
20.00	H-40	2720	*					2500	2500					2500	*				
20.00	K-55	3740	*	*	*	*	*	3000	3000	3000	3400	3200	3400	3400	*	*	*	3200	*
23.00	K-55	4360	*	*	*	3950	*	3000	3000	3000	4000	3200	4000	4000	*	*	*	3200	*
26.00	K-55	4980	*	*	*	3950	*	3000	3000	3000	4600	3200	4600	4600	*	*	*	3200	*
23.00	C-75	5940	*	*	*	5380	*	3000	5400	5400	5400	4300	5400	5400	*	*	*	4300	*
26.00	C-75	6790	*	*	*	5380	*	3000	6200	6200	6200	4300	5500	6200	*	*	*	4300	55001
29.00	C-75	7650	*	*	*	5380	6810	3000	6900	6300	6900	4300	5500	7000	6900	6300	*	4300	55001
32.00	C-75	8490	*	7930	*	5380	6810	3000	6900	6300	6900	4300	5500	7800	6900	6300	*	4300	55001
35.00	C-75	9340	8660	7930	*	5380	6810	3000	6900	6300	6900	4300	5500	8500	6900	6300	8000	4300	55001
38.00	C-75	10130	8660	7930	10050	5380	6810	3000	6900	6300	6900	4300	5500	9300	6900	6300	8000	4300	55001
23.00	L-80	6340	*	*	*	5740	*	3000	5800	5800	5800	4600	5500	5800	*	*	*	4600	55001
26.00	L-80	7240	*	*	*	5740	6810	3000	6600	6600	6600	4600	5500	6600	*	*	*	4600	55001
29.00	L-80	8160	*	*	*	5740	6810	3000	7400	6800	7400	4600	5500	7500	7400	6800	*	4600	55001
32.00	L-80	9060	*	8460	*	5740	6810	3000	7400	6800	7400	4600	5500	8300	7400	6800	8000	4600	55001
35.00	L-80	9960	9240	8460	*	5740	6810	3000	7400	6800	7400	4600	5500	9100	7400	6800	8000	4600	55001
38.00	L-80	10800	9240	8460	10050	5740	6810	3000	7400	6800	7400	4600	5500	9900	7400	6800	8000	4600	55001
23.00	C-95	7530	*	*	—	6810	—	3000	6900	6900	—	5500	—	6900	*	*	—	5500	—
26.00	C-95	8600	*	*	—	6810	—	3000	7900	7900	—	5500	—	7900	*	*	—	5500	—
29.00	C-95	9690	*	*	—	6810	—	3000	8800	8000	—	5500	—	8900	8800	8000	—	5500	—
32.00	C-95	10760	*	10050	—	6810	—	3000	8800	8000	—	5500	—	9800	8800	8000	—	5500	—
35.00	C-95	11830	10970	10050	—	6810	—	3000	8800	8000	—	5500	—	10800	8800	8000	—	5500	—
38.00	C-95	12830	10970	10050	—	6810	—	3000	8800	8000	—	5500	—	11700	8800	8000	—	5500	—
42.70	C-95	14840	10970	10050	—	6810	—	3000	8800	8000	—	5500	—	13600	8800	8000	—	5500	—

Fig. 3J-9 (6B) (Courtesy U.S. Steel)

7⅝" OD

Casing Dimensions

Wt/ft (Lbs)		Casing Body				Areas (Sq. In.)	
Nominal T&C	Plain End	Wall (In.)	ID (In.)	Drift (In.)	D/T Ratio	PE and Buttress	Last Perfect Thread (8Rd)
20.00	19.69	.250	7.125	7.000	30.50	5.792	4.102
24.00	23.47	.300	7.025	6.900	25.42	6.904	5.213
26.40	25.56	.328	6.969	6.844	23.25	7.519	5.828
29.70	29.04	.375	6.875	6.750	20.33	8.541	6.850
33.70	33.04	.430	6.765	6.640	17.73	9.720	8.029
39.00	38.05	.500	6.625	6.500	15.25	11.192	9.501
[1]42.80	42.39	.562	6.501	6.376	13.57	12.470	10.779
[1]47.10	46.73	.625	6.375	6.250	12.20	13.745	12.054
[1]51.20	50.91	.687	6.251	6.126	11.10	14.974	13.283

Coupling Dimensions

OD REG.= 8.500 INCHES
OD SCC= 8.125 INCHES
LENGTH SHORT= 7.500 INCHES
LENGTH LONG= 9.250 INCHES
LENGTH BUTT.=10.375 INCHES
D1 SHORT= 7.440 INCHES
D1 LONG= 7.385 INCHES
D1 BUTT.= 7.460 INCHES
AREA SHORT=13.270 SQ.INCHES
AREA LONG=13.911 SQ.INCHES
AREA BUTTRESS=13.036 SQ.INCHES
AREA BUTT.SCC= 8.140 SQ.INCHES

Wt/ft	Grade	Collapse		[3]Joint Strength												Plain End Strength	
		[1,2] Feet @ 1.125	Psi	Round				Buttress									
				Short		Long		Regular				Special Clearance				Yield (Min.)	[1]Ultimate (Avg.)
								Same Grade		Higher Grade		Same Grade		Higher Grade			
				[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	Kips	Kips
[1]20.00	H-40	2540	1430	4610	166											232	434
24.00	H-40	3630	2040	4910	212											276	518
26.40	K-55	5140	2890	7200	342	7930	377	12230	581	12230	581	12230	581	12230	581	414	805
26.40	C-75	5830	3280			9700	461	13130	624	13130	624	13130	624	13130	624	564	827
29.70	C-75	8300	4670			10140	542	13260	709	13260	709	13260	709	13260	709	641	940
33.70	C-75	11240	6320			10470	635	13290	806	13290	806	12120	735	13290	806	729	1069
39.00	C-75	14990	8430			10700	751	13230	929	13230	929	10470	735	11570	812†	839	1231
26.40	N-80	6040	3400			10310	490	13870	659	13870	659	13870	659	13870	659	602	865
29.70	N-80	8520	4790			10760	575	14010	749	14010	749	14010	749	14010	749	683	982
33.70	N-80	11660	6560			11110	674	14050	852	14050	852	12740	773	14050	852	778	1118
39.00	N-80	15660	8810			11370	798	13970	981	13970	981	11010	773	13770	967†	895	1287
26.40	L-80	6040	3400			10140	482	13360	635	13360	635	13360	635	13360	635	602	827
29.70	L-80	8520	4790			10590	566	13490	721	13490	721	13490	721	13490	721	683	940
33.70	L-80	11660	6560			10950	664	13520	820	13520	820	12120	735	13390	812†	778	1069
39.00	L-80	15660	8810			11200	786	13460	945	13460	945	10470	735	11570	812†	895	1231
26.40	C-95	6600	3710			11780	560	15070	716	—	—	15070	716	—	—	714	932
29.70	C-95	9100	5120			12330	659	15210	813	—	—	15190	812	—	—	811	1059
33.70	C-95	12910	7260			12730	772	15250	925	—	—	13390	812	—	—	923	1205
39.00	C-95	17740	9980			13020	914	15170	1065	—	—	11570	812	—	—	1063	1388
[1]42.80	C-95	22040	12400			13460	1037	15410	1187	—	—	10540	812	—	—	1185	1546
[1]47.10	C-95	25420	14300			13670	1159	15330	1300	—	—	9580	812	—	—	1306	1704
[1]51.20	C-95	27700	15580			13860	1277	14110	1300	—	—	8810	812	—	—	1423	1857
[1]26.40	USS95	8620	4850			11950	568	15570	740	15570	740	15570	740	15570	740	714	955
[1]29.70	USS95	12710	7150			12500	668	15730	841	15730	841	15730	841	15730	841	811	1085
[1]33.70	USS95	15640	8800			12910	783	15780	957	15780	957	14030	851	15780	957	923	1234
26.40	P-110	6990	3930			13760	654	17780	845	17780	845	17780	845	17780	845	827	1045
29.70	P-110	9490	5340			14380	769	17960	960	17960	960	17960	960	17960	960	940	1187
33.70	P-110	13960	7850			14850	901	18020	1093	18020	1093	15940	967	18020	1093	1069	1351
39.00	P-110	19660	11060			15190	1066	17920	1258	17920	1258	13770	967	17620	1237†	1231	1556
[1]42.80	P-110	24730	13910			15710	1210	18200	1402	18200	1402	12550	967	16060	1237†	1372	1733
[1]47.10	P-110	29420	16550			15960	1353	18220	1545	18220	1545	11410	967	14590	1237†	1512	1910
[1]51.20	P-110	32070	18040			16180	1491	16800	1548	18260	1683	10490	967	13420	1237†	1647	2081
[1]33.70	V-150	15750	8860			19900	1207	23480	1424	—	—	20390	1237	—	—	1458	1652
[1]39.00	V-150	23910	13450			20340	1428	23360	1640	—	—	17620	1237	—	—	1679	1903
[1]42.80	V-150	31150	17520			21030	1620	23710	1827	—	—	16060	1237	—	—	1871	2120
[1]47.10	V-150	38490	21650			21370	1812	23370	1981	—	—	14590	1237	—	—	2062	2336
[1]51.20	V-150	43720	24590			21670	1997	21500	1981	—	—	13420	1237	—	—	2246	2546

Fig. 3J-9 (7A) (Courtesy U.S. Steel)

USS Seamless Casing

Maximum Coupling Performance Properties

Grade	Tension (Kips)			Int. Yield Press. (Psi)		
	Round	Buttress		Round	Buttress	
	Long	Regular	SCC	Long	Regular	SCC
K-55	1255	1176	735	7210	6730	4500
C-75	1255	1176	735	9840	9180	6140
N-80	1322	1238	773	10490	9790	6550
L-80	1255	1176	735	10490	9790	6550
C-95	1388	1300	812	12460	11620	7780
1USS95	1454	1362	851	12460	11620	7780
P-110	1652	1548	967	14430	13460	9000
1 V-150	2114	1981	1237	19680	18350	12280

— Not Available

* Same as Round T&C Pipe

1 Next Higher Grade Insufficient for Maximum Pressure of Pipe

Wt/ft	Grade	Minimum Internal Yield Pressure						Mill Test Pressure						80% Minimum Yield Strength Also Maximum Formation Fracturing Pressure					
		Pipe Body	Round	Buttress				PE	Round	Buttress				PE	Round	Buttress			
				Regular		Special Clearance				Regular		Special Clearance				Regular		Special Clearance	
				Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade
20.00	H-40	2300	*					2100	2100					2100	*				
24.00	H-40	2750	*					2500	2500					2500	*				
26.40	K-55	4140	*	*	*	*	*	3000	3000	3000	3800	3600	3800	3800	*	*	*	3600	*
26.40	C-75	5650	*	*	*	*	*	3000	5200	5200	5200	4900	5200	5200	*	*	*	4900	*
29.70	C-75	6450	*	*	*	6140	*	3000	5900	5900	5900	4900	5900	5900	*	*	*	4900	*
33.70	C-75	7400	*	*	*	6140	*	3000	6800	6800	6800	4900	6200	6800	*	*	*	4900	6200†
39.00	C-75	8610	*	*	*	6140	7780	3000	7900	7300	7900	4900	6200	7900	*	7300	*	4900	6200†
26.40	N-80	6020	*	*	*	*	*	3000	5500	5500	5500	5200	5500	5500	*	*	*	5200	*
29.70	N-80	6890	*	*	*	6550	*	3000	6300	6300	6300	5200	6300	6300	*	*	*	5200	*
33.70	N-80	7900	*	*	*	6550	*	3000	7200	7200	7200	5200	7200	7200	*	*	*	5200	*
39.00	N-80	9180	*	*	*	6550	9000	3000	8400	7800	8400	5200	7200	8400	*	7800	*	5200	7200†
26.40	L-80	6020	*	*	*	*	*	3000	5500	5500	5500	5200	5500	5500	*	*	*	5200	*
29.70	L-80	6890	*	*	*	6550	*	3000	6300	6300	6300	5200	6200	6300	*	*	*	5200	6200†
33.70	L-80	7900	*	*	*	6550	7780	3000	7200	7200	7200	5200	6200	7200	*	*	*	5200	6200†
39.00	L-80	9180	*	*	*	6550	7780	3000	8400	7800	8400	5200	6200	8400	*	7800	*	5200	6200†
26.40	C-95	7150	*	*	—	*	—	3000	6500	6500	—	6200	—	6500	*	*	—	6200	—
29.70	C-95	8180	*	*	—	7780	—	3000	7500	7500	—	6200	—	7500	*	*	—	6200	—
33.70	C-95	9380	*	*	—	7780	—	3000	8600	8600	—	6200	—	8600	*	*	—	6200	—
39.00	C-95	10900	*	*	—	7780	—	3000	10000	9300	—	6200	—	10000	*	9300	—	6200	—
42.80	C-95	12250	*	11620	—	7780	—	3000	10000	9300	—	6200	—	11200	10000	9300	—	6200	—
47.10	C-95	13630	12460	11620	—	7780	—	3000	10000	9300	—	6200	—	12500	10000	9300	—	6200	—
51.20	C-95	14980	12460	11620	—	7780	—	3000	10000	9300	—	6200	—	13700	10000	9300	—	6200	—
26.40	USS95	7150	*	*	*	*	*	3000	6500	6500	6500	6200	6500	6500	*	*	*	6200	*
29.70	USS95	8180	*	*	*	7780	*	3000	7500	7500	7500	6200	7200	7500	*	*	*	6200	7200†
33.70	USS95	9380	*	*	*	7780	9000	3000	8600	8600	8600	6200	7200	8600	*	*	*	6200	7200†
26.40	P-110	8280	*	*	*	*	*	3000	7600	7600	7600	7200	7600	7600	*	*	*	7200	*
29.70	P-110	9470	*	*	*	9000	*	3000	8700	8700	8700	7200	8700	8700	*	*	*	7200	*
33.70	P-110	10860	*	*	*	9000	*	3000	9900	9900	9900	7200	9800	9900	*	*	*	7200	9800†
39.00	P-110	12620	*	*	*	9000	12280	3000	10000	10000	11500	7200	9800	11500	*	10800	*	7200	9800†
42.80	P-110	14190	*	13460	*	9000	12280	3000	10000	10000	11500	7200	9800	13000	11500	10800	*	7200	9800†
47.10	P-110	15780	14430	13460	*	9000	12280	3000	10000	10000	11500	7200	9800	14400	11500	10800	*	7200	9800†
51.20	P-110	17340	14430	13460	*	9000	12280	3000	10000	10000	11500	7200	9800	15900	11500	10800	14700	7200	9800†
33.70	V-150	14800	*	*	—	12280	—	3000	12000	12000	—	9800	—	13500	*	*	—	9800	—
39.00	V-150	17210	*	*	—	12280	—	3000	12000	12000	—	9800	—	15700	*	14700	—	9800	—
42.80	V-150	19350	*	18350	—	12280	—	3000	12000	12000	—	9800	—	17700	15700	14700	—	9800	—
47.10	V-150	21520	19680	18350	—	12280	—	3000	12000	12000	—	9800	—	19700	15700	14700	—	9800	—
51.20	V-150	23650	19680	18350	—	12280	—	3000	12000	12000	—	9800	—	21600	15700	14700	—	9800	—

Fig. 3J-9 (7B) (Courtesy U.S. Steel)

8⅝" OD

Casing Dimensions

Wt/ft (Lbs)		Casing Body				Areas (Sq. In.)	
Nominal T&C	Plain End	Wall (In.)	ID (In.)	Drift (In.)	D/T Ratio	PE and Buttress	Last Perfect Thread (8Rd)
24.00	23.57	.264	8.097	7.972	32.67	6.934	5.020
28.00	27.02	.304	8.017	7.892	28.37	7.947	6.032
32.00	31.10	.352	7.921	7.796	24.50	9.149	7.234
36.00	35.14	.400	7.825	7.700	21.56	10.336	8.421
40.00	39.29	.450	7.725	7.600	19.17	11.557	9.642
44.00	43.39	.500	7.625	7.500	17.25	12.763	10.848
49.00	48.00	.557	7.511	7.386	15.48	14.118	12.203

Coupling Dimensions

OD REG.= 9.625 INCHES
OD SCC= 9.125 INCHES
LENGTH SHORT= 7.750 INCHES
LENGTH LONG=10.000 INCHES
LENGTH BUTT.=10.675 INCHES
D1 SHORT= 8.432 INCHES
D1 LONG= 8.361 INCHES
D1 BUTT.= 8.452 INCHES
AREA SHORT=16.919 SQ.INCHES
AREA LONG=17.856 SQ.INCHES
AREA BUTTRESS=16.654 SQ.INCHES
AREA BUTT.SCC= 9.291 SQ.INCHES

Wt/ft	Grade	Collapse		[3]Joint Strength												Plain End Strength	
		[1,2] Feet @ 1.125	Psi	Round				Buttress									
				Short		Long		Regular				Special Clearance				Yield (Min.)	[1]Ultimate (Avg.)
								Same Grade		Higher Grade		Same Grade		Higher Grade			
				[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	Kips	Kips
28.00	H-40	2920	1640	4620	233											318	596
32.00	H-40	3930	2210	4840	279											366	686
24.00	K-55	2440	1370	6090	263											381	742
32.00	K-55	4500	2530	6980	402	7850	452	11980	690	11980	690	11980	690	11980	690	503	979
36.00	K-55	6130	3450	7220	468	8120	526	12040	780	12040	780	12040	780	12040	780	568	1106
36.00	C-75	7150	4020			10000	648	13070	847	13070	847	12950	839	13070	847	775	1137
40.00	C-75	9510	5350			10310	742	13150	947	13150	947	11650	839	12880	927†	867	1271
44.00	C-75	11880	6680			10530	834	13210	1046	13210	1046	10590	839	11700	927†	957	1404
49.00	C-75	14580	8200			10650	939	13120	1157	13120	1157	9510	839	10510	927†	1059	1553
36.00	N-80	7290	4100			10620	688	13810	895	13810	895	13630	883	13810	895	827	1189
40.00	N-80	9810	5520			10940	788	13900	1001	13900	1001	12260	883	13900	1001	925	1329
44.00	N-80	12360	6950			11200	887	13950	1105	13950	1105	11150	883	13930	1103†	1021	1468
49.00	N-80	15240	8570			11300	997	13850	1222	13850	1222	10010	883	12510	1103†	1129	1624
36.00	L-80	7290	4100			10460	678	13330	864	13330	864	12950	839	13330	864	827	1137
40.00	L-80	9810	5520			10780	776	13420	966	13420	966	11650	839	12880	927†	925	1271
44.00	L-80	12360	6950			11040	874	13460	1066	13460	1066	10590	839	11700	927†	1021	1404
49.00	L-80	15240	8570			11150	983	13380	1180	13380	1180	9510	839	10510	927†	1129	1553
36.00	C-95	7750	4360			12180	789	15060	976	—	—	14310	927	—	—	982	1282
40.00	C-95	10680	6010			12560	904	15170	1092	—	—	12880	927	—	—	1098	1433
44.00	C-95	13740	7730			12840	1017	15230	1206	—	—	11700	927	—	—	1212	1533
49.00	C-95	17230	9690			12970	1144	15120	1334	—	—	10510	927	—	—	1341	1751
[1] 36.00	USS95	10770	6060			12350	800	15560	1008	15560	1008	14980	971	15560	1008	982	1313
[1] 40.00	USS95	14040	7900			12710	915	15650	1127	15650	1127	13490	971	15320	1103†	1098	1468
[1] 44.00	USS95	16180	9100			13010	1030	15710	1244	15710	1244	12260	971	13930	1103†	1212	1621
[1] 49.00	USS95	18490	10400			13140	1159	15610	1377	15610	1377	11010	971	12510	1103†	1341	1793
40.00	P-110	11340	6380			14650	1055	17890	1288	17890	1288	15320	1103	17890	1288	1271	1606
44.00	P-110	14930	8400			14970	1186	17970	1423	17970	1423	13930	1103	17830	1412†	1404	1774
49.00	P-110	19040	10710			15140	1335	17850	1574	17850	1574	12510	1103	16010	1412†	1553	1962
[1] 44.00	V-150	17140	9640			20090	1591	23470	1859	—	—	17830	1412	—	—	1914	2170
[1] 49.00	V-150	23020	12950			20280	1789	23310	2056	—	—	16010	1412	—	—	2118	2400

Fig. 3J-9 (8A) (Courtesy U.S. Steel)

USS Seamless Casing

Maximum Coupling Performance Properties

Grade	Tension (Kips)			Int. Yield Press. (Psi)		
	Round	Buttress		Round	Buttress	
	Long	Regular	SCC	Long	Regular	SCC
K-55	1612	1503	839	7220	6700	4060
C-75	1612	1503	839	9850	9140	5530
N-80	1696	1582	883	10510	9750	5900
L-80	1612	1503	839	10510	9750	5900
C-95	1781	1661	927	12480	11580	7010
[1] USS95	1866	1740	971	12480	11580	7010
P-110	2120	1978	1103	14450	13410	8110
[1] V-150	2714	2531	1412	19700	18280	11060

— Not Available

* Same as Round T&C Pipe

† Next Higher Grade Insufficient for Maximum Pressure of Pipe

Wt/ft	Grade	Minimum Internal Yield Pressure						Mill Test Pressure						80% Minimum Yield Strength Also Maximum Formation Fracturing Pressure					
		Pipe Body	Round	Buttress				PE	Round	Buttress				PE	Round	Buttress			
				Regular		Special Clearance				Regular		Special Clearance				Regular		Special Clearance	
				Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade
28.00	H-40	2470	*					2300	2300					2300	*				
32.00	H-40	2860	*					2600	2600					2600	*				
24.00	K-55	2950	*					2700	2700					2700	*				
32.00	K-55	3930	*	*	*	*	*	3000	3000	3000	3600	3200	3600	3600	*	*	*	3200	*
36.00	K-55	4460	*	*	*	4060	*	3000	3000	3000	4100	3200	4100	4100	*	*	*	3200	*
36.00	C-75	6090	*	*	*	5530	*	3000	5600	5600	5600	4400	5600	5600	*	*	*	4400	*
40.00	C-75	6850	*	*	*	5530	*	3000	6300	6300	6300	4400	5600	6300	*	*	*	4400	5600†
44.00	C-75	7610	*	*	*	5530	7010	3000	7000	7000	7000	4400	5600	7000	*	*	*	4400	5600†
49.00	C-75	8480	*	*	*	5530	7010	3000	7700	7300	7700	4400	5600	7700	*	7300	*	4400	5600†
36.00	N-80	6490	*	*	*	5900	*	3000	5900	5900	5900	4700	5900	5900	*	*	*	4700	*
40.00	N-80	7300	*	*	*	5900	*	3000	6700	6700	6700	4700	6500	6700	*	*	*	4700	6500†
44.00	N-80	8120	*	*	*	5900	8110	3000	7400	7400	7400	4700	6500	7400	*	*	*	4700	6500†
49.00	N-80	9040	*	*	*	5900	8110	3000	8300	7800	8300	4700	6500	8300	*	7800	*	4700	6500†
36.00	L-80	6490	*	*	*	5900	*	3000	5900	5900	5900	4700	5600	5900	*	*	*	4700	5600†
40.00	L-80	7300	*	*	*	5900	7010	3000	6700	6700	6700	4700	5600	6700	*	*	*	4700	5600†
44.00	L-80	8120	*	*	*	5900	7010	3000	7400	7400	7400	4700	5600	7400	*	*	*	4700	5600†
49.00	L-80	9040	*	*	*	5900	7010	3000	8300	7800	8300	4700	5600	8300	*	7800	*	4700	5600†
36.00	C-95	7710	*	*	—	7010	—	3000	7000	7000	—	5600	—	7000	*	*	—	5600	—
40.00	C-95	8670	*	*	—	7010	—	3000	7900	7900	—	5600	—	7900	*	*	—	5600	—
44.00	C-95	9640	*	*	—	7010	—	3000	8800	8800	—	5600	—	8800	*	*	—	5600	—
49.00	C-95	10740	*	*	—	7010	—	3000	9800	9300	—	5600	—	9800	*	9300	—	5600	—
36.00	USS95	7710	*	*	*	7010	*	3000	7000	7000	7000	5600	6500	7000	*	*	*	5600	6500†
40.00	USS95	8670	*	*	*	7010	8110	3000	7900	7900	7900	5600	6500	7900	*	*	*	5600	6500†
44.00	USS95	9640	*	*	*	7010	8110	3000	8800	8800	8800	5600	6500	8800	*	*	*	5600	6500†
49.00	USS95	10740	*	*	*	7010	8110	3000	9800	9300	9800	5600	6500	9800	*	9300	*	5600	6500†
40.00	P-110	10040	*	*	*	8110	*	3000	9200	9200	9200	6500	8900	9200	*	*	*	6500	8900†
44.00	P-110	11160	*	*	*	8110	11060	3000	10000	10000	10200	6500	8900	10200	*	*	*	6500	8900†
49.00	P-110	12430	*	*	*	8110	11060	3000	10000	10000	11400	6500	8900	11400	*	10700	*	6500	8900†
44.00	V-150	15220	*	*	—	11060	—	3000	12000	12000	—	8900	—	13900	*	*	—	8900	—
49.00	V-150	16950	*	*	—	11060	—	3000	12000	12000	—	8900	—	15500	*	14600	—	8900	—

Fig. 3J-9 (8B) (Courtesy U.S. Steel)

9⅝" OD

Casing Dimensions

Wt/ft (Lbs)		Casing Body				Areas (Sq. In.)	
Nominal T&C	Plain End	Wall (In.)	ID (In.)	Drift (In.)	D/T Ratio	PE and Buttress	Last Perfect Thread (8Rd)
29.30	28.04	.281	9.063	8.907	34.25	8.249	6.110
32.30	31.03	.312	9.001	8.845	30.85	9.128	6.990
36.00	34.86	.352	8.921	8.765	27.34	10.254	8.116
40.00	38.94	.395	8.835	8.679	24.37	11.454	9.315
43.50	42.70	.435	8.755	8.599	22.13	12.559	10.421
47.00	46.14	.472	8.681	8.525	20.39	13.572	11.434
53.50	52.85	.545	8.535	8.379	17.66	15.547	13.408
[1]59.40	58.64	.609	8.407	8.251	15.80	17.250	15.111
[1]64.90	64.26	.672	8.281	8.125	14.32	18.901	16.763
[1]70.30	69.70	.734	8.157	8.001	13.11	20.502	18.364

Coupling Dimensions

OD REG. = 10.625 INCHES
OD SCC = 10.125 INCHES
LENGTH SHORT = 7.750 INCHES
LENGTH LONG = 10.500 INCHES
LENGTH BUTT. = 10.675 INCHES
D1 SHORT = 9.432 INCHES
D1 LONG = 9.346 INCHES
D1 BUTT. = 9.452 INCHES
AREA SHORT = 18.793 SQ.INCHES
AREA LONG = 20.061 SQ.INCHES
AREA BUTTRESS = 18.496 SQ.INCHES
AREA BUTT.SCC = 10.348 SQ.INCHES

Wt/ft	Grade	Collapse		[3]Joint Strength												Plain End Strength	
		[1,2] Feet @ 1.125	Psi	Round				Buttress								Yield (Min.)	[1]Ultimate (Avg.)
				Short		Long		Regular				Special Clearance					
								Same Grade		Higher Grade		Same Grade		Higher Grade			
				[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	Kips	Kips
[1]29.30	H-40	2030	1140	4210	222											330	619
32.30	H-40	2490	1400	4370	254											365	685
36.00	H-40	3090	1740	4540	294											410	769
[1]29.30	K-55	2170	1220	6050	319											454	883
36.00	K-55	3590	2020	6530	423	7550	489	11650	755	11650	755	11650	755	11650	755	564	1097
40.00	K-55	4570	2570	6750	486	7790	561	11710	843	11710	843	11710	843	11710	843	630	1226
40.00	C-75	5300	2980			9640	694	12860	926	12860	926	12860	926	12860	926	859	1260
43.50	C-75	6670	3750			9910	776	12980	1016	12980	1016	11930	934	12980	1016	942	1381
47.00	C-75	8230	4630			10070	852	12980	1098	12980	1098	11040	934	12200	1032†	1013	1493
53.50	C-75	11340	6390			10370	999	13050	1257	13050	1257	9700	934	10720	1032†	1166	1710
40.00	N-80	5490	3090			10240	737	13600	979	13600	979	13600	979	13600	979	916	1317
43.50	N-80	6770	3810			10540	825	13720	1074	13720	1074	12550	983	13720	1074	1005	1444
47.00	N-80	8440	4750			10700	905	13720	1161	13720	1161	11620	983	13720	1161	1086	1561
53.50	N-80	11770	6620			11030	1062	13800	1329	13800	1329	10210	983	12760	1229†	1244	1788
40.00	L-80	5490	3090			10100	727	13150	947	13150	947	12970	934	13150	947	916	1260
43.50	L-80	6770	3810			10380	813	13260	1038	13260	1038	11930	934	13180	1032†	1005	1381
47.00	L-80	8440	4750			10560	893	13260	1122	13260	1122	11040	934	12200	1032†	1086	1493
53.50	L-80	11770	6620			10870	1047	13350	1286	13350	1286	9700	934	10720	1032†	1244	1710
40.00	C-95	5920	3330			11760	847	14920	1074	—	—	14330	1032	—	—	1088	1420
43.50	C-95	7340	4130			12110	948	15040	1178	—	—	13180	1032	—	—	1193	1557
47.00	C-95	9030	5080			12290	1040	15050	1273	—	—	12200	1032	—	—	1289	1683
53.50	C-95	13030	7330			12670	1220	15140	1458	—	—	10720	1032	—	—	1477	1928
[1]59.40	C-95	16530	9300			12860	1375	15130	1618	—	—	9650	1032	—	—	1639	2139
[1]64.90	C-95	19980	11240			13050	1525	15170	1772	—	—	8830	1032	—	—	1796	2344
[1]70.30	C-95	23400	13160			13200	1670	14580	1845	—	—	8160	1032	—	—	1948	2542
[1]40.00	USS95	7520	4230			11920	858	15360	1106	15360	1106	15010	1081	15360	1106	1088	1455
[1]43.50	USS95	9960	5600			12260	960	15490	1213	15490	1213	13810	1081	15490	1213	1193	1595
[1]47.00	USS95	12620	7100			12450	1053	15500	1311	15500	1311	12780	1081	14530	1229†	1289	1724
[1]53.50	USS95	15730	8850			12820	1235	15600	1502	15600	1502	11230	1081	12760	1229†	1477	1974
[1]59.40	USS95	18120	10190			13010	1391	15580	1666	15580	1666	10110	1081	11490	1229†	1639	2191
43.50	P-110	7880	4430			14130	1106	17730	1388	17730	1388	15700	1229	17730	1388	1381	1746
47.00	P-110	9440	5310			14340	1213	17730	1500	17730	1500	14530	1229	17730	1500	1493	1887
53.50	P-110	14100	7930			14770	1422	17840	1718	17840	1718	12760	1229	16330	1573†	1710	2161
[1]59.40	P-110	18240	10260			14990	1603	17830	1906	17830	1906	11490	1229	14710	1573†	1898	2398
[1]64.90	P-110	22310	12550			15220	1778	17880	2089	17880	2089	10520	1229	13470	1573†	2079	2627
[1]70.30	P-110	26310	14800			15390	1948	17350	2196	17910	2266	9710	1229	12430	1573†	2255	2850
[1]53.50	V-150	15950	8970			19820	1909	23370	2251	—	—	16330	1573	—	—	2332	2643
[1]59.40	V-150	21870	12300			20120	2151	23360	2498	—	—	14710	1573	—	—	2588	2933
[1]64.90	V-150	27680	15570			20420	2386	23430	2737	—	—	13470	1573	—	—	2835	3213
[1]70.30	V-150	33400	18790			20660	2614	22210	2811	—	—	12430	1573	—	—	3075	3485

Fig. 3J-9 (9A) (Courtesy U.S. Steel)

USS Seamless Casing

Maximum Coupling Performance Properties

Grade	Tension (Kips)			Int. Yield Press. (Psi)		
	Round	Buttress		Round	Buttress	
	Long	Regular	SCC	Long	Regular	SCC
K-55	1811	1669	934	6620	6070	3660
C-75	1811	1669	934	9030	8280	4990
N-80	1906	1757	983	9630	8830	5320
L-80	1811	1669	934	9630	8830	5320
C-95	2001	1845	1032	11440	10490	6310
†USS95	2096	1933	1081	11440	10490	6310
P-110	2382	2196	1229	13240	12140	7310
† V-150	3049	2811	1573	18060	16560	9970

— Not Available

* Same as Round T&C Pipe

† Next Higher Grade Insufficient for Maximum Pressure of Pipe

Wt/ft	Grade	Minimum Internal Yield Pressure						Mill Test Pressure						80% Minimum Yield Strength Also Maximum Formation Fracturing Pressure					
		Pipe Body	Round	Buttress				PE	Round	Buttress				PE	Round	Buttress			
				Regular		Special Clearance				Regular		Special Clearance				Regular		Special Clearance	
				Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade
29.30	H-40	2040	*					1900	1900					1900	*				
32.30	H-40	2270	*					2100	2100					2100	*				
36.00	H-40	2560	*					2300	2300					2300	*				
29.30	K-55	2810	*					2600	2600					2600	*				
36.00	K-55	3520	*	*	*	*	*	3000	3000	3000	3200	2900	3200	3200	*	*	*	2900	*
40.00	K-55	3950	*	*	*	3660	*	3000	3000	3000	3600	2900	3600	3600	*	*	*	2900	*
40.00	C-75	5390	*	*	*	4990	*	3000	4900	4900	4900	4000	4900	4900	*	*	*	4000	*
43.50	C-75	5930	*	*	*	4990	*	3000	5400	5400	5400	4000	5100	5400	*	*	*	4000	5100†
47.00	C-75	6440	*	*	*	4990	6310	3000	5900	5900	5900	4000	5100	5900	*	*	*	4000	5100†
53.50	C-75	7430	*	*	*	4990	6310	3000	6800	6600	6800	4000	5100	6800	*	6600	*	4000	5100†
40.00	N-80	5750	*	*	*	5320	*	3000	5300	5300	5300	4300	5300	5300	*	*	*	4300	*
43.50	N-80	6330	*	*	*	5320	*	3000	5800	5800	5800	4300	5800	5800	*	*	*	4300	*
47.00	N-80	6670	*	*	*	5320	*	3000	6300	6300	6300	4300	5800	6300	*	*	*	4300	5600†
53.50	N-80	7930	*	*	*	5320	7310	3000	7200	7100	7200	4300	5800	7200	*	7100	*	4300	5800†
40.00	L-80	5750	*	*	*	5320	*	3000	5300	5300	5300	4300	5100	5300	*	*	*	4300	5100†
43.50	L-80	6330	*	*	*	5320	6310	3000	5800	5800	5800	4300	5100	5800	*	*	*	4300	5100†
47.00	L-80	6870	*	*	*	5320	6310	3000	6300	6300	6300	4300	5100	6300	*	*	*	4300	5100†
53.50	L-80	7930	*	*	*	5320	6310	3000	7200	7100	7200	4300	5100	7200	*	7100	*	4300	5100†
40.00	C-95	6820	*	*	—	6310	—	3000	6200	6200	—	5100	—	6200	*	*	—	5100	—
43.50	C-95	7510	*	*	—	6310	—	3000	6900	6900	—	5100	—	6900	*	*	—	5100	—
47.00	C-95	8150	*	*	—	6310	—	3000	7500	7500	—	5100	—	7500	*	*	—	5100	—
53.50	C-95	9410	*	*	—	6310	—	3000	8600	8400	—	5100	—	8600	*	8400	—	5100	—
59.40	C-95	10520	*	10490	—	6310	—	3000	9200	8400	—	5100	—	9600	9200	8400	—	5100	—
64.90	C-95	11610	11440	10490	—	6310	—	3000	9200	8400	—	5100	—	10600	9200	8400	—	5100	—
70.30	C-95	12680	11440	10490	—	6310	—	3000	9200	8400	—	5100	—	11600	9200	8400	—	5100	—
40.00	USS95	6820	*	*	*	6310	*	3000	6200	6200	6200	5100	5800	6200	*	*	*	5100	5800*
43.50	USS95	7510	*	*	*	6310	7310	3000	6900	6900	6900	5100	5800	6900	*	*	*	5100	5800†
47.00	USS95	8150	*	*	*	6310	7310	3000	7500	7500	7500	5100	5800	7500	*	*	*	5100	5800†
53.50	USS95	9410	*	*	*	6310	7310	3000	8600	8400	8600	5100	5800	8600	*	8400	*	5100	5800†
59.40	USS95	10520	*	10490	*	6310	7310	3000	9200	8400	9200	5100	5800	9600	9200	8400	*	5100	5800†
43.50	P-110	8700	*	*	*	7310	*	3000	8000	8000	8000	5800	8000	8000	*	*	*	5800	*
47.00	P-110	9440	*	*	*	7310	*	3000	8600	8600	8600	5800	8000	8600	*	*	*	5800	8000†
53.50	P-110	10900	*	*	*	7310	9970	3000	10000	9700	10000	5800	8000	10000	*	9700	*	5800	8000†
59.40	P-110	12180	*	12140	*	7310	9970	3000	10000	9700	10600	5800	8000	11100	10600	9700	*	5800	8000†
64.90	P-110	13440	13240	12140	*	7310	9970	3000	10000	9700	10600	5800	8000	12300	10600	9700	*	5800	8000†
70.30	P-110	14680	13240	12140	*	7310	9970	3000	10000	9700	10600	5800	8000	13400	10600	9700	13200	5800	8000†
53.50	V-150	14860	*	*	—	9970	—	3000	12000	12000	—	8000	—	13600	*	13200	—	8000	—
59.40	V-150	16610	*	16560	—	9970	—	3000	12000	12000	—	8000	—	15200	14400	13200	—	8000	—
64.90	V-150	18330	18060	16560	—	9970	—	3000	12000	12000	—	8000	—	16800	14400	13200	—	8000	—
70.30	V-150	20020	18060	16560	—	9970	—	3000	12000	12000	—	8000	—	18300	14400	13200	—	8000	—

Fig. 3J-9 (9B) (Courtesy U.S. Steel)

10¾" OD

Casing Dimensions

Wt/ft (Lbs)		Casing Body				Areas (Sq. In.)	
Nominal T&C	Plain End	Wall (In.)	ID (In.)	Drift (In.)	D/T Ratio	PE and Buttress	Last Perfect Thread (8Rd)
32.75	31.20	.279	10.192	10.036	38.53	9.178	6.788
40.50	38.88	.350	10.050	9.894	30.71	11.435	9.045
45.50	44.22	.400	9.950	9.794	26.88	13.006	10.616
51.00	49.50	.450	9.850	9.694	23.89	14.561	12.171
55.50	54.21	.495	9.760	9.604	21.72	15.947	13.557
60.70	59.40	.545	9.660	9.504	19.72	17.473	15.082
65.70	64.53	.595	9.560	9.404	18.07	18.982	16.592
[1]73.20	72.33	.672	9.406	9.250	16.00	21.276	18.886
[1]79.20	78.52	.734	9.282	9.126	14.65	23.096	20.706

Coupling Dimensions

OD REG.=11.750 INCHES
OD SCC=11.250 INCHES
LENGTH SHORT= 8.000 INCHES
LENGTH BUTT.=10.675 INCHES
D1 SHORT=10.549 INCHES
D1 BUTT.=10.577 INCHES
AREA SHORT=21.034 SQ.INCHES
AREA BUTTRESS=20.569 SQ.INCHES
AREA BUTT.SCC=11.537 SQ.INCHES

Wt/ft	Grade	Collapse		[3]Joint Strength												Plain End Strength	
		[1,2] Feet @ 1.125	Psi	Round				Buttress									
				Short		Long		Regular				Special Clearance				Yield (Min.)	[1]Ultimate (Avg.)
								Same Grade		Higher Grade		Same Grade		Higher Grade			
				[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	Kips	Kips
32.75	H-40	1560	880	3480	205											367	688
40.50	H-40	2520	1420	4310	314											457	858
40.50	K-55	2810	1580	6170	450			11230	819	11230	819	11230	819	11230	819	629	1224
45.50	K-55	3720	2090	6450	528			11370	931	11370	931	11370	931	11370	931	715	1392
51.00	K-55	4800	2700	6600	606			11360	1043	11360	1043	11340	1041	11360	1043	801	1558
51.00	C-75	5510	3100	8240	756			12640	1160	12640	1160	11340	1041	12540	1151†	1092	1602
55.50	C-75	7020	3950	8430	842			12720	1271	12720	1271	10420	1041	11520	1151†	1196	1754
51.00	N-80	5720	3220	8760	804			13380	1228	13380	1228	11940	1096	13380	1228	1165	1675
55.50	N-80	7150	4020	8960	895			13460	1345	13460	1345	10970	1096	13460	1345	1276	1834
51.00	L-80	5720	3220	8650	794			12960	1190	12960	1190	11340	1041	12540	1151†	1165	1602
55.50	L-80	7150	4020	8850	884			13040	1303	13040	1303	10420	1041	11520	1151†	1276	1754
51.00	C-95	6200	3490	10100	927			14750	1354	—	—	12540	1151	—	—	1383	1806
55.50	C-95	7640	4300	10330	1032			14840	1483	—	—	11520	1151	—	—	1515	1977
60.70	C-95	9900	5570	10510	1148			14870	1625	—	—	10530	1151	—	—	1660	2167
65.70	C-95	12360	6950	10680	1263			14920	1765	—	—	9730	1151	—	—	1803	2354
[1] 73.20	C-95	16140	9080	10910	1438			15010	1978	—	—	8740	1151	—	—	2021	2638
[1] 79.20	C-95	19180	10790	11050	1576			14390	2052	—	—	8070	1151	—	—	2194	2864
[1] 51.00	USS95	7930	4460	10210	937			15160	1392	15160	1392	13140	1206	14920	1370†	1383	1849
[1] 55.50	USS95	10580	5950	10440	1043			15260	1524	15260	1524	12070	1206	13710	1370†	1515	2025
[1] 60.70	USS95	13420	7550	10630	1161			15280	1670	15280	1670	11040	1206	12540	1370†	1660	2219
[1] 65.70	USS95	15360	8640	10800	1277			15340	1814	15340	1814	10200	1206	11580	1370†	1803	2411
[1] 73.20	USS95	17880	10060	11030	1453			15440	2034	15440	2034	9150	1206	10400	1370†	2021	2702
51.00	P-110	6520	3670	11760	1080			17360	1594	17360	1594	14920	1370	17360	1594	1602	2024
55.50	P-110	8230	4630	12040	1203			17470	1745	17470	1745	13710	1370	17470	1745	1754	2217
60.70	P-110	10420	5860	12250	1338			17500	1912	17500	1912	12540	1370	16050	1754†	1922	2429
65.70	P-110	13320	7490	12450	1472			17560	2077	17560	2077	11580	1370	14830	1754†	2088	2639
[1] 73.20	P-110	17760	9990	12720	1676			17670	2328	17670	2328	10400	1370	13310	1754†	2340	2957
[1] 79.20	P-110	21350	12010	12890	1837			17140	2443	17730	2528	9610	1370	12300	1754†	2541	3210
[1] 60.70	V-150	11660	6560	16460	1798			23000	2513	—	—	16050	1754	—	—	2621	2970
[1] 65.70	V-150	14810	8330	16730	1978			23080	2730	—	—	14830	1754	—	—	2847	3227
[1] 73.20	V-150	21190	11920	17090	2252			23220	3060	—	—	13310	1754	—	—	3191	3617
[1] 79.20	V-150	26310	14800	17320	2469			21930	3126	—	—	12300	1754	—	—	3464	3926

Fig. 3J-9 (10A) (Courtesy U.S. Steel)

USS Seamless Casing

Maximum Coupling Performance Properties

Grade	Tension (Kips)			Int. Yield Press. (Psi)		
	Round	Buttress		Round	Buttress	
	Short	Regular	SCC	Short	Regular	SCC
K-55	1898	1856	1041	5620	5490	3290
C-75	1898	1856	1041	7670	7490	4490
N-80	1998	1954	1096	8180	7990	4790
L-80	1898	1856	1041	8180	7990	4790
C-95	2098	2052	1151	9710	9480	5680
† USS95	2198	2149	1206	9710	9480	5680
P-110	2498	2443	1370	11240	10980	6580
† V-150	3197	3126	1754	15330	14970	8970

— Not Available

• Same as Round T&C Pipe

† Next Higher Grade Insufficient for Maximum Pressure of Pipe

Wt/ft	Grade	Minimum Internal Yield Pressure						Mill Test Pressure						80% Minimum Yield Strength Also Maximum Formation Fracturing Pressure					
		Pipe Body	Round	Buttress				PE	Round	Buttress				PE	Round	Buttress			
				Regular		Special Clearance				Regular		Special Clearance				Regular		Special Clearance	
				Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade
32.75	H-40	1820	•					1200	1200					1700	•				
40.50	H-40	2280	•					1600	1600					2100	•				
40.50	K-55	3130	•	•	•	•	•	2100	2100	2100	2100	2100	2100	2900	•	•	•	2600	•
45.50	K-55	3580	•	•	•	3290	•	2500	2500	2500	2500	2500	2500	3300	•	•	•	2600	•
51.00	K-55	4030	•	•	•	3290	•	2800	2800	2800	2800	2600	2800	3700	•	•	•	2600	•
51.00	C-75	5490	•	•	•	4490	•	3000	5000	5000	5000	3600	4500	5000	•	•	•	3600	4500•
55.50	C-75	6040	•	•	•	4490	5680	3000	5500	5500	5500	3600	4500	5500	•	•	•	3600	4500•
51.00	N-80	5860	•	•	•	4790	•	3000	5400	5400	5400	3800	5300	5400	•	•	•	3800	5300•
55.50	N-80	6450	•	•	•	4790	•	3000	5900	5900	5900	3800	5300	5900	•	•	•	3800	5300•
51.00	L-80	5860	•	•	•	4790	5680	3000	5400	5400	5400	3800	4500	5400	•	•	•	3800	4500•
55.50	L-80	6450	•	•	•	4790	5680	3000	5900	5900	5900	3800	4500	5900	•	•	•	3800	4500•
51.00	C-95	6960	•	•	—	5680	—	3000	6400	6400	—	4500	—	6400	•	•	—	4500	—
55.50	C-95	7660	•	•	—	5680	—	3000	7000	7000	—	4500	—	7000	•	•	—	4500	—
60.70	C-95	8430	•	•	—	5680	—	3000	7700	7600	—	4500	—	7700	•	7600	—	4500	—
65.70	C-95	9200	•	•	—	5680	—	3000	7800	7600	—	4500	—	8400	7800	7600	—	4500	—
73.20	C-95	10390	9710	9480	—	5680	—	3000	7800	7600	—	4500	—	9500	7800	7600	—	4500	—
79.20	C-95	11350	9710	9480	—	5680	—	3000	7800	7600	—	4500	—	10400	7800	7600	—	4500	—
51.00	USS95	6960	•	•	•	5680	6580	3000	6400	6400	6400	4500	5300	6400	•	•	•	4500	5300•
55.50	USS95	7660	•	•	•	5680	6580	3000	7000	7000	7000	4500	5300	7000	•	•	•	4500	5300•
60.70	USS95	8430	•	•	•	5680	6580	3000	7700	7600	7700	4500	5300	7700	•	7600	•	4500	5300•
65.70	USS95	9200	•	•	•	5680	6580	3000	7800	7600	7800	4500	5300	8400	7800	7600	•	4500	5300•
73.20	USS95	10390	9710	9480	•	5680	6580	3000	7800	7600	7800	4500	5300	9500	7800	7600	8800	4500	5300•
51.00	P-110	8060	•	•	•	6580	•	3000	7400	7400	7400	5300	7200	7400	•	•	•	5300	7200•
55.50	P-110	8860	•	•	•	6580	•	3000	8100	8100	8100	5300	7200	8100	•	•	•	5300	7200•
60.70	P-110	9760	•	•	•	6580	8970	3000	8900	8800	8900	5300	7200	8900	•	8800	•	5300	7200•
65.70	P-110	10650	•	•	•	6580	8970	3000	9000	8800	9000	5300	7200	9700	9000	8800	•	5300	7200•
73.20	P-110	12030	11240	10980	•	6580	8970	3000	9000	8800	9000	5300	7200	11000	9000	8800	•	5300	7200•
79.20	P-110	13140	11240	10980	•	6580	8970	3000	9000	8800	9000	5300	7200	12000	9000	8800	•	5300	7200•
60.70	V-150	13310	•	•	—	8970	—	3000	12000	12000	—	7200	—	12200	•	12000	—	7200	—
65.70	V-150	14530	•	•	—	8970	—	3000	12000	12000	—	7200	—	13300	12300	12000	—	7200	—
73.20	V-150	16410	15330	14970	—	8970	—	3000	12000	12000	—	7200	—	15000	12300	12000	—	7200	—
79.20	V-150	17920	15330	14970	—	8970	—	3000	12000	12000	—	7200	—	16400	12300	12000	—	7200	—

Fig. 3J-9 (10B) (Courtesy U.S. Steel)

11¾" OD

Casing Dimensions

Wt/ft (Lbs)		Casing Body				Areas (Sq. In.)	
Nominal T&C	Plain End	Wall (In.)	ID (In.)	Drift (In.)	D/T Ratio	PE and Buttress	Last Perfect Thread (8Rd)
42.00	40.60	.333	11.084	10.928	35.29	11.944	9.330
47.00	45.56	.375	11.000	10.844	31.33	13.401	10.787
54.00	52.57	.435	10.880	10.724	27.01	15.463	12.849
60.00	58.81	.489	10.772	10.616	24.03	17.300	14.685
[1]66.70	65.45	.547	10.656	10.500	21.48	19.252	16.638
[1]73.60	72.46	.609	10.532	10.376	19.29	21.315	18.701
[1]80.50	79.51	.672	10.406	10.250	17.49	23.387	20.773
[1]87.20	86.36	.734	10.282	10.126	16.01	25.402	22.788

Coupling Dimensions

OD REG.=12.750 INCHES
LENGTH SHORT= 8.000 INCHES
LENGTH BUTT.=10.675 INCHES
D1 SHORT=11.549 INCHES
D1 BUTT.=11.577 INCHES
AREA SHORT=22.920 SQ.INCHES
AREA BUTTRESS=22.412 SQ.INCHES

Wt/ft	Grade	Collapse		[3]Joint Strength												Plain End Strength	
		[1,2] Feet @ 1.125	Psi	Round				Buttress								Yield (Min.)	[1]Ultimate (Avg.)
				Short		Long		Regular				Special Clearance					
								Same Grade		Higher Grade		Same Grade		Higher Grade			
				[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	Kips	Kips
42.00	H-40	1900	1070	4060	307											478	896
47.00	K-55	2680	1510	6020	509			11050	935	11050	935					737	1434
54.00	K-55	3680	2070	6230	606			11100	1079	11100	1079					850	1655
60.00	K-55	4730	2660	6420	693			11190	1208	11190	1208					952	1851
60.00	C-75	5460	3070	8050	869			12600	1361	12600	1361					1298	1903
60.00	N-80	5650	3180	8560	924			13330	1440	13330	1440					1384	1990
60.00	L-80	5650	3180	8450	913			12950	1399	12950	1399					1384	1903
60.00	C-95	6120	3440	9870	1066			14780	1596	—	—					1644	2145
[1]66.70	C-95	7820	4400	10060	1208			14790	1776	—	—					1829	2387
[1]73.60	C-95	10490	5900	10250	1358			14850	1967	—	—					2025	2643
[1]80.50	C-95	13330	7500	10410	1509			14890	2158	—	—					2222	2900
[1]87.20	C-95	16110	9060	10540	1655			14250	2236	—	—					2413	3150
[1]60.00	USS95	7840	4410	9970	1077			15170	1638	15170	1638					1644	2197
[1]66.70	USS95	10900	6130	10170	1221			15180	1823	15180	1823					1829	2445
[1]73.60	USS95	13900	7820	10360	1372			15230	2018	15230	2018					2025	2707
[1]80.50	USS95	15930	8960	10520	1524			15280	2214	15280	2214					2222	2970
[1]87.20	USS95	17880	10060	10650	1672			14920	2342	15320	2405					2413	3226
[1]60.00	P-110	6420	3610	11500	1242			17380	1877	17380	1877					1903	2405
[1]66.70	P-110	8430	4740	11730	1408			17400	2089	17400	2089					2113	2676
[1]73.60	P-110	11130	6260	11940	1582			17450	2312	17450	2312					2345	2963
[1]80.50	P-110	14450	8130	12130	1757			17510	2537	17510	2537					2573	3251
[1]87.20	P-110	17740	9980	12280	1928			16950	2661	17560	2756					2794	3531
[1]66.70	V-150	9260	5210	15770	1893			22920	2752	—	—					2838	3273
[1]73.60	V-150	12320	6930	16060	2128			22990	3046	—	—					3197	3624
[1]80.50	V-150	16460	9250	16310	2364			23070	3343	—	—					3508	3976
[1]87.20	V-150	21140	11890	16520	2593			21710	3407	—	—					3810	4318

Fig. 3J-9 (11A) (Courtesy U.S. Steel)

USS Seamless Casing

Maximum Coupling Performance Properties

Grade	Tension (Kips)			Int. Yield Press. (Psi)		
	Round	Buttress		Round	Buttress	
	Short	Regular	SCC	Short	Regular	SCC
K-55	2069	2023		5180	5060	
C-75	2069	2023		7060	6900	
N-80	2177	2129		7540	7360	
L-80	2069	2023		7540	7360	
C-95	2286	2236		8950	8740	
1 USS95	2395	2342		8950	8740	
1 P-110	2722	2661		10360	10120	
1 V-150	3484	3407		14130	13800	

— Not Available

* Same as Round T&C Pipe

† Next Higher Grade Insufficient for Maximum Pressure of Pipe

Wt/ft	Grade	Minimum Internal Yield Pressure				Mill Test Pressure				80% Minimum Yield Strength Also Maximum Formation Fracturing Pressure			
		Pipe Body	Round	Buttress		PE	Round	Buttress		PE	Round	Buttress	
				Regular	Special Clearance			Regular	Special Clearance			Regular	Special Clearance
				Same Grade / Higher Grade	Same Grade / Higher Grade			Same Grade / Higher Grade	Same Grade / Higher Grade			Same Grade / Higher Grade	Same Grade / Higher Grade
42.00	H-40	1980	*	* / *		1400	1400	1400 / 1400		1600	*	* / *	
47.00	K-55	3070	*	* / *		2100	2100	2100 / 2100		2500	*	* / *	
54.00	K-55	3560	*	* / *		2400	2400	2400 / 2400		3300	*	* / *	
60.00	K-55	4010	*	* / *		2700	2700	2700 / 2700		3700	*	* / *	
60.00	C-75	5460	*	* / *		3000	5000	5000 / 5000		5000	*	* / *	
60.00	N-80	5830	*	* / *		3000	5300	5300 / 5300		5300	*	* / *	
60.00	L-80	5830	*	* / *		3000	5300	5300 / 5300		5300	*	* / *	
60.00	C-95	6920	*	* / —		3000	6300	6300 / —		6300	*	* / —	
66.70	C-95	7740	*	* / —		3000	7100	7000 / —		7100	*	7000 / —	
73.60	C-95	8620	*	* / —		3000	7200	7000 / —		7900	7200	7000 / —	
80.50	C-95	9510	8950	8740 / —		3000	7200	7000 / —		8700	7200	7000 / —	
87.20	C-95	10390	8950	8740 / —		3000	7200	7000 / —		9500	7200	7000 / —	
60.00	USS95	6920	*	* / *		3000	6300	6300 / 6300		6300	*	* / *	
66.70	USS95	7740	*	* / *		3000	7100	7000 / 7100		7100	*	7000 / *	
73.60	USS95	8620	*	* / *		3000	7200	7000 / 7200		7900	7200	7000 / *	
80.50	USS95	9510	8950	8740 / *		3000	7200	7000 / 7200		8700	7200	7000 / 8100	
87.20	USS95	10390	8950	8740 / 10120		3000	7200	7000 / 7200		9500	7200	7000 / 8100	
60.00	P-110	8010	*	* / *		3000	7300	7300 / 7300		7300	*	* / *	
66.70	P-110	8960	*	* / *		3000	8200	8100 / 8200		8200	*	8100 / *	
73.60	P-110	9980	*	* / *		3000	8300	8100 / 8300		9100	8300	8100 / *	
80.50	P-110	11010	10360	10120 / *		3000	8300	8100 / 8300		10100	8300	8100 / *	
87.20	P-110	12030	10360	10120 / *		3000	8300	8100 / 8300		11000	8300	8100 / *	
66.70	V-150	12220	*	* / —		3000	11200	11000 / —		11200	*	11000 / —	
73.60	V-150	13610	*	* / —		3000	11300	11000 / —		12400	11300	11000 / —	
80.50	V-150	15010	14130	13800 / —		3000	11300	11000 / —		13700	11300	11000 / —	
87.20	V-150	16400	14130	13800 / —		3000	11300	11000 / —		15000	11300	11000 / —	

Fig. 3J-9 (11B) (Courtesy U.S. Steel)

13⅜" OD

Casing Dimensions

Wt/ft (Lbs)		Casing Body				Areas (Sq. in.)	
Nominal T&C	Plain End	Wall (In.)	ID (In.)	Drift (In.)	D/T Ratio	PE and Buttress	Last Perfect Thread (8Rd)
48.00	45.98	.330	12.715	12.559	40.53	13.524	10.546
54.50	52.74	.380	12.615	12.459	35.20	15.514	12.536
61.00	59.45	.430	12.515	12.359	31.10	17.487	14.509
68.00	66.11	.480	12.415	12.259	27.86	19.445	16.467
72.00	70.60	.514	12.347	12.191	26.02	20.768	17.790
ı 76.60	74.94	.547	12.281	12.125	24.45	22.044	19.066
ı 84.50	83.03	.609	12.157	12.001	21.96	24.424	21.446
ı 92.50	91.17	.672	12.031	11.875	19.90	26.818	23.840
ı100.30	99.10	.734	11.907	11.751	18.22	29.149	26.171

Coupling Dimensions

OD REG.=14.375 INCHES
LENGTH SHORT= 8.000 INCHES
LENGTH BUTT.=10.675 INCHES
D1 SHORT=13.174 INCHES
D1 BUTT.=13.202 INCHES
AREA SHORT=25.986 SQ.INCHES
AREA BUTTRESS=25.406 SQ.INCHES

Wt/ft	Grade	Collapse		[3]Joint Strength										Plain End Strength	
		[1,2] Feet @ 1.125	Psi	Round				Buttress							
				Short		Long		Regular				Special Clearance		Yield (Min.)	ıUlti-mate (Avg.)
								Same Grade		Higher Grade		Same Grade	Higher Grade		
				ıFt @ 1.8	Kips	ıFt @ 1.8	Kips	ıFt @ 1.8	Kips	ıFt @ 1.8	Kips	ıFt @ 1.8	ıFt @ 1.8 Kips	Kips	Kips
48.00	H-40	1370	770	3730	322			7030	607	7030	607			541	1014
54.50	K-55	2010	1130	5580	547			10580	1038	10580	1038			853	1660
61.00	K-55	2740	1540	5770	633			10650	1169	10650	1169			962	1871
68.00	K-55	3470	1950	5870	718			10620	1300	10620	1300			1069	2081
68.00	C-75	3950	2220	7400	906			12220	1496	12220	1496			1458	2139
72.00	C-75	4600	2590	7550	978			12330	1598	12330	1598			1558	2284
68.00	N-80	4020	2260	7870	963			12950	1585	12950	1585			1556	2236
72.00	N-80	4750	2670	8020	1040			13060	1693	13060	1693			1661	2388
68.00	L-80	4020	2260	7780	952			12620	1545	12620	1545			1556	2139
72.00	L-80	4750	2670	7940	1029			12730	1650	12730	1650			166[illegible]	2294
68.00	C-95	4120	2320	9100	1114			14480	1772	—	—			184[illegible]	2411
72.00	C-95	5010	2820	9290	1204			14610	1893	—	—			1973	2575
ı 76.60	C-95	5870	3300	9360	1290			14570	2009	—	—			2094	2733
ı 84.50	C-95	7470	4200	9540	1451			14640	2226	—	—			2320	3029
ı 92.50	C-95	9650	5430	9690	1613			14680	2444	—	—			2548	3325
ı100.30	C-95	12110	6810	9810	1771			14040	2534	—	—			2769	3614
ı 72.00	USS95	6170	3470	9380	1215			14930	1935	14930	1935			1973	2638
ı 76.60	USS95	7450	4190	9440	1302			14900	2054	14900	2054			2094	2800
ı 84.50	USS95	10220	5750	9630	1465			14960	2276	14960	2276			2320	3102
ı 92.50	USS95	13210	7430	9780	1629			15010	2499	15010	2499			2548	3406
ı100.30	USS95	15160	8530	9900	1788			14710	2655	15050	2717			2769	3702
ı 72.00	P-110	5140	2890	10820	1402			17140	2221	17140	2221			2284	2887
ı 76.60	P-110	6130	3450	10890	1502			17090	2357	17090	2357			2425	3064
ı 84.50	P-110	8020	4510	11110	1690			17170	2612	17170	2612			2687	3395
ı 92.50	P-110	10130	5700	11280	1878			17230	2868	17230	2868			2950	3728
ı100.30	P-110	13010	7320	11420	2062			16710	3017	17260	3117			3206	4052
ı 92.50	V-150	11400	6410	15190	2529			22790	3795	—	—			4023	4559
ı100.30	V-150	14400	8100	15380	2776			21390	3862	—	—			4372	4955

1. 3J-9 (12A) (Courtesy U.S. Steel)

USS Seamless Casing

Maximum Coupling Performance Properties

Grade	Tension (Kips)			Int. Yield Press. (Psi)		
	Round	Buttress		Round	Buttress	
	Short	Regular	SCC	Short	Regular	SCC
K-55	2345	2293		4600	4490	
C-75	2345	2293		6270	6120	
N-80	2469	2414		6680	6530	
L-80	2345	2293		6680	6530	
C-95	2592	2534		7940	7750	
†USS95	2716	2655		7940	7750	
† P-110	3086	3017		9190	8980	
† V-150	3950	3862		12530	12240	

— Not Available

* Same as Round T&C Pipe

† Next Higher Grade Insufficient for Maximum Pressure of Pipe

Wt/ft	Grade	Minimum Internal Yield Pressure						Mill Test Pressure						80% Minimum Yield Strength Also Maximum Formation Fracturing Pressure					
		Pipe Body	Round	Buttress				PE	Round	Buttress				PE	Round	Buttress			
				Regular		Special Clearance				Regular		Special Clearance				Regular		Special Clearance	
				Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade
48.00	H-40	1730	*	*	*			1200	1200	1200	1200			1600	*	*	*		
54.50	K-55	2730	*	*	*			1900	1900	1900	1900			2500	*	*	*		
61.00	K-55	3090	*	*	*			2100	2100	2100	2100			2800	*	*	*		
68.00	K-55	3450	*	*	*			2400	2400	2400	2400			3200	*	*	*		
68.00	C-75	4710	*	*	*			3000	4300	4300	4300			4300	*	*	*		
72.00	C-75	5040	*	*	*			3000	4600	4600	4600			4600	*	*	*		
68.00	N-80	5020	*	*	*			3000	4600	4600	4600			4600	*	*	*		
72.00	N-80	5380	*	*	*			3000	4900	4900	4900			4900	*	*	*		
68.00	L-80	5020	*	*	*			3000	4600	4600	4600			4600	*	*	*		
72.00	L-80	5380	*	*	*			3000	4900	4900	4900			4900	*	*	*		
68.00	C-95	5970	*	*	—			3000	5500	5500	—			5500	*	*	—		
72.00	C-95	6390	*	*	—			3000	5800	5800	—			5800	*	*	—		
76.60	C-95	6800	*	*	—			3000	6200	6200	—			6200	*	*	—		
84.50	C-95	7570	*	*	—			3000	6400	6200	—			6500	6400	6200	—		
92.50	C-95	8350	7940	7750	—			3000	6400	6200	—			7600	6400	6200	—		
100.30	C-95	9120	7940	7750	—			3000	6400	6200	—			8300	6400	6200	—		
72.00	USS95	6390	*	*	*			3000	5800	5800	5800			5800	*	*	*		
76.60	USS95	6800	*	*	*			3000	6200	6200	6200			6200	*	*	*		
84.50	USS95	7570	*	*	*			3000	6400	6200	6400			6900	6400	6200	*		
92.50	USS95	8350	7940	7750	*			3000	6400	6200	6400			7600	6400	6200	7200		
100.30	USS95	9120	7940	7750	8980			3000	6400	6200	6400			8300	6400	6200	7200		
72.00	P-110	7400	*	*	*			3000	6800	6800	6800			6800	*	*	*		
76.60	P-110	7870	*	*	*			3000	7200	7200	7200			7200	*	*	*		
84.50	P-110	8770	*	*	*			3000	7400	7200	7400			8000	7400	7200	*		
92.50	P-110	9670	9190	8980	*			3000	7400	7200	7400			8800	7400	7200	*		
100.30	P-110	10560	9190	8980	*			3000	7400	7200	7400			9700	7400	7200	*		
92.50	V-150	13190	12530	12240	—			3000	10000	9800	—			12100	10000	9800	—		
100.30	V-150	14410	12530	12240	—			3000	10000	9800	—			13200	10000	9800	—		

Fig. 3J-9 (12B) (Courtesy U.S. Steel)

16" OD

Casing Dimensions

Wt/ft (Lbs)		Casing Body				Areas (Sq. In.)	
Nominal T&C	Plain End	Wall (In.)	ID (In.)	Drift (In.)	D/T Ratio	PE and Buttress	Last Perfect Thread (8Rd)
65.00	62.58	.375	15.250	15.062	42.67	18.408	14.842
75.00	72.80	.438	15.124	14.936	36.53	21.414	17.848
84.00	81.97	.495	15.010	14.822	32.32	24.112	20.546
[1] 109.00	107.50	.656	14.688	14.500	24.39	31.622	28.057
[1] 128.00	126.94	.781	14.438	14.250	20.49	37.341	33.776
[1] 146.00	146.05	.906	14.188	14.000	17.66	42.962	39.396

Coupling Dimensions

OD REG.=17.000 INCHES
LENGTH SHORT= 9.000 INCHES
LENGTH BUTT.=10.675 INCHES
D1 SHORT=15.768 INCHES
D1 BUTT.=15.708 INCHES
AREA SHORT=31.707 SQ.INCHES
AREA BUTTRESS=33.190 SQ.INCHES

Wt/ft	Grade	Collapse		[3]Joint Strength												Plain End Strength	
		[1,2] Feet @ 1.125	Psi	Round				Buttress									
				Short		Long		Regular				Special Clearance				Yield (Min.)	[1] Ultimate (Avg.)
								Same Grade		Higher Grade		Same Grade		Higher Grade			
				[1] Ft @ 1.8	Kips	[1] Ft @ 1.8	Kips	[1] Ft @ 1.8	Kips	[1] Ft @ 1.8	Kips	[1] Ft @ 1.8	Kips	[1] Ft @ 1.8	Kips	Kips	Kips
65.00	H-40	1190	670	3750	439			6680	781	6680	781					736	1381
75.00	K-55	1810	1020	5570	752			9860	1331	9860	1331					1178	2291
84.00	K-55	2510	1410	5720	865			9910	1499	9910	1499					1326	2580
[1] 109.00	K-55	4550	2560	6020	1181			10020	1965	10020	1965					1739	3384
[1] 109.00	C-75	5300	2990	7640	1499			11960	2346	11960	2346					2372	3478
[1] 128.00	C-75	8140	4580	7830	1804			12020	2770	12020	2770					2801	4108
[1] 146.00	C-75	11340	6380	8010	2104			11400	2995	12130	3187					3222	4726
[1] 109.00	N-80	5480	3080	8120	1594			12690	2489	12690	2489					2530	3637
[1] 128.00	N-80	8360	4700	8330	1919			12760	2940	12760	2940					2987	4294
[1] 146.00	N-80	11770	6620	8520	2238			12000	3153	12870	3382					3437	4941

Fig. 3J-9 (13A) (Courtesy U.S. Steel)

USS Seamless Casing

Maximum Coupling Performance Properties

Grade	Tension (Kips)			Int. Yield Press. (Psi)		
	Round	Buttress		Round	Buttress	
	Short	Regular	SCC	Short	Regular	SCC
K-55	2862	2995		3990	4180	
† C-75	2862	2995		5440	5700	
† N-80	3012	3153		5800	6080	

— Not Available

* Same as Round T&C Pipe

† Next Higher Grade Insufficient for Maximum Pressure of Pipe

Wt/ft	Grade	Minimum Internal Yield Pressure						Mill Test Pressure						80% Minimum Yield Strength Also Maximum Formation Fracturing Pressure					
		Pipe Body	Round	Buttress				PE	Round	Buttress				PE	Round	Buttress			
				Regular		Special Clearance				Regular		Special Clearance				Regular		Special Clearance	
				Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade
65.00	H-40	1640	*	*	*			1100	1100	1100	1100			1500	*	*	*		
75.00	K-55	2630	*	*	*			1800	1800	1800	1800			2400	*	*	*		
84.00	K-55	2980	*	*	*			2000	2000	2000	2000			2700	*	*	*		
109.00	K-55	3950	*	*	*			2500	TESTED PLAIN END					3600	3200	3300	*		
109.00	C-75	5380	*	*	*			2500	TESTED PLAIN END					4900	4400	4600	*		
126.00	C-75	6410	5440	5700	*			2500	TESTED PLAIN END					5900	4400	4600	5800		
146.00	C-75	7430	5440	5700	7220			2500	TESTED PLAIN END					6800	4400	4600	5800		
109.00	N-80	5740	*	*	*			2500	TESTED PLAIN END					5200	4600	4900	*		
126.00	N-80	6630	5800	6080	*			2500	TESTED PLAIN END					6200	4600	4900	*		
146.00	N-80	7930	5800	6080	*			2500	TESTED PLAIN END					7200	4600	4900	6700		

Fig. 3J-9 (13B) (Courtesy U.S. Steel)

20" OD

Casing Dimensions

Wt/ft (Lbs)		Casing Body				Areas (Sq. In.)	
Nominal T&C	Plain End	Wall (In.)	ID (In.)	Drift (In.)	D/T Ratio	PE and Buttress	Last Perfect Thread (8Rd)
94.00	91.51	.438	19.124	18.936	45.66	26.918	22.457
106.50	104.13	.500	19.000	18.812	40.00	30.631	26.170
133.00	131.33	.635	18.730	18.542	31.50	38.632	34.171
[1]169.00	166.40	.812	18.376	18.188	24.63	48.948	44.487

Coupling Dimensions

OD REG.=21.000 INCHES

LENGTH SHORT= 9.000 INCHES
LENGTH LONG=11.750 INCHES
LENGTH BUTT.=10.675 INCHES

D1 SHORT=19.768 INCHES
D1 LONG=19.690 INCHES
D1 BUTT.=19.708 INCHES

AREA SHORT=39.448 SQ.INCHES
AREA LONG=41.865 SQ.INCHES
AREA BUTTRESS=41.308 SQ.INCHES

Wt/ft	Grade	Collapse		[3]Joint Strength												Plain End Strength	
		[1, 2] Feet @ 1.125	Psi	Round				Buttress									
				Short		Long		Regular				Special Clearance				Yield (Min.)	[1]Ultimate (Avg.)
								Same Grade		Higher Grade		Same Grade		Higher Grade			
				[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	[1]Ft @ 1.8	Kips	Kips	Kips
94.00	H-40	920	520	3430	581	3940	667	6150	1041	6150	1041					1077	2019
94.00	K-55	920	520	4870	824	5600	947	8740	1479	8740	1479					1480	2880
106.50	K-55	1370	770	5010	960	5760	1104	8780	1683	8780	1683					1685	3278
133.00	K-55	2670	1500	5230	1253	6020	1441	8870	2123	8870	2123					2125	4134
[1]169.00	K-55	4440	2500	5360	1631	6170	1876	8840	2689	8840	2689					2692	5237
[1]169.00	N-80	5370	3020	7300	2222	8380	2548	11980	3645	11980	3645					3916	5629

USS Seamless Casing

Maximum Coupling Performance Properties

Grade	Tension (Kips)			Int. Yield Press. (Psi)		
	Round	Buttress		Round	Buttress	
	Short	Regular	SCC	Short	Regular	SCC
K-55	3560	3728		3230	3380	
†C-75	3560	3728		4400	4610	
†N-80	3748	3924		4690	4920	

— Not Available

* Same as Round T&C Pipe

† Next Higher Grade Insufficient for Maximum Pressure of Pipe

Wt/ft	Grade	Minimum Internal Yield Pressure						Mill Test Pressure						80% Minimum Yield Strength Also Maximum Formation Fracturing Pressure					
		Pipe Body	Round	Buttress				PE	Round	Buttress				PE	Round	Buttress			
				Regular		Special Clearance				Regular		Special Clearance				Regular		Special Clearance	
				Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade			Same Grade	Higher Grade	Same Grade	Higher Grade
94.00	H-40	1530	*	*	*			1100	TESTED PLAIN END					1400	*	*	*		
94.00	K-55	2110	*	*	*			1400	TESTED PLAIN END					1900	*	*	*		
106.50	K-55	2410	*	*	*			1600	TESTED PLAIN END					2200	*	*	*		
133.00	K-55	3060	*	*	*			2100	TESTED PLAIN END					2800	2600	2700	*		
169.00	K-55	3910	3230	3380	*			2500	TESTED PLAIN END					3600	2600	2700	*		
169.00	N-80	5680	4690	4920	*			2500	TESTED PLAIN END					5200	3800	3900	*		

USS Seamless Tubing

API Non-Upset Round Threads

Wt/ft		Dimensions				Steel Grade	Min. Properties				Internal Pressures		
		Tubing			Coupling		Collapse		Tension[4]				
Nominal: T&C	Plain End	Wall Thickness	ID	Drift Diameter	OD		[1, 2] Setting Depth SF-1⅛	Pressure	[1] Setting Depth SF-1½	Yield Load	[3] Mill Test	[3] 80% Min. Yield Strength	Min. Yield Strength
Lb		In					Ft	psi	Ft	Lb	psi		
Size: Outside Diameter—2-3/8 in.													
4.00	3.94	.167	2.041	1.947	2.875	H-40	9300	5230	5020	30130	3000	4500	4920
						J-55	12780	7190	6910	41430	3000	6200	6770
						C-75	16920	9520	9420	56500	8400	8400	9230
						L-80 / N-80	17740	9980	10040	60260	9000	9000	9840
4.60	4.43	.190	1.995	1.901	2.875	H-40	10470	5890	5210	35960	3000	5100	5600
						J-55	14390	8100	7180	49450	3000	7000	7700
						C-75	19630	11040	9770	67430	9600	9600	10500
						L-80 / N-80	20940	11780	10420	71930	10000	10200	11200
						P-105	27480	15460	13680	94410	10000	13400	14700
5.80	5.75	.254	1.867	1.773	2.875	C-75	25470	14330	11100	96560	10000	12000**	14040
						L-80 / N-80	27170	15280	11840	102990	10000	12800**	14970
						P-105	35660	20060	15540	135180	10000	16800**	19650
Size: Outside Diameter—2-7/8 in.													
6.40	6.16	.217	2.441	2.347	3.500	H-40	9920	5580	5500	52780	3000	4800	5280
						J-55	13650	7680	7560	72580	3000	6600	7260
						C-75	18610	10470	10310	98970	9100	9100	9910
						L-80 / N-80	19850	11160	11000	105570	9700	9700	10570
						P-105	24910	14010	14430	138560	10000	12700	13870
8.60	8.44	.308	2.259	2.165	3.500	C-75	25510	14350	11580	149360	10000	12500**	14060
						L-80 / N-80	27210	15300	12350	159310	10000	13300**	15000
						P-105	35710	20090	16210	209100	10000	17500**	19690

**Pressures based on coupling.

Fig. 3J-10 (1A) (Courtesy U.S. Steel)

USS Seamless Tubing

API Non-Upset Round Threads

Wt ft		Dimensions				Steel Grade	Min. Properties				Internal Pressures		
		Tubing			Coupling		Collapse		Tension[4]				
Nominal: T&C	Plain End	Wall Thickness	ID	Drift Diameter	OD		[1, 2] Setting Depth SF-1⅛	Pressure	[1] Setting Depth SF-1½	Yield Load	[3] Mill Test	[3] 80% Min. Yield Strength	Min. Yield Strength
Lb		In					Ft	psi	Ft	Lb	psi		
Size: Outside Diameter—3-1/2 in.													
7.70	7.58	.216	3.068	2.943	4.250	H-40	8240	4630	5630	65070	3000	3900	4320
						J-55	10610	5970	7750	89470	3000	5400	5940
						C-75	13410	7540	10560	122010	7400	7400	8100
						L-80 / N-80	13980	7870	11270	130140	7900	7900	8640
9.20	8.81	.254	2.992	2.867	4.250	H-40	9570	5380	5760	79540	3000	4600	5080
						J-55	13160	7400	7930	109370	3000	6400	6980
						C-75	17840	10040	10810	149140	8700	8700	9520
						L-80 / N-80	18720	10530	11530	159090	9300	9300	10160
						P-105	23190	13050	15130	208800	10000	12200	13330
10.20	9.91	.289	2.922	2.797	4.250	H-40	10770	6060	6050	92550	3000	5300	5780
						J-55	14810	8330	8320	127250	3000	7300	7950
						C-75	20200	11360	11340	173530	9900	9900	10840
						L-80 / N-80	21550	12120	12100	185100	10000	10600	11560
12.70	12.52	.375	2.750	2.625	4.250	C-75	25510	14350	12125	230990	10000	12300**	14060
						L-80 / N-80	27210	15310	12930	246390	10000	13100**	15000
						P-105	35710	20090	16980	323390	10000	17200**	19690
Size: Outside Diameter—4 in.													
9.50	9.11	.226	3.548	3.423	4.750	H-40	7220	4060	5050	72000	3000	3600	3960
						J-55	9090	5110	6950	99010	3000	5000	5440
						C-75	11280	6350	9470	135010	6800	6800	7420
						L-80 / N-80	11710	6590	10110	144010	7200	7200	7910
Size: Outside Diameter—4-1/2 in.													
12.60	12.24	.271	3.958	3.833	5.200	H-40	8000	4500	5520	104360	3000	3900	4220
						J-55	10170	5720	7590	143500	3000	5300	5800
						C-75	12800	7200	10350	195680	7200	7200	7900
						L-80 / N-80	13330	7500	11040	208730	7700	7700	8430

**Pressures based on coupling.

Fig. 3J-10 (1B) (Courtesy U.S. Steel)

USS Seamless Tubing

API External-Upset Round Threads

Wt./ft		Dimensions					Steel Grade	Min. Properties					
		Tubing			Coupling OD			Collapse		Tension[4]			
										Regular Coupling		Special Clearance[5]	
Nominal: T&C	Plain End	Wall Thickness	ID	Drift Diameter	Regular	Special Clearance		[1,2] Setting Depth SF-1⅛	Pressure	[1] Setting Depth SF-1½	Yield Load	[1] Setting Depth SF-1½	Yield Load
Lb		In						Ft	psi	Ft	Lb	Ft	Lb
Size: Outside Diameter—2-3/8 in.													
4.70	4.43	.190	1.995	1.901	3.063	2.910	H-40	10470	5890	7400	52170	7400	52170
							J-55	14390	8100	10170	71730	10170	71730
							C-75	19630	11040	13880	97820	13880	97820
							L-80, N-80	20940	11780	14800	104340	14800	104340
							P-105	27480	15460	19420	136940	19420	136940
5.95	5.75	.254	1.867	1.773	3.063	2.910	C-75	25470	14330	14220	126940	14220	126904
							L-80, N-80	27170	15280	15170	135400	15170	135400
							P-105	35660	20060	19910	177710	19910	177710
Size: Outside Diameter—2-7/8 in.													
6.50	6.16	.217	2.441	2.347	3.668	3.460	H-40	9920	5580	7430	72480	7430	72480
							J-55	13650	7680	10220	99660	10220	99660
							C-75	18610	10470	13940	135900	13940	135900
							L-80, N-80	19850	11160	14870	144960	14870	144960
							P-105	24910	14010	19510	190260	19510	190260
8.70	8.44	.308	2.259	2.165	3.668	3.460	C-75	25510	14350	14280	186290	13460	175600
							L-80, N-80	27210	15300	15230	198710	14350	187310
							P-105	35710	20090	19990	260810	18840	245850
Size: Outside Diameter—3-1/2 in.													
9.30	8.81	.254	2.992	2.867	4.500	4.180	H-40	9570	5380	7430	103610	7430	103610
							J-55	13160	7400	10210	142460	10210	142460
							C-75	17840	10040	13930	194260	13930	194260
							L-80, N-80	18720	10530	14860	207220	14860	207220
							P-105	23190	13050	19500	271970	19500	271970
12.95	12.52	.375	2.750	2.625	4.500	4.180	C-75	25510	14350	14210	276120	12830	249260
							L-80, N-80	27210	15310	15160	294530	13890	265870
							P-105	35710	20090	19900	386570	17960	348960

Fig. 3J-10 (2A) (Courtesy U.S. Steel)

Wt /ft	Steel Grade	Internal Pressures *Indicates same pressure as pipe. —Not available.											
		Mill Test[3]				80% Min. Yield Strength[3]				Min. Yield Strength			
Nominal: T&C		Pipe	Regular Coupling	Special Clearance Coupling		Pipe	Regular Coupling	Special Clearance Coupling		Pipe	Regular Coupling	Special Clearance Coupling[5]	
				Same Grade	Higher Grade			Same Grade	Higher Grade			Same Grade	Higher Grade
Lb		psi											
4.70	H-40	3000	*	*	*	5100	*	4600	*	5600	*	*	*
	J-55	3000	*	*	*	7000	*	6300	*	7700	*	*	*
	C-75	9600	*	8600	—	9600	*	8600	—	10500	*	*	—
	L-80, N-80	10000	*	9100	*	10200	*	9100	*	11200	*	*	*
	P-105	10000	*	*	—	13400	*	12000	—	14700	*	*	—
5.95	C-75	10000	*	8600	—	12800	11200	8600	—	14040	13960	10720	—
	L-80, N-80	10000	*	9100	*	13700	11900	9100	12000	14970	14890	11440	15010
	P-105	10000	*	*	—	18000	15600	12000	—	19650	19540	15010	—
6.50	H-40	3000	*	*	*	4800	*	4400	*	5280	*	*	*
	J-55	3000	*	*	*	6600	*	6100	*	7260	*	*	*
	C-75	9100	*	8300	—	9100	*	8300	—	9910	*	*	—
	L-80, N-80	9700	*	8800	*	9700	*	8800	*	10570	*	*	*
	P-105	10000	*	*	—	12700	*	11600	—	13870	*	*	—
8.70	C-75	10000	*	8300	—	12900	11200	8300	—	14060	14010	10340	—
	L-80, N-80	10000	*	8800	*	13700	12000	8800	11600	15000	14940	11030	14480
	P-105	10000	*	*	—	18000	15700	11600	—	19690	19610	14480	—
9.30	H-40	3000	*	*	*	4600	*	4300	*	5080	*	*	*
	J-55	3000	*	*	*	6400	*	5900	*	6980	*	*	*
	C-75	8700	*	8000	—	8700	*	8000	—	9520	*	*	—
	L-80, N-80	9300	*	8500	*	9300	*	8500	*	10160	*	*	*
	P-105	10000	*	*	—	12200	*	11200	—	13330	*	*	—
12.95	C-75	10000	*	8000	—	12900	11700	8000	—	14060	*	9990	—
	L-80, N-80	10000	*	8500	*	13700	12500	8500	11200	15000	*	10660	13990
	P-105	10000	*	*	—	18000	16400	11200	—	19690	*	13990	—

Fig. 3J-10 (2B) (Courtesy U.S. Steel)

USS Seamless Tubing

API External-Upset Round Threads

Wt ft		Dimensions					Steel Grade	Min. Properties					
		Tubing			Coupling OD			Collapse		Tension[4]			
										Regular Coupling		Special Clearance[5]	
Nominal: T&C	Plain End	Wall Thickness	ID	Drift Diameter	Regular	Special Clearance		[1, 2] Setting Depth SF-1⅛	Pressure	[1] Setting Depth SF-1½	Yield Load	[1] Setting Depth SF-1½	Yield Load
Lb		In						Ft	psi	Ft	Lb	Ft	Lb
Size: Outside Diameter—4 in.													
11.00	10.46	.262	3.476	3.351	5.000	—	H-40	8710	4900	7460	123070	—	—
							J-55	11720	6590	10260	169220	—	—
							C-75	14960	8410	13980	230750	—	—
							L-80, N-80	15640	8800	14920	246140	—	—
Size: Outside Diameter—4-1/2 in.													
12.75	12.24	.271	3.958	3.833	5.563	—	H-40	8000	4500	7530	144020	—	—
							J-55	10170	5720	10350	198030	—	—
							C-75	12800	7200	14120	270040	—	—
							L-80, N-80	13330	7500	15060	288040	—	—

Fig. 3J-10 (3A) (Courtesy U.S. Steel)

Wt ft Nominal: T&C Lb	Steel Grade	Internal Pressures *Indicates same pressure as pipe. —Not available.											
		Mill Test[3]				80% Min. Yield Strength[3]				Min. Yield Strength			
		Pipe	Regular Coupling	Special Clearance Coupling		Pipe	Regular Coupling	Special Clearance Coupling		Pipe	Regular Coupling	Special Clearance Coupling[5]	
				Same Grade	Higher Grade			Same Grade	Higher Grade			Same Grade	Higher Grade
		psi											
11.00	H-40	3000	*	—	—	4200	*	—	—	4580	*	—	—
	J-55	3000	*	—	—	5800	*	—	—	6300	*	—	—
	C-75	7900	*	—	—	7900	*	—	—	8600	*	—	—
	L-80 / N-80	8400	*	—	—	8400	*	—	—	9170	*	—	—
12.75	H-40	3000	*	—	—	3900	*	—	—	4220	*	—	—
	J-55	3000	*	—	—	5300	*	—	—	5800	*	—	—
	C-75	7200	*	—	—	7200	*	—	—	7900	*	—	—
	L-80 / N-80	7700	*	—	—	7700	*	—	—	8430	*	—	—

Fig. 3J-10 (3B) (Courtesy U.S. Steel)

USS Seamless Tubing

Buttress Thread

Wt/ft		Dimensions					Steel Grade	Min. Properties					
		Tubing			Coupling OD			Collapse		Tension[4]			
										Regular Coupling		Special Clearance[5]	
Nominal: T&C	Plain End	Wall Thickness	ID	Drift Diameter	Regular	Special Clearance		[1,2] Setting Depth SF-1⅛	Pressure	[1] Setting Depth SF-1½	Yield Load	[1] Setting Depth SF-1½	Yield Load
Lb		In						Ft	psi	Ft	Lb	Ft	Lb
Size: Outside Diameter—2-3/8 in.													
4.60	4.43	.190	1.995	1.901	2.875	2.700	J-55	14390	8100	10400	71730	10400	71730
							C-75	19630	11040	14180	97820	14180	97820
							L-80, N-80	20940	11780	15120	104340	15120	104340
							P-105	27480	15460	19840	136940	19840	136940
5.80	5.75	.254	1.867	1.773	2.875	2.700	J-55	18680	10510	10700	93090	10700	93090
							C-75	25470	14330	14590	126940	14590	126940
							L-80, N-80	27170	15280	15560	135400	15560	135400
							P-105	35660	20060	20430	177710	20430	177710
Size: Outside Diameter—2-7/8 in.													
6.40	6.16	.217	2.441	2.347	3.500	3.220	J-55	13650	7680	10380	99660	10380	99660
							C-75	18610	10470	14160	135900	14160	135900
							L-80, N-80	19850	11160	15100	144960	15100	144960
							P-105	24910	14010	19320	190260	19820	190260
7.90	7.66	.276	2.323	2.229	3.500	3.220	J-55	16970	9550	10460	123940	9980	118280
							C-75	23140	13020	14260	169010	13610	161290
							L-80, N-80	24690	13890	15210	180280	14520	172040
							P-105	32400	18220	19970	236620	19050	225800
8.60	8.44	.308	2.259	2.165	3.500	3.220	J-55	18710	10520	10590	136610	9170	118230
							C-75	25510	14350	14440	186290	12500	161290
							L-80, N-80	27210	15300	15400	198710	13340	172040
							P-105	35710	20090	20220	260810	17500	225800

Fig. 3J-10 (4A) (Courtesy U.S. Steel)

Wt /ft Nominal: T&C	Steel Grade	Internal Pressures *Indicates same pressure as pipe. —Not available.											
		Mill Test[3]				80% Min. Yield Strength[3]				Min. Yield Strength			
		Pipe	Regular Coupling	Special Clearance Coupling		Pipe	Regular Coupling	Special Clearance Coupling		Pipe	Regular Coupling	Special Clearance Coupling[5]	
				Same Grade	Higher Grade			Same Grade	Higher Grade			Same Grade	Higher Grade
Lb		psi											
4.60	J-55	3000	*	*	*	7000	*	*	*	7700	*	*	*
	C-75	9600	*	*	—	9600	*	*	—	10500	*	*	—
	L-80, N-80	10000	*	*	*	10200	*	*	*	11200	*	*	*
	P-105	10000	*	*	—	13400	*	*	—	14700	*	*	—
5.80	J-55	3000	*	*	*	9400	*	7400	*	10290	*	9310	*
	C-75	10000	*	*	—	12800	*	10200	—	14040	*	12690	—
	L-80, N-80	10000	*	*	*	13700	*	10800	*	14970	*	13540	*
	P-105	10000	*	*	—	18000	*	14200	—	19650	*	17770	—
6.40	J-55	3000	*	*	*	6600	*	6500	*	7260	*	*	*
	C-75	9100	*	8900	—	9100	*	8900	—	9910	*	*	—
	L-80, N-80	9700	*	9500	*	9700	*	9500	*	10570	*	*	*
	P-105	10000	*	*	—	12700	*	12400	—	13870	*	*	—
7.90	J-55	3000	*	*	*	8400	*	6500	*	9240	*	8150	*
	C-75	10000	*	8900	—	11500	*	8900	—	12600	*	11110	—
	L-80, N-80	10000	*	9500	*	12300	*	9500	*	13440	*	11850	*
	P-105	10000	*	*	—	16100	*	12400	—	17640	*	15550	—
8.60	J-55	3000	*	*	*	9400	*	6500	*	10310	*	8150	*
	C-75	10000	*	8900	—	12900	*	8900	—	14060	*	11110	—
	L-80, N-80	10000	*	9500	*	13700	*	9500	12400	15000	*	11850	*
	P-105	10000	*	*	—	18000	*	12400	—	19690	*	15550	—

Fig. 3J-10 (4B) (Courtesy U.S. Steel)

USS Seamless Tubing

Buttress Thread

Wt/ft		Dimensions					Steel Grade	Min. Properties					
		Tubing			Coupling OD			Collapse		Tension[4]			
										Regular Coupling		Special Clearance[5]	
Nominal: T&C	Plain End	Wall Thick-ness	ID	Drift Diameter	Regular	Special Clearance		[1, 2] Setting Depth SF-1⅛	Pressure	[1] Setting Depth SF-1½	Yield Load	[1] Setting Depth SF-1½	Yield Load
Lb		In						Ft	psi	Ft	Lb	Ft	Lb
Size: Outside Diameter—3-1/2 in.													
9.20	8.81	.254	2.992	2.867	4.250	3.865	J-55	13160	7400	10320	142460	10320	142460
							C-75	17840	10040	14080	194260	14080	194260
							L-80 } N-80	18720	10530	15020	207220	15020	207220
							P-105	23190	13050	19710	271970	19710	271970
10.20	9.91	.289	2.922	2.797	4.250	3.865	J-55	14810	8330	10480	160340	9780	149660
							C-75	20200	11360	14290	218650	13340	204080
							L-80 } N-80	21550	12120	15240	233220	14230	217690
							P-105	28280	15910	20010	306110	18670	285720
11.90	14.63	.449	2.602	2.477	4.250	—	J-55	21870	12300	10590	236700	—	—
							C-75	29820	16770	14440	322780	—	—
							L-80 } N-80	31810	17890	15400	344300	—	—
							P-105	41750	23480	20220	451890	—	—
Size: Outside Diameter—4 in.													
10.90	10.46	.262	3.476	3.351	4.750	4.400	J-55	11720	6590	10350	169220	10350	169220
							C-75	14960	8410	14110	230750	14110	230750
							L-80 } N-80	15640	8800	15050	246140	15050	246140
							P-105	19020	10700	19760	323050	19760	323050
Size: Outside Diameter—4-1/2 in.													
12.60	12.24	.271	3.958	3.833	5.200	4.920	J-55	10170	5720	10480	198030	10480	198030
							C-75	12800	7200	14290	270040	14290	270040
							L-80 } N-80	13330	7500	15240	288040	15240	288040
							P-105	15910	8950	20000	378050	20000	378050

Fig. 3J-10 (5A) (Courtesy U.S. Steel)

Internal Pressures *Indicates same pressure as pipe. —Not available.

Wt/ft Nominal: T&C (Lb)	Steel Grade	Mill Test[3] Pipe	Mill Test[3] Regular Coupling	Mill Test[3] Special Clearance Coupling: Same Grade	Mill Test[3] Special Clearance Coupling: Higher Grade	80% Min. Yield Strength[3] Pipe	80% Min. Yield Strength[3] Regular Coupling	80% Min. Yield Strength[3] Special Clearance Coupling: Same Grade	80% Min. Yield Strength[3] Special Clearance Coupling: Higher Grade	Min. Yield Strength Pipe	Min. Yield Strength Regular Coupling	Min. Yield Strength Special Clearance Coupling[5]: Same Grade	Min. Yield Strength Special Clearance Coupling[5]: Higher Grade
		psi											
9.20	J-55	3000	*	*	*	6400	*	5700	*	6980	*	*	*
	C-75	8700	*	7700	—	8700	*	7700	—	9520	*	*	—
	L-80, N-80	9300	*	8200	*	9300	*	8200	*	10160	*	*	*
	P-105	10000	*	*	—	12200	*	10800	—	13330	*	*	—
10.20	J-55	3000	*	*	*	7300	*	5700	*	7950	*	7070	*
	C-75	9900	*	7700	—	9900	*	7700	—	10840	*	9640	—
	L-80, N-80	10000	*	8200	*	10600	*	8200	*	11560	*	10290	*
	P-105	10000	*	*	—	13900	*	10800	—	15170	*	13500	—
14.90	J-55	3000	*	*	*	11300	9100	5700	8200	12350	11410	—	—
	C-75	10000	*	7700	—	15400	12500	7700	—	16840	15560	—	—
	L-80, N-80	10000	*	8200	*	16400	13300	8200	10800	17960	16600	—	—
	P-105	10000	*	*	—	21600	17400	10800	—	23570	21790	—	—
10.90	J-55	3000	*	*	*	5800	*	5500	*	6300	*	*	*
	C-75	7900	*	7500	—	7900	*	7500	—	8600	*	*	—
	L-80, N-80	8400	*	8000	*	8400	*	8000	*	9170	*	*	*
	P-105	10000	*	*	—	11000	*	10500	—	12040	*	*	—
12.60	J-55	3000	*	*	*	5300	*	5200	*	5800	*	*	*
	C-75	7200	*	7100	—	7200	*	7100	—	7900	*	*	—
	L-80, N-80	7700	*	7600	*	7700	*	7600	*	8430	*	*	*
	P-105	10000	*	*	—	10100	*	10000	—	11070	*	*	—

Fig. 3J-10 (5B) (Courtesy U.S. Steel)

USS Seamless Internal Upset Drill Pipe

API Upset for Attachment of Weld-On Tool Joint

Wt/ft			Dimensions		Steel Grade	Min. Properties			
[2] Nominal: T&C	Plain End (non upset)	Calculated Weight Upset	Pipe			Internal Pressures		[3] Tension	Collapse
			Wall Thickness	ID		Yield	Ultimate[1]	Yield Load	Pressure
Lb			In.			psi		Lb	psi
Size: Outside Diameter—2⅞ in.									
10.40	9.72	9.88 9.99	.362	2.151	D E X-95 G-105 S-135	12120 16530 20930 23140 29750	23920 25180 31480 34000 36510	157180 214340 271500 300080 385820	12110 16510 20910 23110 29720
Size: Outside Diameter—3½ in.									
9.50	8.81	9.03	.254	2.992	E D	9520 10120	14510 19980	194260 199150	10040 10350
13.30	12.31	12.53 12.68	.368	2.764	E X-95 G-105 S-135	13800 17480 19320 24840	21030 26290 28390 29990	271580 343990 380190 488810	14110 17880 19760 24920
15.50	14.63	14.80	.449	2.602	D E	12350 16840	24370 25660	236700 322780	12300 16770
Size: Outside Diameter—4 in.									
14.00	12.93	13.16 13.37	.330	3.340	D E X-95 G-105 S-135	7940 10830 13720 15160 19490	15680 16500 20630 22280 23930	209260 285360 361460 399500 513650	8330 11350 14380 15900 20170
Size: Outside Diameter—4½ in.									
13.75	12.24	12.50	.271	3.958	E	7900	12040	270040	7200
Size: Outside Diameter—5 in.									
16.25	14.87	15.55	.296	4.408	X-95 G-105 S-135	9840 10880 13990	14800 15980 17170	415560 459300 590530	8090 8610 9860

Fig. 3J-11 (1) (Courtesy U.S. Steel)

USS Seamless External Upset Drill Pipe

API Upset for Attachment of Weld-On Tool Joint

Wt./ft			Dimensions		Steel Grade	Min. Properties			
[2] Nominal: T&C	Plain End (non upset)	Calculated Weight Upset	Pipe			Internal Pressures		[3] Tension	Collapse
			Wall Thickness	ID		Yield	Ultimate[1]	Yield Load	Pressure
Lb			In.			psi		Lb	psi
Size: Outside Diameter—2⅜ in.									
6.65	6.26	6.35	.280	1.815	D	11350	22400	101360	11440
					E	15470	23580	138220	15600
		6.49			X-95	19600	29470	175080	19760
					G-105	21660	31830	193500	21840
					S-135	27850	34190	248790	28080
Size: Outside Diameter—2⅞ in.									
10.40	9.72	9.84	.362	2.151	D	12120	23920	157180	12110
					E	16530	25180	214340	16510
		10.03			X-95	20930	31480	271500	20910
					G-105	23140	34000	300080	23110
					S-135	29750	36510	385820	29720
Size: Outside Diameter—3½ in.									
9.50	8.81	8.94	.254	2.992	E	9520	14510	194250	10040
13.30	12.31	12.51	.368	2.764	D	10120	19980	199150	10350
					E	13800	21030	271570	14110
		12.82			X-95	17480	26290	343990	17880
					G-105	19320	28390	380190	19760
					S-135	24840	29990	488840	24920
15.50	14.63	14.77	.449	2.602	D	12350	24370	236700	12300
					E	16840	25660	322780	16770
		15.04			X-95	21330	32070	408850	21250
					G-105	23570	34640	451890	23480
					S-135	30310	37200	581040	30180
Size: Outside Diameter—4 in.									
14.00	12.93	13.18	.330	3.340	D	7940	15680	209280	8330
					E	10830	16500	285380	11350
		13.65			X-95	13720	20630	361460	14380
					G-105	15160	22280	399500	15900
					S-135	19490	23930	513650	20170
Size: Outside Diameter—4½ in.									
13.75	12.24	12.52	.271	3.958	E	7900	12040	270000	7200
16.60	14.98	15.26	.337	3.826	D	7210	14230	242410	7620
					E	9830	14980	330560	10390
		15.84			X-95	12450	18720	418700	12750
					G-105	13760	20220	522320	16420
					S-135	17690	21720	594940	16880
20.00	18.69	18.97	.430	3.640	D	9200	18160	302400	9510
					E	12540	19110	412360	12960
		19.49			X-95	15890	23890	522320	16420
					G-105	17560	25800	577300	18150
					S-135	22580	27710	742240	23330
Size: Outside Diameter—5 in.									
19.50	17.93	18.77	.362	4.276	X-95	12040	18100	501040	12010
					G-105	13300	19550	553830	12990
					S-135	17100	21000	712070	15000
25.60	24.03	24.80	.500	4.000	X-95	16620	25000	671520	17100
					G-105	18380	27000	742200	18900
					S-135	23630	29000	954260	24300

Fig. 3J-11 (2) (Courtesy U.S. Steel)

USS Seamless Internal-External Upset Drill Pipe

API Upset for Attachment of Weld-On Tool Joint									
Wt/ft			Dimensions		Steel Grade	Min. Properties			
			Pipe			Internal Pressures		[3] Tension	Collapse
[2] Nominal: T&C	Plain End (non upset)	Calculated Weight Upset	Wall Thickness	ID		Yield	Ultimate[1]	Yield Load	Pressure
Lb			In.			psi		Lb	psi
Size: Outside Diameter—3½ in.									
15.50	14.63	15.18	.449	2.602	X-95	21330	32070	408850	21250
					G-105	23570	34640	451890	23480
					S-135	30310	37200	581000	30190
Size: Outside Diameter—4½ in.									
16.60	14.98	15.39	.337	3.826	D	7210	14230	242410	7620
					E	9830	14980	330560	10390
		15.42			X-95	12450	18720	418700	12750
					G-105	13760	20220	462780	13820
					S-135	17690	21720	577300	16800
20.00	18.69	19.12	.430	3.640	D	9200	18160	302400	9510
					E	12540	19110	412360	12960
		19.57			X-95	15890	23890	522320	16420
					G-105	17560	25800	577300	18150
					S-135	22580	27710	742240	23320
Size: Outside Diameter—5 in.									
19.50	17.93	18.36	.362	4.276	D	6970	13760	290100	7390
					E	9500	14480	395600	10000
		18.77			X-95	12040	18100	501090	12010
					G-105	13300	19550	553830	12990
					S-135	17100	21000	712070	15000
25.60	24.03	24.42	.500	4.000	D	9630	19000	388800	9900
					E	13130	20000	530190	13500
		24.80			X-95	16620	25000	671520	17100
					G-105	18380	27000	742200	18900
					S-135	23630	29000	954260	24300
Size: Outside Diameter—5½ in.									
21.90	19.81	20.34	.361	4.778	D	6320	12470	320550	6610
					E	8610	13130	437120	8440
		20.86			X-95	10910	16410	553680	10000
					G-105	12060	17720	611960	10740
					S-135	15510	19030	786610	12710
24.70	22.54	22.99	.415	4.670	D	7260	14340	364630	7670
					E	9900	15090	497220	10460
		23.46			X-95	12540	18860	629810	12920
					G-105	13860	20370	696110	14000
					S-135	17830	21880	895000	17050
Size: Outside Diameter—6⅝ in.									
25.20	22.19	—	.330	5.965	D	4790	9460	358930	4010
					E	6540	9960	489450	4810

Fig. 3J-11 (3) (Courtesy U.S. Steel)

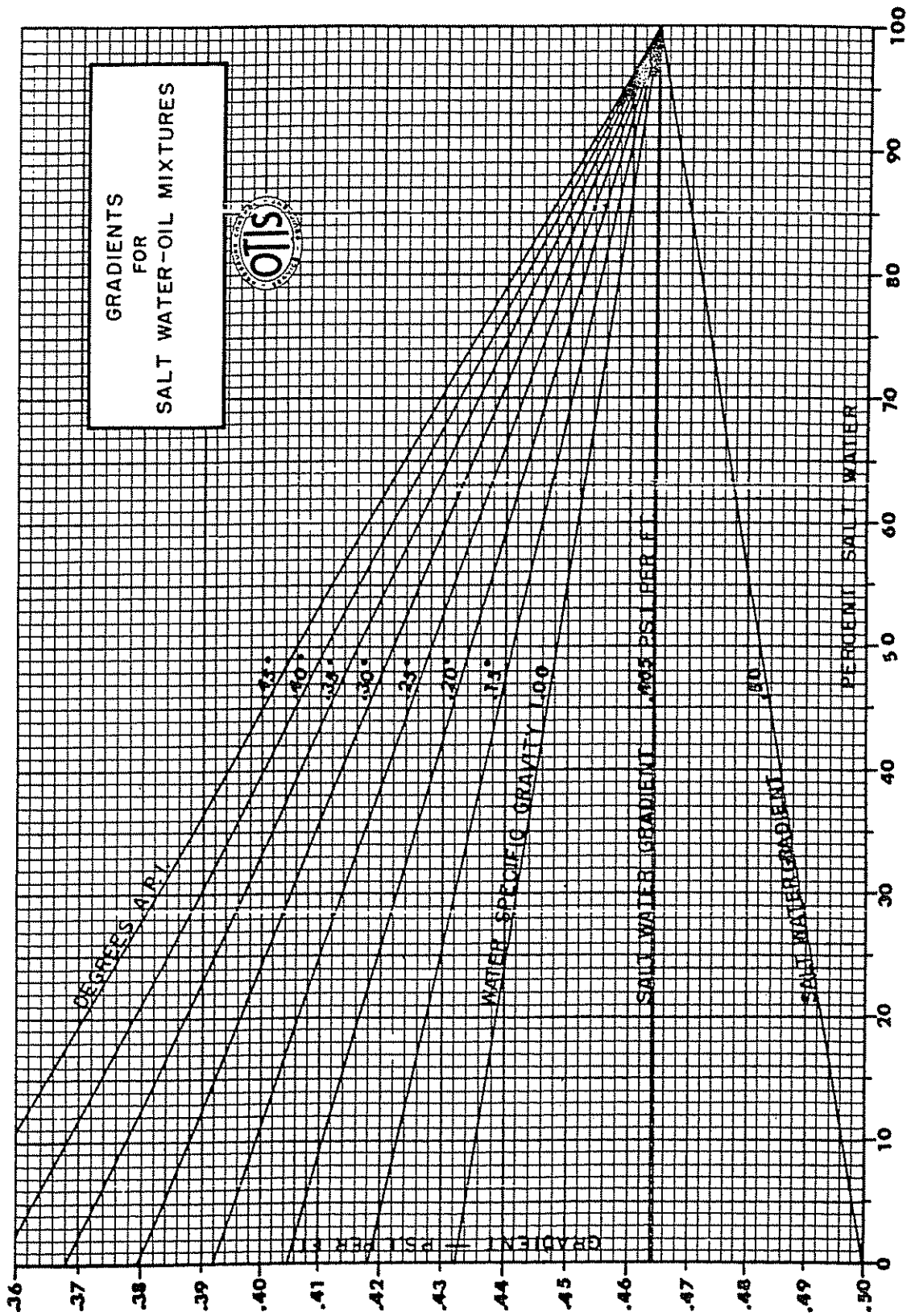


Fig. 3K (Courtesy Otis Engineering Corp.)

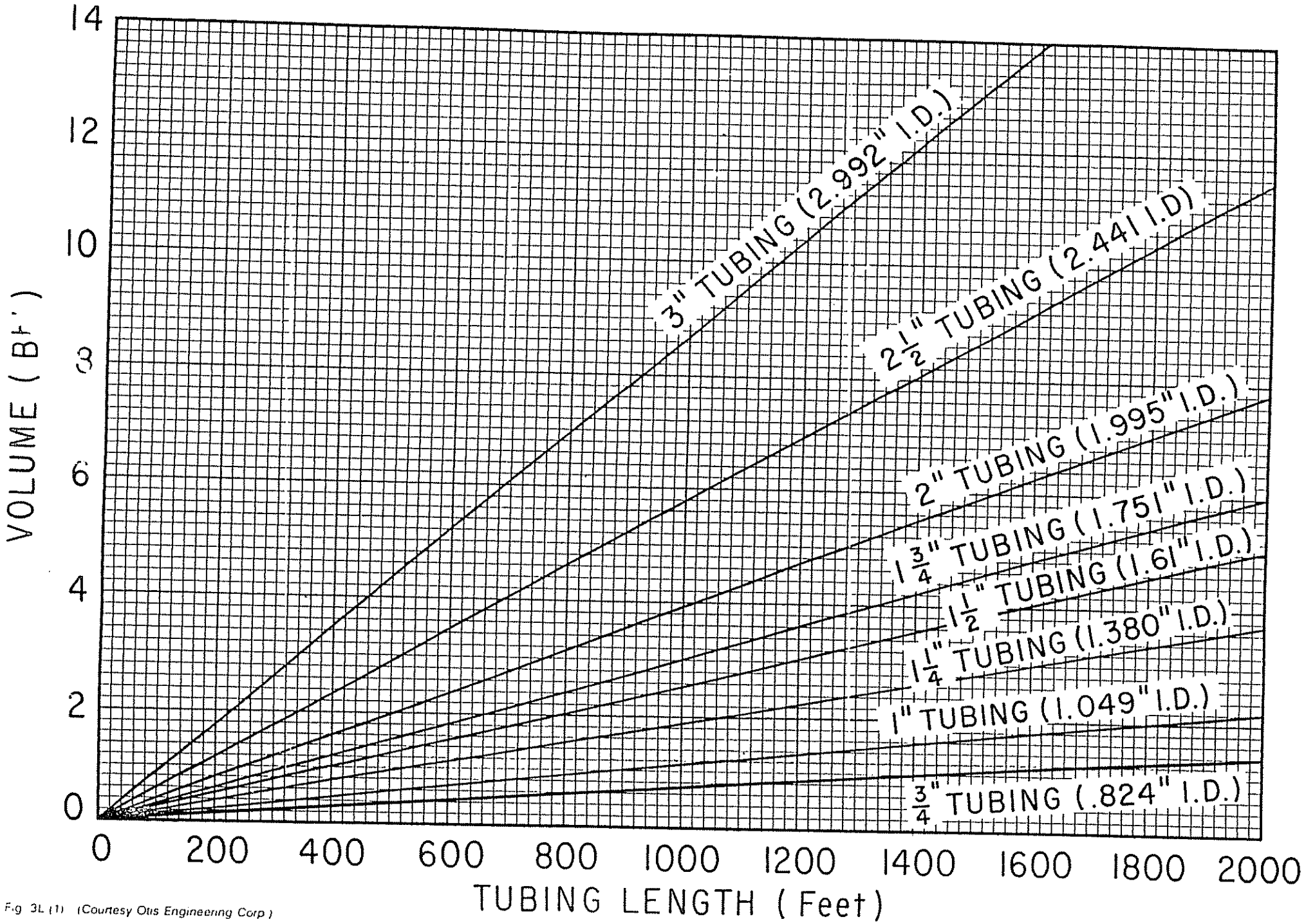


Fig 3L (1) (Courtesy Otis Engineering Corp)

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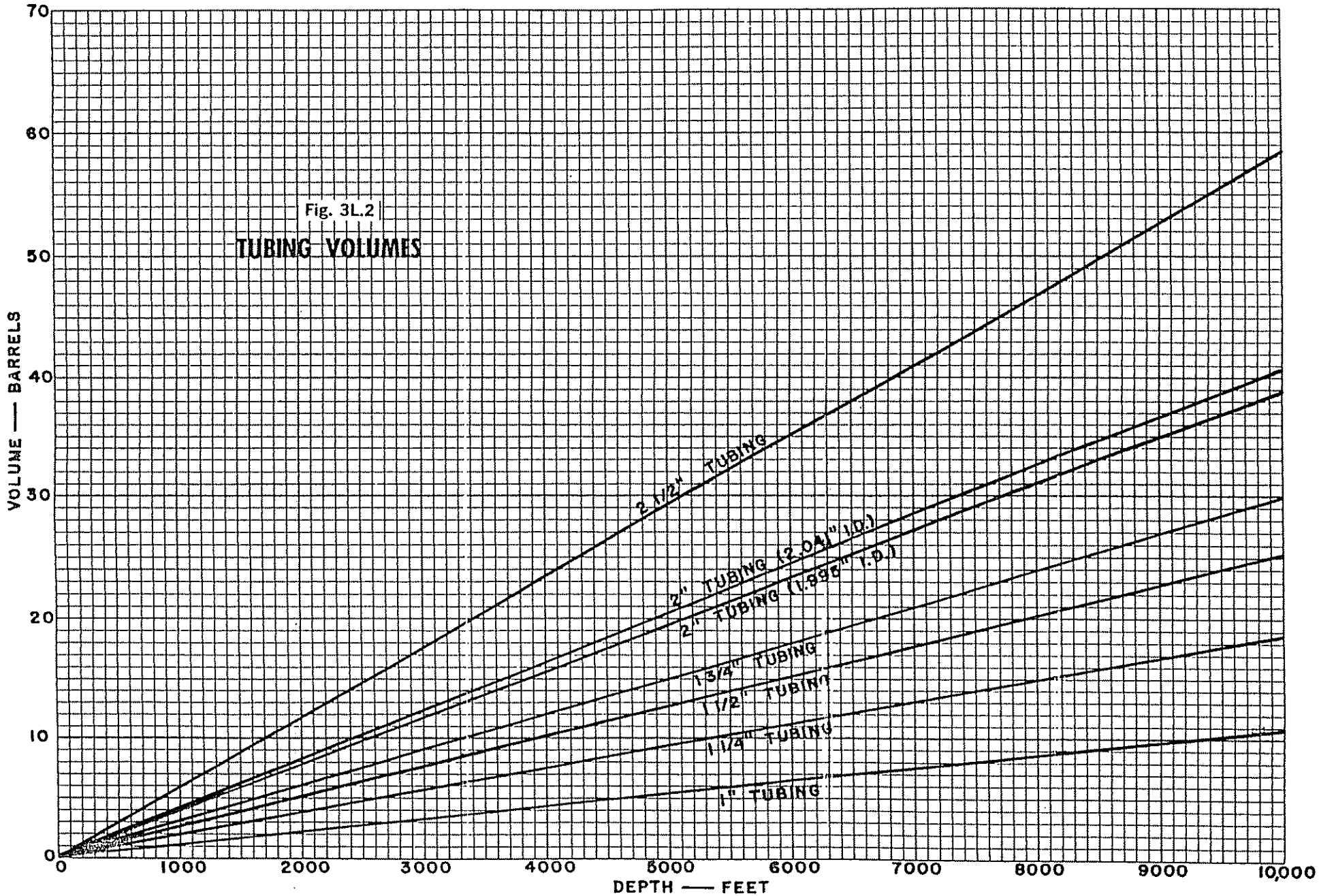


Fig. 3L (2) (Courtesy Otis Engineering Corp.)

M.1 CONVERSION FACTORS (Reproduced through courtesy of Atlantic Richfield Research Department)

In the following tables, the conversion factors to five significant figures are given for the following items: area, density, diffusivity, distance, energy, force, mass, permeability, pressure, power, rate (volume), specific gravity, temperature, time (units conversion), time (table of day number in year), velocity, viscosity (dynamic), viscosity (kinematic), and volume.

$x \times y = z$

Multiply ↘				z		
	to obtain ↑					
x by ——→				y		

Use of Conversion Tables

In conversion tables an * indicates the exact conversion factor. Other factors are exact only to the fifth significant figure given.

Area (L^2)

Multiply ↘	square centimeters	square meters	square inches	square feet	acres	square miles
	to obtain ↑					
square centimeters by ——→	—	10^{-4}*	1.5500×10^{-1}	1.0764×10^{-3}	2.4710×10^{-8}	3.8610×10^{-11}
square meters by ——→	10^{4}*	—	1.5500×10^{3}	1.0764×10	2.4710×10^{-4}	3.8610×10^{-7}
square inch by ——→	6.4516	6.4516×10^{-4}	—	6.9444×10^{-3}	1.5942×10^{-7}	2.4910×10^{-10}
square feet by ——→	9.2903×10^{2}	9.2903×10^{-2}	144*	—	2.2957×10^{-5}	3.5870×10^{-8}
acres by ——→	4.0469×10^{7}	4.0469×10^{3}	6.2726×10^{6}	43,560*	—	1.5625×10^{-3}*
square miles by ——→	2.5900×10^{10}	2.5900×10^{6}	4.0145×10^{9}	2.7878×10^{7}	640*	—

Density (mL^{-3})

Multiply ↘	pounds per cubic inch	pounds per cubic foot	pounds per gallon	grams per cubic centimeter	kilograms per cubic meter	pounds per barrel
	to obtain ↑					
pounds per cubic inch by →	—	1728*	231*	2.7680×10	2.7680×10^{4}	9702*
pounds per cubic foot by →	5.7870×10^{-4}	—	1.3368×10^{-1}	1.6018×10^{-2}	1.6018×10	5.6146
pounds per gallon by ——→	4.3290×10^{-3}	7.4805	—	1.1983×10^{-1}	1.1983×10^{2}	42*
grams per cubic centimeter by ——→	3.6127×10^{-2}	6.2428×10	8.3454	—	1000*	3.5051×10^{2}
kilograms per cubic meter by ——→	3.6128×10^{-5}	6.2428×10^{-2}	8.3454×10^{-3}	10^{-3}*	—	3.5051×10^{-1}
pounds per barrel by ——→	1.0307×10^{-4}	1.7811×10^{-1}	2.3810×10^{-2}	2.8530×10^{-3}	2.8530	—

Fig. 3M (1)

Diffusivity (L^2t^{-1})

Multiply ↘	md-psi / cp	square feet per second	square centimeters per second	square meters per second
	to obtain →			
millidarcies-psi / centipoise by ——→	—	7.3244×10^{-8}	6.8046×10^{-5}	6.8046×10^{-9}
square feet per second by →	1.3653×10^{7}	—	9.2903×10^{2}	9.2903×10^{-2}
square centimeters per second by ——→	1.4696×10^{5}	1.0764×10^{-3}	—	10^{-4}*
square meters per second by→	1.4696×10^{9}	1.0764×10	10^{4}*	—

Distance (L)

Multiply ↘	centimeters	meters	inches	feet (U.S.)	yards (U.S.)	miles (U.S.)	kilometers
	to obtain →						
centimeters by →	—	10^{-2}*	3.937×10^{-1}*	3.2808×10^{-2}	1.0936×10^{-2}	6.2137×10^{-6}	10^{-5}*
meters by ——→	100*	—	3.937×10^{1}*	3.2808	1.0936	6.2137×10^{-4}	10^{-3}*
inches by ——→	2.5400	2.5400×10^{-2}	—	8.3333×10^{-2}	2.7778×10^{-2}	1.5783×10^{-5}	2.5400×10^{-5}
feet (U.S.) by —→	3.0480×10	3.0480×10^{-1}	12*	—	3.3333×10^{-1}	1.8939×10^{-4}	3.0480×10^{-4}
yards (U.S.) by →	9.1440×10	9.1440×10^{-1}	36*	3*	—	5.6818×10^{-4}	9.1440×10^{-4}
miles (statute U.S.) by ——→	1.6093×10^{5}	1.6093×10^{3}	63,360*	5280*	1,760*	—	1.6093
kilometers by —→	10^{5}*	10^{3}*	3.937×10^{4}	3.2808×10^{3}	1.0936×10^{3}	6.2137×10^{-1}	—

Energy (mL^2t^{-2})

Multiply ↘	foot-pounds	foot-poundals	horsepower-hours	kilowatt-hours	BTUs (mean)	kilogram-calories(mean)	ergs	joules (abs)
	to obtain →							
foot-pounds by —→	—	3.2174×10	5.0505×10^{-7}	3.7662×10^{-7}	1.2353×10^{-3}	3.2389×10^{-4}	1.3558×10^{7}	1.3558
foot-poundals by →	3.1081×10^{-2}	—	1.5698×10^{-8}	1.1706×10^{-8}	3.9948×10^{-5}	1.0067×10^{-5}	4.2140×10^{5}	4.2140×10^{-2}
horsepower-hours by ——→	1.98×10^{6}*	6.3705×10^{7}	—	7.4570×10^{-1}	2.5449×10^{3}	6.4130×10^{2}	2.6845×10^{13}	2.6845×10^{6}
kilowatt-hours by →	2.6552×10^{6}	8.5429×10^{7}	1.3410	—	3.4128×10^{3}	860*	3.6×10^{13}*	3.6×10^{6}*
BTUs (mean) by →	7.7803×10^{2}	2.5032×10^{4}	3.9294×10^{-4}	2.9302×10^{-4}	—	2.5200×10^{-1}	1.0549×10^{10}	1.0549×10^{3}
kilogram-calories (mean) by ——→	3.0875×10^{3}	9.9336×10^{4}	1.5593×10^{-3}	1.1628×10^{-3}	3.9683	—	4.1861×10^{10}	4.1861×10^{3}
ergs by ——→	7.3756×10^{-8}	2.3730×10^{-6}	3.7251×10^{-14}	2.7778×10^{-14}	9.4799×10^{-11}	2.3889×10^{-11}	—	10^{-7}*
joules (abs) by —→	7.3756×10^{-1}	2.3730×10	3.7251×10^{-7}	2.7778×10^{-7}	9.4799×10^{-4}	2.3889×10^{-4}	10^{7}*	—

Fig. 3M (1) (Cont'd)

Force (mLt^{-2})

Multiply ↘	dynes	poundals	pounds	grams	newtons
	to obtain →				
dynes by →	—	7.2330×10^{-5}	2.2481×10^{-6}	1.0197×10^{-3}	10^{-5}*
poundals by →	1.3826×10^{4}	—	3.1081×10^{-2}	1.4098×10	1.3826×10^{-1}
pounds (force) by →	4.4482×10^{5}	3.2174×10	—	4.5359×10^{2}	4.4482
grams by →	9.8067×10^{2}	7.0932×10^{-2}	2.2046×10^{-3}	—	980.66×10^{-3}
newtons by →	10^{5}*	7.2330	2.2481×10^{-1}	1.0197×10^{2}	—

Mass (m)

Multiply ↘	pounds	grams	kilograms	slugs	grains
	to obtain →				
pounds(avdp) by →	—	4.5359×10^{2}	4.5359×10^{-1}	3.1081×10^{-2}	7000*
grams by →	2.2046×10^{-3}	—	10^{-3}*	6.8521×10^{-5}	1.5432×10
kilograms by →	2.2046	10^{3}*	—	6.8522×10^{-2}	1.5432×10^{4}
slugs by →	32.174	1.4594×10^{4}	1.4594×10	—	2.2522×10^{5}
grains by →	1.4286×10^{-4}	6.4799×10^{-2}	6.4799×10^{-5}	4.4401×10^{-6}	—

Permeability (L^2)

Multiply ↘	darcies	millidarcies	square feet	square centimeters	square meters
	to obtain →				
darcies by →	—	1000*	1.0623×10^{-11}	9.8692×10^{-9}	9.8692×10^{-13}
millidarcies by →	10^{-3}*	—	1.0623×10^{-14}	9.8692×10^{-12}	9.8692×10^{-16}
square feet by →	9.4135×10^{10}	9.4135×10^{13}	—	9.2903×10^{2}	9.2903×10^{-2}
square centimeters by →	1.0133×10^{8}	1.0133×10^{11}	1.0764×10^{-3}	—	10^{-4}*
square meters by →	1.0133×10^{12}	1.0133×10^{15}	1.0764×10	10^{4}*	—
permeability by →	1.1270	1.1270×10^{3}	1.1972×10^{-11}	1.1122×10^{-8}	1.1122×10^{-12}

Power (mL^2t^{-3})

Multiply ↘	foot-pounds per second	horsepower	foot-poundals per second	watts (abs)	ergs per second
	to obtain →				
foot-pounds per second by →	—	1.8182×10^{-3}	3.2174×10	1.3558	1.3558×10^{7}
horsepower by →	550*	—	1.7696×10^{4}	7.4570×10^{2}	7.4570×10^{9}
foot-poundals per second by →	3.1081×10^{-2}	5.6511×10^{-5}	—	4.2140×10^{-2}	4.2140×10^{5}
watts (abs) by →	7.3756×10^{-1}	1.3410×10^{-3}	2.3730×10	—	10^{7}*
ergs per second by →	7.3756×10^{-8}	1.3410×10^{-10}	2.3730×10^{-6}	10^{-7}*	—

Fig. 3M (1) (Cont'd)

Pressure ($mL^{-1}t^{-2}$)

Multiply ↘ / to obtain →	pounds per square inch	atmospheres	inches Hg at 32°F	inches H_2O at 39.2°F	poundals per square foot	dynes per square centimeter	newtons per square meter
pounds per square inch by →	—	6.8046×10^{-2}	2.0360	2.7681×10	4.6331×10^{3}	6.8948×10^{4}	6.8948×10^{3}
atmospheres by →	1.4696×10	—	2.9921×10	4.0679×10^{2}	6.8085×10^{4}	1.0133×10^{6}	1.0132×10^{5}
inches Hg at 32°F by →	4.9115×10^{-1}	3.3421×10^{-2}	—	1.3595×10	2.2755×10^{3}	3.3864×10^{4}	3.3864×10^{3}
inches H_2O at 39.2°F by →	3.6126×10^{-2}	2.4583×10^{-3}	7.3554×10^{-2}	—	1.6737×10^{2}	2.4908×10^{3}	2.4908×10^{2}
poundals per square foot by →	2.1584×10^{-4}	1.4687×10^{-5}	4.3946×10^{-4}	5.9746×10^{-3}	—	1.4882×10	1.4882
dynes per square centimeter by →	1.4504×10^{-5}	9.8692×10^{-7}	2.9530×10^{-5}	4.0147×10^{-4}	6.7197×10^{-2}	—	10^{-1}*
newtons per square meter by →	1.4504×10^{-4}	9.8692×10^{-6}	2.9530×10^{-4}	4.0147×10^{-3}	6.7197×10^{-1}	10*	—

Rate of flow (volumetric) (L^3t^{-1})

Multiply ↘ / to obtain →	cubic centimeters per second	cubic meters per second	cubic feet per second	cubic feet per hour	cubic feet per day	barrels per day	gallons per day
cubic centimeters per second by →	—	10^{-6}*	3.5314×10^{-5}	1.2713×10^{-1}	3.0512	5.4343×10^{-1}	2.2824×10
cubic meters per second by →	10^{6}*	—	3.5314×10	1.2713×10^{5}	3.0512×10^{6}	5.4343×10^{5}	2.2824×10^{7}
cubic feet per second by →	2.8317×10^{4}	2.8317×10^{-2}	—	3600*	86,400*	1.5388×10^{4}	6.4632×10^{5}
cubic feet per hour by →	7.8658	7.8658×10^{-6}	2.7778×10^{-4}	—	2.4000×10*	4.2746	1.7953×10^{2}
cubic feet per day by →	3.2774×10^{-1}	3.2774×10^{-7}	1.1574×10^{-5}	4.1667×10^{-2}	—	1.7801×10^{-1}	7.4805
barrels per day by →	1.8401	1.8401×10^{-6}	6.4984×10^{-5}	2.3394×10^{-1}	5.6146	—	42*
gallons per day by →	4.3813×10^{-2}	4.3813×10^{-8}	1.5472×10^{-6}	5.5700×10^{-3}	1.3368×10^{-1}	2.3810×10^{-2}	—

Specific gravity

Liquids lighter than water:
Specific gravity $\frac{60°F}{60°F} = \frac{140}{°Baume + 130}$
Specific gravity $\frac{60°F}{60°F} = \frac{141.5}{°_{API} + 131.5}$
Specific gravity $\frac{60°F}{60°F} = \frac{400}{°Brix + 400}$
Liquids heavier than water:
Specific gravity $\frac{60°F}{60°F} = \frac{145}{145 - °Baume}$
Specific gravity $\frac{60°F}{60°F} = \frac{°Twaddell}{200} + 1$

Fig. 3M (1) (Cont'd)

Temperature conversion formulas (T)

To convert from ↘ / to →	°Fahrenheit	°Centigrade	°Rankine	°Kelvin
°Fahrenheit	—	$(°F - 32)\left(\frac{1}{1.8}\right)$	°F + 460	$(°F + 460)\left(\frac{1}{1.8}\right)$
°Centigrade	(°C)(1.8) + 32	—	(°C)(1.8) + 492	°C + 273
°Rankine	°R − 460	$(°R - 492)\left(\frac{1}{1.8}\right)$	—	$°R\left(\frac{1}{1.8}\right)$
°Kelvin	(°K)(1.8) − 460	°K − 273	(°K)(1.8)	—

Time (t)—table of day number in year

Day of month	Jan.	Feb.	Mar.	Apr.	May	June	July	Aug.	Sept.	Oct.	Nov.	Dec.
1	1	32	60	91	121	152	182	213	244	274	305	335
2	2	33	61	92	122	153	183	214	245	275	306	336
3	3	34	62	93	123	154	184	215	246	276	307	337
4	4	35	63	94	124	155	185	216	247	277	308	338
5	5	36	64	95	125	156	186	217	248	278	309	339
6	6	37	65	96	126	157	187	218	249	279	310	340
7	7	38	66	97	127	158	188	219	250	280	311	341
8	8	39	67	98	128	159	189	220	251	281	312	342
9	9	40	68	99	129	160	190	221	252	282	313	343
10	10	41	69	100	130	161	191	222	253	283	314	344
11	11	42	70	101	131	162	192	223	254	284	315	345
12	12	43	71	102	132	163	193	224	255	285	316	346
13	13	44	72	103	133	164	194	225	256	286	317	347
14	14	45	73	104	134	165	195	226	257	287	318	348
15	15	46	74	105	135	166	196	227	258	288	319	349
16	16	47	75	106	136	167	197	228	259	289	320	350
17	17	48	76	107	137	168	198	229	260	290	321	351
18	18	49	77	108	138	169	199	230	261	291	322	352
19	19	50	78	109	139	170	200	231	262	292	323	353
20	20	51	79	110	140	171	201	232	263	293	324	354
21	21	52	80	111	141	172	202	233	264	294	325	355
22	22	53	81	112	142	173	203	234	265	295	326	356
23	23	54	82	113	143	174	204	235	266	296	327	357
24	24	55	83	114	144	175	205	236	267	297	328	358
25	25	56	84	115	145	176	206	237	268	298	329	359
26	26	57	85	116	146	177	207	238	269	299	330	360
27	27	58	86	117	147	178	208	239	270	300	331	361
28	28	59	87	118	148	179	209	240	271	301	332	362
29	29	—*	88	119	149	180	210	241	272	302	333	363
30	30	—	89	120	150	181	211	242	273	303	334	364
31	31	—	90	—	151	—	212	243	—	304	—	365

*For leap year, one must be added to each day number after February 29. (Leap years are every fourth year, e.g.: 1920, 1924, 1928, 1932, 1936, 1940, 1944, 1948, 1952, 1956, 1960, 1964, 1968, 1972, 1976, 1980, etc.)

Fig. 3M (1) (Cont'd)

Time (t)

Multiply ↘	seconds	minutes	hours	days	months (mean calendar)	years (common)
	to obtain →					
seconds by ——→	—	1.6667×10^{-2}	2.7778×10^{-4}	1.1574×10^{-5}	3.7313×10^{-7}	3.1710×10^{-8}
minutes by ——→	60*	—	1.6667×10^{-2}	6.9444×10^{-4}	2.2831×10^{-5}	1.9026×10^{-6}
hours by ——→	3600*	60*	—	4.1667×10^{-2}	1.3699×10^{-3}	1.1416×10^{-4}
days by ——→	86,400*	1440*	24*	—	3.2877×10^{-2}	2.7397×10^{-3}
months(mean calendar) by ——→	2.628×10^{6}*	43,800*	730*	30.417	—	8.3333×10^{-2}
years (common) by →	3.1536×10^{7}*	5.2560×10^{5}*	8760*	365*	12*	—

Viscosity (Dynamic) $(mL^{-1}t^{-1})$

Multiply ↘	centipoise	poise [gm/(cm-sec)]	lb/(ft-sec)	kg/(m-sec)	slugs/(ft-sec)
	to obtain →				
centipoise by ——→	—	10^{-2}*	6.7197×10^{-4}	10^{-3}*	2.0885×10^{-5}
poise [gm/(cm-sec)] by ——→	100*	—	6.7197×10^{-2}	0.1*	2.0885×10^{-3}
pounds per foot-second by ——→	1.4882×10^{3}	1.4882×10	—	1.4882	3.1081×10^{-2}
kilograms per meter-second by —→	1000*	10*	6.7197×10^{-1}	—	2.0885×10^{-2}

$$\text{Kinematic viscosity} = \frac{\text{dynamic viscosity}}{\text{density}} = (L^2t^{-1})$$

$$= \frac{\text{centipoise}}{\text{gm/cm}^3} = \text{centistokes}$$

$$\text{Saybolt Universal Viscosity (100°F)} = (\text{kinematic viscosity at 100°F})(4.620)$$
$$= (\text{centistokes})(4.620)$$

$$\text{Saybolt Universal Viscosity (210°F)} = (\text{kinematic viscosity at 210°F})(4.6524)$$
$$= (\text{centistokes})(4.6524)$$

Velocity (Lt^{-1})

Multiply ↘	centimeters per second	feet per second	feet per minute	feet per hour	inches per second	miles per hour	meters per second
	to obtain →						
centimeters per second by →	—	3.2808×10^{-2}	1.9685	1.1811×10^{2}	3.937×10^{-1}*	2.2369×10^{-2}	10^{-2}*
feet per second by ——→	3.0480×10	—	60*	3600*	12*	6.8182×10^{-1}	3.0480×10^{-1}
feet per minute by ——→	5.0800×10^{-1}	1.6667×10^{-2}	—	60*	0.2*	1.1364×10^{-2}	5.0800×10^{-3}
feet per hour by ——→	8.4667×10^{-3}	2.7778×10^{-4}	1.6667×10^{-2}	—	3.3333×10^{-3}	1.8939×10^{-4}	8.4667×10^{-5}
inches per second by ——→	2.5400	8.3333×10^{-2}	5*	300*	—	5.6818×10^{-2}	2.5400×10^{-2}
miles per hour by ——→	4.4704×10	1.4667	88*	5280*	17.6*	—	4.4704×10^{-1}
meters per second by ——→	100*	3.2808	1.9685×10^{2}	1.1811×10^{4}	3.937×10^{1}*	2.2369	—

Fig. 3M (1) (Cont'd)

Volume (L^3)

Multiply ↘ to obtain →	acre-feet	barrels oil (U.S.)	cubic feet	gallons liquid (U.S.)	cubic centimeters	liters	cubic meters
acre-feet by	—	7.7584×10^{3}	43,560*	3.2585×10^{5}	1.2335×10^{9}	1.2334×10^{6}	1.2335×10^{3}
barrels oil (U.S.) by	1.2889×10^{-4}	—	5.6146	42*	1.5899×10^{5}	1.5894×10^{2}	1.5899×10^{-1}
cubic feet by	2.2957×10^{-5}	1.7811×10^{-1}	—	7.4805	2.8317×10^{4}	2.8316×10	2.8317×10^{-2}
gallons liquid (U.S.) by	3.0689×10^{-6}	2.3810×10^{-2}	1.3368×10^{-1}	—	3.7854×10^{3}	3.7853	3.7854×10^{-3}
cubic centimeters by	8.1071×10^{-10}	6.2898×10^{-6}	3.5314×10^{-5}	2.6417×10^{-4}	—	9.9997×10^{-4}	10^{-6}*
liters by	8.1073×10^{-7}	6.2899×10^{-3}	3.5315×10^{-2}	2.6418×10^{-1}	1.0000×10^{3}	—	1.0000×10^{-3}
cubic meters by	8.1071×10^{-4}	6.2898	3.5314×10	2.6417×10^{2}	10^{6}*	9.9997×10^{2}	—

Fig. 3M(1) Con't.

Conversion factors (applying to oil country calculations)

Acre	=	43,560.	square feet	Foot-pound	=	0.001286	British Thermal Unit	Mile	=	1,900.8	varas (Texas)
Acre	=	4,047.	square meters	Foot-pound per second	=	0.001818	horsepower	Mile per hour	=	1.4667	feet per second
Acre	=	160	square rods	Gallon (U.S.)	=	0.02381	barrel	Ounce (Avoirdupois)	=	437.5	grains
Acre	=	5,645.4	square varas (Texas)	Gallon (U.S.)	=	0.1337	cubic feet	Ounce (Avoirdupois)	=	28.3495	grams
Acre	=	0.4047	hectares	Gallon (U.S.)	=	231.000	cubic inches	Part per million	=	0.05835	grain per gallon
Acre foot	=	7,758.	barrels	Gallon (U.S.)	=	3.785	liters	Part per million	=	8.345	pounds per million gals
Atmosphere	=	33.94	feet of water	Gallon (U.S.)	=	0.8327	gallon (Imperial)	Pood (Russian)	=	36.112	pounds
Atmosphere	=	29.92	inches of mercury	Gallon (Imperial)	=	1.2009	gallons (U.S.)	Pound	=	7,000.	grains
Atmosphere	=	760.0	millimeters of mercury	Gallon (Imperial)	=	277.274	cubic inches	Pound	=	0.4536	kilogram
Atmosphere	=	14.70	pounds per square inch	Gallon per minute	=	1.429	barrels per hour	Pound per square inch	=	2.309	feet of water @ 60°F
Barrel	=	5.6146	cubic feet	Gallon per minute	=	0.1337	cubic feet per minute	Pound per square inch	=	2.0353	inches of mercury
Barrel	=	42.0	gallons	Gallon per minute	=	34.286	barrels per day	Pound per square inch	=	51.697	millimeters of mercury
Barrel of water @ 60°F	=	0.1588	metric ton	Grain (Avoirdupois)	=	0.06430	gram	Pound per square inch	=	0.0703	kilograms per sq cm
Barrel (36° API)	=	0.1342	metric ton	Grain per gallon	=	17.118	parts per million	Pound per million gals	=	0.00700	grain per gallon
Barrel per hour	=	0.0936	cubic feet per minute	Grain per gallon	=	142.86	pounds per million gals	Pound per million gals	=	0.11982	parts per million
Barrel per hour	=	0.700	gallon per minute	Grain per gallon	=	0.01714	gram per liter	Quart (Liquid)	=	0.946	liter
Barrel per hour		2.695	cubic inches per second	Gram	=	15.432	grains	Quintal (Mexican)	=	101.467	pounds
Barrel per day	=	0.02917	gallon per minute	Gram	=	0.03527	ounce	Rod	=	16.5	feet
British Thermal Unit	=	0.2520	kilogram calorie	Gram per liter	=	58.418	grains per gallon	Rod	=	25.0	links
British Thermal Unit	=	0.2928	watt hour	Hectare	=	2.471	acres	Sack cement (set)	=	1.1	cubic feet
B.T.U. per minute	=	0.02356	horsepower	Hectare	=	0.010	square kilometer	Square centimeter	=	0.1550	square inch
Centimeter	=	0.3937	inch	Horsepower	=	33,000.	foot-pounds per minute	Square foot	=	0.0929	square meter
Centimeter of mercury	=	0.1934	pound per square inch	Horsepower	=	550.	foot-pounds per second	Square foot	=	0.1296	square vara (Texas)
Chain	=	66.00	feet	Horsepower	=	1.014	horsepower (metric)	Square inch	=	6.452	square centimeters
Chain	=	4.00	rods	Horsepower	=	0.7457	kilowatt	Square kilometer	=	0.3861	square mile
Cubic centimeter	=	0.06102	cubic inch	Horsepower hour	=	2,547.	British Thermal Units	Square meter	=	10.76	square feet
Cubic foot	=	0.1781	barrel	Inch	=	2.540	centimeters	Square mile	=	2.590	square kilometers
Cubic foot	=	7.4805	gallons (U.S.)	Inch of mercury	=	1.134	feet of water	Square vara (Texas)	=	7.716	square feet
Cubic foot	=	0.02832	cubic meter	Inch of mercury	=	0.4912	pound per square inch	Temp Centigrade	=	5/9 (Temp Fahr.—32)	
Cubic foot	=	0.9091	sacks cement (set)	Inch of water @ 60°F	=	0.0361	pound per square inch	Temp Fahrenheit	=	9/5 Temp Cent. + 32	
Cubic foot per minute	=	10.686	barrels per hour	Kilogram	=	2.2046	pounds	Temp Absolute °C	=	Temp °C + 273	
Cubic foot per minute	=	28.800	cubic inches per second	Kilogram Calorie	=	3.968	British Thermal Units	Temp Absolute °F	=	Temp °F + 460	
Cubic foot per minute		7.481	gallons per minute	Kilogram per square cm	=	14.223	pounds per square inch	Ton (Long)	=	2,240.	pounds
Cubic inch	=	16.387	cubic centimeters	Kilometer	=	3,281.	feet	Ton (Metric)	=	2,205.	pounds
Cubic meter	=	6.2897	barrels	Kilometer	=	0.6214	mile	Ton (Short or Net)	=	2,000.	pounds
Cubic meter	=	35.314	cubic feet	Kilowatt	=	1.341	horsepower	Ton (Metric)	=	1.102	tons (short or net)
Cubic meter	=	1.308	cubic yards	Link (Surveyor's)	=	7.92	inches	Ton (Metric)	=	1,000.	kilograms
Cubic yard	=	4.8089	barrels	Liter	=	0.2642	gallon	Ton (Metric)	=	6.297	barrels of water @ 60°F
Cubic yard	=	46,656.	cubic inches	Liter	=	1.0567	quarts	Ton (Metric)	=	7.454	barrels (36° API)
Cubic yard	=	0.7646	cubic meter	Meter	=	3.281	feet	Ton (Short or Net)	=	0.907	ton (metric)
Foot	=	30.48	centimeters	Meter	=	39.37	inches	Vara (Texas)	=	2.7778	feet
Foot	=	0.3048	meter	Mile	=	5,280.	feet	Vara (Texas)	=	33.3333	inches
Foot	=	0.3600	vara (Texas)	Mile	=	1.609	kilometers	Watt-hour	=	3.415	British Thermal Units
Foot of water @ 60°F	=	0.4331	pound per square inch					Yard	=	0.9144	meter
Foot per second	=	0.68182	mile per hour								

Fig. M-2 Various Conversions

API Plain End Line Pipe

Dimensions, Weights and Test Pressures

Size: Outside Diameter, In.	Weight, lb per ft Calculated Plain Ends	Wall Thickness, In.	Test Pressures, psi					Tons per Mile	Feet per Ton
			Grade A	Grade B	Grade X42	Grade X46	Grade X52		
3½	4.51	.125	1300	1500	—	—	—	11.9	444
	5.58	.156	1600	1900	—	—	—	14.7	359
	6.63	.188	1900	2200	—	—	—	17.5	302
	7.58	.216	2200	2500	—	—	—	20.0	264
	8.68	.250	2500	2500	—	—	—	22.9	231
	9.67	.281	2500	2500	—	—	—	25.5	207
4	5.17	.125	1100	1300	—	—	—	13.7	387
	6.41	.156	1400	1600	—	—	—	16.9	313
	7.63	.188	1700	2000	—	—	—	20.1	263
	9.11	.226	2000	2400	—	—	—	24.1	220
	10.01	.250	2200	2500	—	—	—	26.4	200
	11.17	.281	2500	2500	—	—	—	29.5	180
4½	5.84	.125	1000	1200	—	—	—	15.4	344
	7.25	.156	1200	1500	—	—	—	19.1	276
	8.64	.188	1500	1800	—	—	—	22.8	232
	10.00	.219	1700	2000	—	—	—	26.4	200
	10.79	.237	1900	2200	—	—	—	28.5	186
	11.35	.250	2000	2300	—	—	—	30.0	177
	12.67	.281	2200	2500	—	—	—	33.5	158
	13.98	.312	2500	2500	—	—	—	36.9	144
5¾	9.02	.156	1000	1200	—	—	—	23.8	222
	10.76	.188	1200	1400	—	—	—	28.4	186
	12.49	.219	1400	1700	—	—	—	33.0	161
	14.62	.258	1700	1900	—	—	—	38.6	137
	15.87	.281	1800	2100	—	—	—	41.9	127
	17.52	.312	2000	2400	—	—	—	46.3	115
	19.16	.344	2200	2500	—	—	—	50.6	105
	20.78	.375	2300	2500	—	—	—	54.0	96
	27.04	.500	2500	2500	—	—	—	71.0	73
6⅝	12.89	.188	1000	1200	1790	1960	2220	34.0	156
	13.92	.203	1100	1300	1940	2120	2390	37.0	144
	14.97	.219	1200	1400	2090	2290	2580	39.5	134
	17.02	.250	1400	1600	2380	2610	2950	44.9	118
	18.97	.280	1500	1800	2670	2920	3000	50.1	106
	21.07	.312	1700	2000	2970	3000	3000	55.6	95
	23.06	.344	1900	2200	3000	3000	3000	60.9	87
	25.03	.375	2000	2400	3000	3000	3000	66.1	80
	28.57	.432	2300	2500	3000	3000	3000	75.4	70
	32.71	.500	2500	2500	3000	3000	3000	86.0	60
8⅝	14.11	.156	650	750	1140	1250	1420	37.0	175
	16.90	.188	800	900	1380	1510	1710	44.6	118
	18.27	.203	—	—	1490	1630	1840	48.2	109
	19.64	.219	900	1100	1600	1760	1990	51.8	102
	22.36	.250	1000	1200	1830	2010	2270	59.0	89
	24.70	.277	1200	1300	2030	2220	2510	65.2	81

Fig. 3N-1 (Courtesy Bethlehem Steel)

Extra Strong Plain End Line Pipe[1]

Dimensions, Weights and Test Pressures

Nominal Size, in.	Weight, lb per ft Calculated Plain Ends	Wall Thickness, in.	Diameters, in.		Test Pressures, psi		
			O D	I D	Butt-welded	ERW and/or Seamless	
						Grade A	Grade B
⅛	.31	.095	.405	.215	850	850	850
¼	.54	.119	.540	.302	850	850	850
⅜	.74	.126	.675	.423	850	850	850
½	1.09	.147	.840	.546	850	850	850
¾	1.47	.154	1.050	.742	850	850	850
1	2.17	.179	1.315	.957	850	850	850
1¼	3.00	.191	1.660	1.278	1300	1500	1600
1½	3.63	.200	1.900	1.500	1300	1500	1600
2	5.02	.218	2.375	1.939	1300	1500	1600
2½	7.66	.276	2.875	2.323	1300	1500	1600
3	10.25	.300	3.500	2.900	1300	1500	1600
3½	12.51	.318	4.000	3.364	1700	1700	1800
4	14.98	.337	4.500	3.826	1700	1700	1800
5	20.78	.375	5.563	4.813	—	1700	1800
6	28.57	.432	6.625	5.761	—	1700	1800
8	43.39	.500	8.625	7.625	—	1700	2400
10	54.74	.500	10.750	9.750	—	1600	1900
12	65.42	.500	12.750	11.750	—	1600	1900

[1] Extra strong line pipe can be specified threaded and coupled.

(a) The customary variation in weight for any length of pipe is 10 pct above and 3½ pct below; the carload weight is customarily not more than 1¾ pct under the calculated weight.

(b) API STD 5L covers these data in detail.

Length Range—Buttweld S/R/L, 12/22 ft.
ERW S/R/L, 12/25 ft.
ERW D/R/L, 32/50 ft.

End Finish—Beveled in accordance with standard practice.

Fig. 3N-1 (Cont'd) (Courtesy Bethlehem Steel)

Extra Strong Plain End Line Pipe[1]

Dimensions, Weights and Test Pressures

Nominal Size, In.	Weight, lb per ft Calculated Plain Ends	Wall Thickness, In.	Diameters, In.		Test Pressures, psi		
			O D	I D	Butt-welded	ERW and/or Seamless	
						Grade A	Grade B
⅛	.31	.095	.405	.215	850	850	850
¼	.54	.119	.540	.302	850	850	850
⅜	.74	.126	.675	.423	850	850	850
½	1.09	.147	.840	.546	850	850	850
¾	1.47	.154	1.050	.742	850	850	850
1	2.17	.179	1.315	.957	850	850	850
1¼	3.00	.191	1.660	1.278	1300	1500	1600
1½	3.63	.200	1.900	1.500	1300	1500	1600
2	5.02	.218	2.375	1.939	1300	1500	1600
2½	7.66	.276	2.875	2.323	1300	1500	1600
3	10.25	.300	3.500	2.900	1300	1500	1600
3½	12.51	.318	4.000	3.364	1700	1700	1800
4	14.98	.337	4.500	3.826	1700	1700	1800
5	20.78	.375	5.563	4.813	—	1700	1800
6	28.57	.432	6.625	5.761	—	1700	1800
8	43.39	.500	8.625	7.625	—	1700	2400
10	54.74	.500	10.750	9.750	—	1600	1900
12	65.42	.500	12.750	11.750	—	1600	1900

[1] Extra strong line pipe can be specified threaded and coupled.

(a) The customary variation in weight for any length of pipe is 10 pct above and 3½ pct below; the carload weight is customarily not more than 1¾ pct under the calculated weight.

(b) API STD 5L covers these data in detail.

Length Range—Buttweld S/R/L, 12/22 ft.
ERW S/R/L, 12/25 ft.
ERW D/R/L, 32/50 ft.

End Finish—Beveled in accordance with standard practice.

Fig. 3N-1 (Cont'd) (Courtesy Bethlehem Steel)

API Plain End Line Pipe (Continued)

Dimensions, Weights and Test Pressures

Size: Outside Diameter, In.	Weight, lb per ft Calculated Plain Ends	Wall Thickness, in.	Test Pressures, psi					Tons per Mile	Feet per Ton
			Grade A	Grade B	Grade X42	Grade X46	Grade X52		
8⅝	27.74	.312	1300	1500	2280	2500	2830	73.2	72
	28.55	.322	1300	1600	2360	2580	2920	75.4	70
	30.40	.344	1400	1700	2520	2760	3000	80.3	66
	33.04	.375	1600	1800	2740	3000	3000	87.2	61
	38.26	.438	1800	2100	3000	3000	3000	101.0	52
	43.39	.500	2100	2400	3000	3000	3000	114.5	46
10¾	21.15	.188	650	750	1250	1370	1550	55.8	95
	22.88	.203	—	—	1350	1480	1670	60.4	87
	24.60	.219	750	850	1460	1600	1810	64.9	81
	28.04	.250	850	1000	1670	1820	2060	74.0	71
	31.20	.279	1000	1200	1860	2030	2300	82.4	64
	34.24	.307	1000	1200	2040	2240	2530	90.4	58
	38.20	.344	1100	1300	2290	2510	2830	100.8	52
	40.48	.365	1200	1400	2430	2660	3000	106.9	49
	48.19	.438	1500	1700	2910	3000	3000	127.2	42
	54.74	.500	1700	2000	3000	3000	3000	144.5	37
12¾	25.16	.188	550	600	1060	1160	1310	66.0	80
	27.22	.203	—	—	1140	1250	1410	71.9	73
	29.28	.219	600	700	1230	1350	1520	77.3	68
	33.38	.250	700	800	1410	1540	1740	88.1	60
	37.45	.281	800	950	1580	1730	1950	98.9	53
	41.51	.312	900	1000	1750	1920	2170	109.6	48
	43.77	.330	1000	1200	1850	2030	2290	115.6	46
	45.55	.344	1000	1200	1930	2110	2390	120.3	44
	49.56	.375	1100	1200	2100	2300	2600	130.8	40
	53.56	.406	—	—	2280	2490	2820	141.4	38
	57.53	.438	1200	1400	2460	2690	3000	151.9	35
	65.42	.500	1400	1600	2810	3000	3000	172.7	31
14	30.93	.210	—	—	1080	1180	1330	81.7	65
	32.20	.219	—	—	1120	1230	1390	85.0	63
	36.71	.250	650	750	1280	1400	1580	96.9	55
	41.21	.281	700	850	1440	1570	1780	108.8	49
	45.68	.312	800	950	1600	1750	1980	120.6	44
	50.14	.344	900	1000	1760	1930	2180	132.4	40
	54.57	.375	950	1100	1920	2100	2370	144.1	37
	58.98	.406	—	—	2080	2270	2570	156.0	34
	63.37	.438	1100	1300	2240	2450	2770	167.3	32
	67.74	.469	—	—	2400	2620	2970	178.8	30
	72.09	.500	1300	1500	2550	2800	3000	190.3	28
16	36.87	.219	—	—	980	1080	1220	97.3	54
	42.05	.250	550	650	1120	1230	1390	111.0	48
	47.22	.281	650	750	1260	1380	1560	124.7	42
	52.36	.312	700	800	1400	1530	1730	138.2	38
	57.48	.344	750	900	1540	1690	1910	151.7	35

Fig. 3N-1 (Cont'd) (Courtesy Bethlehem Steel)

API Plain End Line Pipe (Continued)

Dimensions, Weights and Test Pressures

Size: Outside Diameter, In.	Weight, lb per ft Calculated Plain Ends	Wall Thickness, In.	Test Pressures, psi					Tons per Mile	Feet per Ton
			Grade A	Grade B	Grade X42	Grade X46	Grade X52		
16	62.58	.375	850	1000	1680	1840	2080	165.2	32
	67.66	.406	—	—	1820	1990	2250	179.0	30
	72.72	.438	1000	1100	1960	2150	2430	192.0	28
	77.75	.469	—	—	2100	2300	2600	205.3	26
	82.77	.500	1100	1300	2240	2450	2770	218.5	24
18	47.39	.250	500	600	1000	1090	1230	125.1	42
	53.22	.281	550	650	1120	1230	1380	140.5	38
	59.03	.312	600	750	1240	1360	1540	155.8	34
	64.82	.344	700	800	1370	1500	1690	171.1	31
	70.59	.375	750	900	1490	1630	1850	186.4	28
	76.34	.406	—	—	1620	1770	2000	201.5	26
	82.06	.438	900	1000	1740	1910	2160	216.6	24
	87.77	.469	—	—	1870	2040	2310	231.7	23
	93.45	.500	1000	1200	1990	2180	2460	246.7	21
20	52.73	.250	450	500	950	1040	1170	139.2	38
	59.23	.281	500	600	1070	1170	1320	156.4	34
	65.71	.312	550	650	1180	1300	1470	173.5	30
	72.16	.344	600	700	1310	1430	1610	190.5	28
	78.60	.375	700	800	1420	1560	1760	207.5	25
	85.01	.406	—	—	1540	1690	1910	224.0	24
	91.41	.438	800	900	1660	1820	2050	241.3	22
	97.78	.469	—	—	1780	1950	2200	258.1	21
	104.13	.500	900	1000	1890	2070	2340	274.9	19
22	58.07	.250	—	—	860	950	1070	153.3	35
	65.24	.281	—	—	970	1060	1200	172.2	31
	72.38	.312	500	600	1080	1180	1330	191.1	28
	79.51	.344	550	650	1190	1300	1470	209.9	26
	86.61	.375	600	700	1290	1420	1600	228.7	24
	93.69	.406	—	—	1400	1530	1730	247.3	22
	100.75	.438	700	850	1510	1650	1870	265.9	20
	107.79	.469	—	—	1620	1770	2000	284.6	19
	114.81	.500	800	950	1720	1890	2130	303.1	18
24	63.41	.250	—	—	790	870	980	167.4	32
	71.25	.281	—	—	890	970	1100	188.1	29
	79.06	.312	450	550	990	1080	1220	208.7	26
	86.85	.344	500	600	1090	1190	1350	229.3	24
	94.62	.375	550	650	1190	1300	1470	249.8	22
	102.37	.406	—	—	1280	1410	1590	270.3	20
	110.10	.438	650	750	1380	1520	1710	290.7	19
	117.81	.469	—	—	1480	1620	1830	311.1	17
	125.49	.500	750	850	1580	1730	1950	331.3	16

(a) API STD 5L and API STD 5LX cover these data in detail.

(b) Test pressures shown for all grades for information purposes only. For grades available contact nearest Bethlehem district office.

Fig. 3N-1 (Cont'd)

DIMENSIONS, WEIGHTS &
MILL TEST PRESSURES

THREADED & COUPLED API LINE PIPE (3)

Standard Weight

NOMINAL SIZE	WALL THICKNESS	WEIGHTS PER FOOT			DIAMETERS		MILL TEST PRESSURES			COUPLINGS		
		NOMINAL	CALCULATED									
		T & C (1)	PLAIN END	WEIGHT GAIN DUE TO END FINISHING*	OUTSIDE (D_4)	INSIDE (d)	*Starweld* GRADE A-25 (2)	GRADE A (2)	GRADE B (2)	OUTSIDE DIAMETER (W)	LENGTH (N_L)	CAL. WEIGHT
INCHES	INCHES	POUNDS PER FT		POUNDS	INCHES		POUNDS PER SQUARE INCH			INCHES	INCHES	POUNDS
1/8	.068	.25	.24	.20	.405	.269	700	2500	2500	.563	1-1/16	.04
1/4	.088	.43	.42	.20	.540	.364	700	2500	2500	.719	1-5/8	.09
3/8	.091	.57	.57	.20	.675	.493	700	2500	2500	.875	1-5/8	.13
1/2	.109	.86	.85	.20	.840	.622	700	2500	2500	1.063	2-1/8	.24
3/4	.113	1.14	1.13	.20	1.050	.824	700	2500	2500	1.313	2-1/8	.34
1	.133	1.70	1.68	.20	1.315	1.049	700	2500	2500	1.576	2-5/8	.54
1-1/4	.140	2.30	2.27	.60	1.660	1.380	1000	2500	2500	2.054	2-3/4	1.03
1-1/2	.145	2.75	2.72	.40	1.900	1.610	1000	2500	2500	2.200	2-3/4	.90
2	.154	3.75	3.65	1.20	2.375	2.067	1000	2500	2500	2.875	2-7/8	1.86
2-1/2	.203	5.90	5.79	1.80	2.875	2.469	1000	2500	2500	3.375	4-1/8	3.27
3	.216	7.70	7.58	1.80	3.500	3.068	1000	2500	2500	4.000	4-1/4	4.09
3-1/2	.226	9.25	9.11	3.20	4.000	3.548	1200	2700	2800	4.625	4-3/8	5.92
4	.237	11.00	10.79	4.40	4.500	4.026	1200	2500	2800	5.200	4-1/2	7.59
5	.258	15.00	14.62	5.60	5.563	5.047	---	2200	2600	6.296	4-5/8	9.98
6	.280	19.45	18.97	7.20	6.625	6.065	---	1900	2200	7.390	4-7/8	12.92
8	.277	25.55	24.70	14.80	8.625	8.071	---	1400	1700	9.625	5-1/4	23.18
8	.322	29.35	28.55	14.00	8.625	7.981	---	1700	2000	9.625	5-1/4	23.18
10	.279	32.75	31.20	20.00	10.750	10.192	---	1200	1400	11.750	5-3/4	31.55
10	.307	35.75	34.24	19.20	10.750	10.136	---	1300	1500	11.750	5-3/4	31.55
10	.365	41.85	40.48	17.40	10.750	10.020	---	1500	1800	11.750	5-3/4	31.55
12	.330	45.45	43.77	32.60	12.750	12.090	---	1200	1400	14.000	6-1/8	49.27
12	.375	51.15	49.56	30.80	12.750	12.000	---	1300	1500	14.000	6-1/8	49.27
14 OD	.375	57.00	54.57	24.60	14.000	13.250	---	1200	1400	15.000	6-3/8	45.83
16 OD	.375	65.30	62.58	30.00	16.000	15.250	---	1100	1200	17.000	6-3/4	55.83

*Calculated weights shall be determined in accordance with the following formula:

$$W_L = w_{pe} \times L + e_w$$

where

W_L = *calculated weight of a piece of pipe of length L, lb*

w_{pe} = *plain-end weight, lb/ft*

L = *length of pipe, including end finish*

e_w = *weight gain or loss due to end finishing*

Nominal weights T&C are shown for purpose of identification in ordering.

Fig. 3N-2 (Courtesy Bethlehem Steel)

Line Pipe Data
Seamless, Electric Resistance Weld

Actual OD Inches	Wall Thickness Inches	Weight Lb/Ft Plain End	Weight Tons/Mile	Class	Mill Test Pressure*—psi						
					Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60
2.375	0.063	1.56	4.11		950	1110	1670	—	—	—	—
	0.078	1.91	5.05		1180	1380	2070	—	—	—	—
	0.083	2.03	5.36		1260	1470	2200	—	—	—	—
	0.094	2.29	6.05		1420	1660	2490	2730	3000	—	—
	0.109	2.64	6.96		1650	1930	2890	3000	3000	—	—
	0.125	3.00	7.93		1890	2210	3000	3000	3000	—	—
	0.154	3.65	9.64	Std.	2330	2500	3000	3000	3000	3000	3000
	0.218	5.02	13.26	XS	2500	2500	3000	3000	3000	3000	3000
	0.344	7.46	19.70		2500	2500	3000	3000	3000	3000	3000
	0.436	9.03	23.84	XXS	2500	2500	3000	3000	3000	—	—
2.875	0.078	2.33	6.15		980	1140	1710	—	—	—	—
	0.083	2.47	6.53		1040	1210	1820	—	—	—	—
	0.094	2.79	7.37		1180	1370	2060	2260	2550	—	—
	0.109	3.22	8.50		1360	1590	2390	2620	2960	—	—
	0.125	3.67	9.69		1570	1830	2740	3000	3000	—	—
	0.141	4.12	10.87		1770	2060	3000	3000	3000	—	—
	0.156	4.53	11.96		1950	2280	3000	3000	3000	—	—
	0.188	5.40	14.24		2350	2500	3000	3000	3000	—	—
	0.203	5.79	15.29	Std.	2500	2500	3000	3000	3000	3000	3000
	0.276	7.66	20.23	XS	2500	2500	3000	3000	3000	3000	3000
	0.375	10.01	26.43		2500	2500	3000	3000	3000	3000	3000
	0.552	13.69	36.15	XXS	2500	2500	3000	3000	3000	—	—
3.500	0.078	2.85	7.53		800	940	1400	—	—	—	—
	0.083	3.03	8.00		850	1000	1490	—	—	—	—
	0.094	3.42	9.03		970	1130	1690	1850	2090	—	—
	0.109	3.95	10.42		1120	1310	1960	2150	2430	—	—
	0.125	4.51	11.89		1290	1500	2250	2460	2790	—	—
	0.141	5.06	13.35		1450	1690	2540	2780	3000	—	—
	0.156	5.57	14.71		1600	1870	2810	3000	3000	—	—
	0.188	6.65	17.56		1930	2260	3000	3000	3000	—	—
	0.216	7.58	20.00	Std.	2220	2500	3000	3000	3000	3000	3300
	0.250	8.68	22.91		2500	2500	3000	3000	3000	3000	3000
	0.281	9.66	25.50		2500	2500	3000	3000	3000	3000	3000
	0.300	10.25	27.07	XS	2500	2500	3000	3000	3000	3300	3000
	0.438	14.32	37.81		2500	2500	3000	3000	3000	3000	3000
	0.600	18.58	49.06	XXS	2500	2500	3000	3000	3000	—	—
4.000	0.094	3.92	10.35		850	990	1480	1620	1830	—	—
	0.109	4.53	11.96		980	1140	1720	1880	2130	—	—
	0.125	5.17	13.66		1120	1310	1970	2160	2440	—	—
	0.156	6.40	16.91		1400	1640	2460	2690	3000	—	—
	0.188	7.65	20.21		1690	1970	2960	3000	3000	—	—
	0.226	9.11	24.05	Std.	2030	2370	3000	3000	3000	3000	3000
	0.250	10.01	26.43		2250	2620	3000	3000	3000	3000	3000
	0.281	11.16	29.47		2530	2800	3000	3000	3000	3000	3000
	0.318	12.50	33.01	XS	2800	2800	3000	3000	3000	3000	3000
	0.636	22.85	60.32		2800	2800	3000	3000	3000	—	—
4.500	0.094	4.42	11.68		750	880	1320	1440	1630	—	—
	0.109	5.11	13.49		870	1020	1530	1670	1890	—	—
	0.125	5.84	15.42		1000	1170	1750	1920	2170	—	—
	0.141	6.56	17.33		1130	1320	1970	2160	2440	2630	2820

Note: Some outside diameters, walls and grades are listed for information only and are not necessarily regular production items. Non-production sizes may be negotiated upon inquiry.

Some outside diameters, walls and grades are non-API

*See notes at end of table on page 33.

Fig. 3N-3 (Courtesy U.S. Steel)

Line Pipe Data
Seamless, Electric Resistance Weld

Internal Pressure at Minimum Yield Strength*—psi							Ultimate Bursting Pressure*—psi							Actual OD Inches
Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60	Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60	
1590	1860	2230	—	—	—	—	2550	3180	3180	—	—	—	—	
1970	2300	2760	—	—	—	—	3150	3940	3940	—	—	—	—	
2100	2450	2940	—	—	—	—	3350	4190	4190	—	—	—	—	
2370	2770	3320	3640	4120	—	—	3800	4750	4750	4990	5220	—	—	
2750	3210	3860	4220	4770	—	—	4410	5510	5510	5780	6060	—	—	
3160	3680	4420	4840	5470	—	—	5050	6320	6320	6630	6950	—	—	2⅜
3890	4540	5450	5970	6740	7260	7780	6220	7780	7780	8170	8560	9210	9730	
5510	6430	7710	8440	9550	10280	11010	8810	11010	11010	11570	12120	13030	13770	
8690	10140	12170	13330	15060	16220	17380	13900	17380	17380	18250	19120	20570	21730	
11010	12850	15420	16890	19090	—	—	17620	22030	22030	23130	24230	—	—	
1630	1900	2280	—	—	—	—	2600	3260	3260	—	—	—	—	
1730	2020	2430	—	—	—	—	2770	3460	3460	—	—	—	—	
1960	2290	2750	3010	3400	—	—	3140	3920	3920	4120	4320	—	—	
2270	2650	3180	3490	3940	—	—	3640	4550	4550	4780	5000	—	—	
2610	3040	3650	4000	4520	—	—	4170	5220	5220	5480	5740	—	—	
2940	3430	4120	4510	5100	—	—	4710	5890	5890	6180	6470	—	—	
3260	3800	4560	4990	5640	—	—	5210	6510	6510	6840	7160	—	—	2⅞
3920	4580	5490	6020	6800	—	—	6280	7850	7850	8240	8630	—	—	
4240	4940	5930	6500	7340	7910	8470	6780	8470	8470	8900	9320	10030	10590	
5760	6720	8060	8830	9980	10750	11520	9220	11520	11520	12100	12670	13630	14400	
7830	9130	10960	12000	13570	14610	15650	12520	15650	15650	16430	17220	18520	19570	
11520	13440	16130	17660	19970	—	—	18430	23040	23040	24190	25340	—	—	
1340	1560	1870	—	—	—	—	2140	2670	2670	—	—	—	—	
1420	1660	1990	—	—	—	—	2280	2850	2850	—	—	—	—	
1610	1880	2260	2470	2790	—	—	2580	3220	3220	3380	3550	—	—	
1870	2180	2620	2870	3240	—	—	2990	3740	3740	3920	4110	—	—	
2140	2500	3000	3290	3710	—	—	3430	4290	4290	4500	4710	—	—	
2420	2820	3380	3710	4190	—	—	3870	4830	4830	5080	5320	—	—	
2670	3120	3740	4100	4640	—	—	4280	5350	5350	5620	5880	—	—	
3220	3760	4510	4940	5590	—	—	5160	6450	6450	6770	7090	—	—	3½
3700	4320	5180	5680	6420	6910	7410	5920	7410	7410	7780	8150	8760	9260	
4290	5000	6000	6570	7430	8000	8570	6860	8570	8570	9000	9430	10140	10710	
4820	5620	6740	7390	8350	8990	9630	7710	9630	9630	10120	10600	11400	12040	
5140	6000	7200	7890	8910	9600	10290	8230	10290	10290	10800	11310	12170	12860	
7510	8760	10510	11510	13010	14020	15020	12010	15020	15020	15770	16520	17770	18770	
10290	12000	14400	15770	17830	—	—	16460	20570	20570	21600	22630	—	—	
1410	1640	1970	2160	2440	—	—	2260	2820	2820	2960	3100	—	—	
1630	1910	2290	2510	2830	—	—	2620	3270	3270	3430	3600	—	—	
1880	2190	2630	2880	3250	—	—	3000	3750	3750	3940	4130	—	—	
2340	2730	3280	3590	4060	—	—	3740	4680	4680	4910	5150	—	—	
2820	3290	3950	4320	4890	—	—	4510	5640	5640	5920	6200	—	—	
3390	3950	4750	5200	5880	6330	6780	5420	6780	6780	7120	7460	8020	8480	4
3750	4380	5250	5750	6500	7000	7500	6000	7500	7500	7880	8250	8880	9380	
4210	4920	5900	6460	7310	7870	8430	6740	8430	8430	8850	9270	9980	10540	
4770	5560	6680	7310	8270	8900	9540	7630	9540	9540	10020	10490	11290	11930	
9540	11130	13360	14630	16540	—	—	15260	19080	19080	20030	20990	—	—	
1250	1460	1750	1920	2170	—	—	2010	2510	2510	2630	2760	—	—	
1450	1700	2030	2230	2520	—	—	2330	2910	2910	3050	3200	—	—	
1670	1940	2330	2560	2890	—	—	2670	3330	3330	3500	3670	—	—	4½
1880	2190	2630	2880	3260	3510	3760	3010	3760	3760	3950	4140	4450	4700	

Fig. 3N-3 (Cont'd) (Courtesy U.S. Steel)

Line Pipe Data

Seamless, Electric Resistance Weld

Actual OD Inches	Wall Thickness Inches	Weight Lb/Ft Plain End	Weight Tons/Mile	Class	Mill Test Pressure*—psi						
					Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60
4.500	0.156	7.24	19.11		1250	1460	2180	2390	2700	2910	3000
	0.172	7.95	20.99		1380	1610	2410	2640	2980	3000	3000
	0.188	8.66	22.86		1500	1750	2630	2880	3000	3000	3000
	0.203	9.32	24.59		1620	1890	2840	3000	3000	3000	3000
	0.219	10.01	26.43		1750	2040	3000	3000	3000	3000	3000
	0.237	10.79	28.49	Std.	1900	2210	3000	3000	3000	3000	3000
	0.250	11.35	29.96		2000	2330	3000	3000	3000	3000	3000
	0.281	12.66	33.43		2250	2620	3000	3000	3000	3000	3000
	0.312	13.96	36.84		2500	2800	3000	3000	3000	3000	3000
	0.337	14.98	39.56	XS	2700	2800	3000	3000	3000	3000	3000
	0.438	19.00	50.16		2800	2800	3000	3000	3000	3000	3000
	0.531	22.51	59.42		2800	2800	3000	3000	3000	—	—
	0.674	27.54	72.71	XXS	2800	2800	3000	3000	3000	—	—
5.563	0.094	5.49	14.49		610	710	1060	1170	1320	—	—
	0.109	6.35	16.76		710	820	1230	1350	1530	—	—
	0.125	7.26	19.17		810	940	1420	1550	1750	—	—
	0.141	8.16	21.56		910	1060	1600	1750	1980	—	—
	0.156	9.01	23.78		1010	1180	1770	1930	2190	—	—
	0.188	10.79	28.49		1220	1420	2130	2330	2640	—	—
	0.219	12.50	33.00		1420	1650	2480	2720	3000	—	—
	0.258	14.62	38.59	Std.	1670	1950	2920	3000	3000	3000	3000
	0.281	15.85	41.85		1820	2120	3000	3000	3000	3000	3000
	0.312	17.50	46.19		2020	2360	3000	3000	3000	3000	3000
	0.344	19.17	50.62		2230	2600	3000	3000	3000	3000	3000
	0.375	20.78	54.85	XS	2430	2800	3000	3000	3000	3000	3000
	0.500	27.04	71.38		2800	2800	3000	3000	3000	3000	3000
	0.625	32.96	87.02		2800	2800	3000	3000	3000	—	—
	0.750	38.55	101.78	XXS	2800	2800	3000	3000	3000	—	—
	0.875	43.81	115.66		2800	2800	3000	3000	3000	—	—
	0.938	46.33	122.32		2800	2800	3000	3000	3000	—	—
6.625	0.094	6.56	17.31		640	740	890	980	1110	—	—
	0.109	7.59	20.03		740	860	1040	1140	1280	—	—
	0.125	8.68	22.91		850	990	1190	1300	1470	—	—
	0.141	9.76	25.78		960	1120	1340	1470	1660	1790	1920
	0.156	10.78	28.45		1060	1240	1480	1620	1840	1980	2120
	0.172	11.85	31.29		1170	1360	1640	1790	2030	2180	2340
	0.188	12.92	34.12		1280	1490	1790	1960	2210	2380	2550
	0.203	13.92	36.76		1380	1610	1930	2110	2390	2570	2760
	0.219	14.98	39.56		1490	1740	2080	2280	2580	2780	2980
	0.250	17.02	44.94		1700	1980	2380	2600	2940	3000	3000
	0.280	18.97	50.09	Std.	1900	2220	2660	2920	3000	3000	3000
	0.312	21.04	55.53		2120	2470	2970	3000	3000	3000	3000
	0.344	23.08	60.92		2340	2730	3000	3000	3000	3000	3000
	0.375	25.03	66.08		2550	2800	3000	3000	3000	3000	3000
	0.432	28.57	75.43	XS	2800	2800	3000	3000	3000	3000	3000
	0.500	32.71	86.35		2800	2800	3000	3000	3000	3000	3000

Note: Some outside diameters, walls and grades are listed for information only and are not necessarily regular production items. Non-production sizes may be negotiated upon inquiry.

Some outside diameters, walls and grades are non-API.

*See notes at end of table on page 33.

Fig. 3N-3 (Cont'd) (Courtesy U.S. Steel)

Line Pipe Data
Seamless, Electric Resistance Weld

Internal Pressure at Minimum Yield Strength*—psi							Ultimate Bursting Pressure*—psi							Actual OD Inches
Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60	Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60	
2080	2430	2910	3190	3610	3880	4160	3330	4160	4160	4370	4580	4920	5200	
2290	2680	3210	3520	3980	4280	4590	3670	4590	4590	4820	5050	5430	5730	
2510	2920	3510	3840	4340	4680	5010	4010	5010	5010	5260	5510	5930	6270	
2710	3160	3790	4150	4690	5050	5410	4330	5410	5410	5680	5950	6410	6770	
2920	3410	4090	4480	5060	5450	5840	4670	5840	5840	6130	6420	6910	7300	
3160	3690	4420	4850	5480	5900	6320	5060	6320	6320	6640	6950	7480	7900	
3330	3890	4670	5110	5780	6220	6670	5330	6670	6670	7000	7330	7890	8330	
3750	4370	5250	5740	6490	6990	7490	5990	7490	7490	7870	8240	8870	9370	4½
4160	4850	5820	6380	7210	7770	8320	6660	8320	8320	8740	9150	9850	10400	
4490	5240	6290	6890	7790	8390	8990	7190	8990	8990	9440	9890	10630	11230	
5840	6810	8180	8950	10120	10900	11680	9340	11680	11680	12260	12850	13820	14600	
7080	8260	9910	10860	12270	—	—	11330	14160	14160	14870	15580	—	—	
8990	10480	12580	13780	15580	—	—	14380	17970	17970	18870	19770	—	—	
1010	1180	1420	1550	1760	—	—	1620	2030	2030	2130	2220	—	—	
1180	1370	1650	1800	2040	—	—	1880	2350	2350	2470	2590	—	—	
1350	1570	1890	2070	2340	—	—	2160	2700	2700	2830	2970	—	—	
1520	1770	2130	2330	2640	—	—	2430	3040	3040	3190	3350	—	—	
1680	1960	2360	2580	2920	—	—	2690	3370	3370	3530	3700	—	—	
2030	2370	2840	3110	3510	—	—	3240	4060	4060	4260	4460	—	—	
2360	2760	3310	3620	4090	—	—	3780	4720	4720	4960	5200	—	—	
2780	3250	3900	4270	4820	5190	5570	4450	5570	5570	5840	6120	6590	6960	
3030	3540	4240	4650	5250	5660	6060	4850	6060	6060	6360	6670	7170	7580	
3370	3930	4710	5160	5830	6280	6730	5380	6730	6730	7070	7400	7960	8410	
3710	4330	5190	5690	6430	6930	7420	5940	7420	7420	7790	8160	8780	9280	
4040	4720	5660	6200	7010	7550	8090	6470	8090	8090	8490	8900	9570	10110	5⁹⁄₁₆
5390	6290	7550	8270	9350	10070	10790	8630	10790	10790	11320	11860	12760	13480	
6740	7860	9440	10340	11680	—	—	10790	13480	13480	14160	14830	—	—	
8090	9440	11320	12400	14020	—	—	12940	16180	16180	16990	17800	—	—	
9440	11010	13210	14470	16360	—	—	15100	18870	18870	19820	20760	—	—	
10120	11800	14160	15510	17540	—	—	16190	20230	20230	21250	22260	—	—	
850	990	1190	1310	1480	—	—	1360	1700	1700	1790	1870	—	—	
990	1150	1380	1510	1710	—	—	1580	1970	1970	2070	2170	—	—	
1130	1320	1580	1740	1960	—	—	1810	2260	2260	2380	2490	—	—	
1280	1490	1790	1960	2210	2380	2550	2040	2550	2550	2680	2810	3020	3190	
1410	1650	1980	2170	2450	2640	2830	2260	2830	2830	2970	3110	3340	3530	
1560	1820	2180	2390	2700	2910	3120	2490	3120	3120	3270	3430	3690	3890	
1700	1990	2380	2610	2950	3180	3410	2720	3410	3410	3580	3750	4030	4260	
1840	2140	2570	2820	3190	3430	3680	2940	3680	3680	3860	4040	4350	4600	6⅝
1980	2310	2780	3040	3440	3700	3970	3170	3970	3970	4170	4360	4690	4960	
2260	2640	3170	3470	3920	4230	4530	3620	4530	4530	4750	4980	5360	5660	
2540	2960	3550	3890	4400	4730	5070	4060	5070	5070	5330	5580	6000	6340	
2830	3300	3960	4330	4900	5270	5650	4520	5650	5650	5930	6220	6690	7060	
3120	3630	4360	4780	5400	5820	6230	4980	6230	6230	6540	6850	7370	7790	
3400	3960	4750	5210	5890	6340	6790	5430	6790	6790	7130	7470	8040	8490	
3910	4560	5480	6000	6780	7300	7820	6260	7820	7820	8220	8610	9260	9780	
4530	5280	6340	6940	7850	8450	9060	7250	9060	9060	9510	9960	10720	11320	

Fig. 3N-3 (Cont'd) (Courtesy U.S. Steel)

Line Pipe Data

Seamless, Electric Resistance Weld

Actual OD Inches	Wall Thickness Inches	Weight Lb/Ft Plain End	Weight Tons/Mile	Class	Mill Test Pressure*—psi						
					Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60
6.625	0.562	36.39	96.07	XXS	2800	2800	3000	3000	3000	—	—
	0.625	40.05	105.73		2800	2800	3000	3000	3000	—	—
	0.719	45.35	119.73		2800	2800	3000	3000	3000	—	—
	0.864	53.16	140.34		2800	2800	3000	3000	3000	—	—
	1.000	60.08	158.60		2800	2800	3000	3000	3000	—	—
	1.125	66.08	174.46		2800	2800	3000	3000	3000	—	—
8.625	0.109	9.91	26.17	Std.	570	660	800	870	990	1060	1140
	0.125	11.35	29.96		650	760	910	1000	1130	1220	1300
	0.141	12.78	33.73		740	860	1030	1130	1280	1370	1470
	0.156	14.11	37.25		810	950	1140	1250	1410	1520	1630
	0.172	15.53	40.99		900	1050	1260	1380	1560	1680	1790
	0.188	16.94	44.72		980	1140	1370	1500	1700	1830	1960
	0.203	18.26	48.20		1060	1240	1480	1620	1840	1980	2120
	0.219	19.66	51.90		1140	1330	1600	1750	1980	2130	2290
	0.250	22.36	59.03		1300	1520	1830	2000	2260	2430	2610
	0.277	24.70	65.20		1450	1690	2020	2220	2510	2700	2890
	0.312	27.70	73.13		1630	1900	2280	2500	2820	3000	3000
	0.322	28.55	75.38		1680	1960	2350	2580	2910	3000	3000
	0.344	30.42	80.32	XS	1790	2090	2510	2750	3000	3000	3000
	0.375	33.04	87.23		1960	2280	2740	3000	3000	3000	3000
	0.406	35.64	94.08		2120	2470	2970	3000	3000	3000	3000
	0.438	38.30	101.11		2290	2670	3000	3000	3000	3000	3000
	0.500	43.39	114.54		2610	2800	3000	3000	3000	3000	3000
	0.562	48.40	127.76		2800	2800	3000	3000	3000	—	—
	0.594	50.95	134.50		2800	2800	3000	3000	3000	—	—
	0.625	53.40	140.98		2800	2800	3000	3000	3000	—	—
	0.719	60.71	160.27	XXS	2800	2800	3000	3000	3000	—	—
	0.812	67.76	178.87		2800	2800	3000	3000	3000	—	—
	0.875	72.42	191.20		2800	2800	3000	3000	3000	—	—
	0.906	74.69	197.18		2800	2800	3000	3000	3000	—	—
	1.000	81.44	214.99		2800	2800	3000	3000	3000	—	—
	1.125	90.11	237.90		2800	2800	3000	3000	3000	—	—
10.750	0.125	14.18	37.45	Std.	520	610	830	910	1030	1110	1190
	0.141	15.98	42.18		590	690	940	1030	1160	1250	1340
	0.156	17.65	46.60		650	760	1040	1130	1280	1380	1480
	0.172	19.43	51.30		720	840	1140	1250	1410	1520	1630
	0.188	21.21	55.99		790	920	1250	1370	1550	1660	1780
	0.203	22.87	60.37		850	990	1350	1480	1670	1800	1930
	0.219	24.63	65.03		920	1070	1450	1590	1800	1940	2080
	0.250	28.04	74.01		1050	1220	1660	1820	2060	2210	2370
	0.279	31.20	82.37		1170	1360	1850	2030	2290	2470	2650
	0.307	34.24	90.39		1290	1500	2040	2230	2520	2720	2910
	0.344	38.23	100.93		1440	1680	2280	2500	2830	3000	3000
	0.365	40.48	106.87		1530	1780	2420	2660	3000	3000	3000
	0.438	48.24	127.35	XS	1830	2140	2910	3000	3000	3000	3000
	0.500	54.74	144.50		2090	2440	3000	3000	3000	3000	3000
	0.562	61.15	161.44		2350	2740	3000	3000	3000	—	—
	0.594	64.43	170.09		2490	2800	3000	3000	3000	—	—

Note: Some outside diameters, walls and grades are listed for information only and are not necessarily regular production items. Non-production sizes may be negotiated upon inquiry.

Some outside diameters, walls and grades are non-API.

*See notes at end of table on page 33.

Fig. 3N-3 (Cont'd) (Courtesy U.S. Steel)

Line Pipe Data
Seamless, Electric Resistance Weld

Internal Pressure at Minimum Yield Strength*—psi							Ultimate Bursting Pressure*—psi							Actual OD Inches
Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60	Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60	
5090	5940	7130	7800	8820	—	—	8140	10180	10180	10690	11200	—	—	
5660	6600	7920	8680	9810	—	—	9060	11320	11320	11890	12450	—	—	
6510	7600	9120	9980	11290	—	—	10420	13020	13020	13670	14330	—	—	
7820	9130	10950	12000	13560	—	—	12520	15650	15650	16430	17210	—	—	6⅝
9060	10570	12680	13890	15700	—	—	14490	18110	18110	19020	19920	—	—	
10190	11890	14260	15620	17660	—	—	16300	20380	20380	21400	22420	—	—	
760	880	1060	1160	1310	1420	1520	1210	1520	1520	1590	1670	1790	1900	
870	1010	1220	1330	1510	1620	1740	1390	1740	1740	1830	1910	2060	2170	
980	1140	1370	1500	1700	1830	1960	1570	1960	1960	2060	2160	2320	2450	
1090	1270	1520	1660	1880	2030	2170	1740	2170	2170	2280	2390	2570	2710	
1200	1400	1680	1830	2070	2230	2390	1910	2390	2390	2510	2630	2830	2990	
1310	1530	1830	2010	2270	2440	2620	2090	2620	2620	2750	2880	3100	3270	
1410	1650	1980	2170	2450	2640	2820	2260	2820	2820	2970	3110	3340	3530	
1520	1780	2130	2340	2640	2840	3050	2440	3050	3050	3200	3350	3610	3810	
1740	2030	2430	2670	3010	3250	3480	2780	3480	3480	3650	3830	4120	4350	
1930	2250	2700	2950	3340	3600	3850	3080	3850	3850	4050	4240	4560	4820	
2170	2530	3040	3330	3760	4050	4340	3470	4340	4340	4560	4770	5140	5430	
2240	2610	3140	3430	3880	4180	4480	3580	4480	4480	4700	4930	5300	5600	
2390	2790	3350	3670	4150	4470	4790	3830	4790	4790	5030	5260	5660	5980	
2610	3040	3650	4000	4520	4870	5220	4170	5220	5220	5480	5740	6170	6520	8⅝
2820	3300	3950	4330	4900	5270	5650	4520	5650	5650	5930	6210	6680	7060	
3050	3550	4270	4670	5280	5690	6090	4880	6090	6090	6400	6700	7210	7620	
3480	4060	4870	5330	6030	6490	6960	5570	6960	6960	7300	7650	8230	8700	
3910	4560	5470	5990	6780	—	—	6260	7820	7820	8210	8600	—	—	
4130	4820	5790	6340	7160	—	—	6610	8260	8260	8680	9090	—	—	
4350	5070	6090	6670	7540	—	—	6960	8700	8700	9130	9570	—	—	
5000	5840	7000	7670	8670	—	—	8000	10000	10000	10500	11000	—	—	
5650	6590	7910	8660	9790	—	—	9040	11300	11300	11860	12430	—	—	
6090	7100	8520	9330	10550	—	—	9740	12170	12170	12780	13390	—	—	
6300	7350	8820	9660	10920	—	—	10080	12610	12610	13240	13870	—	—	
6960	8120	9740	10670	12060	—	—	11130	13910	13910	14610	15300	—	—	
7830	9130	10960	12000	13570	—	—	12520	15650	15650	16430	17220	—	—	
700	810	980	1070	1210	1300	1400	1120	1400	1400	1470	1530	1650	1740	
790	920	1100	1210	1360	1470	1570	1260	1570	1570	1650	1730	1860	1970	
870	1020	1220	1340	1510	1630	1740	1390	1740	1740	1830	1920	2060	2180	
960	1120	1340	1470	1660	1790	1920	1540	1920	1920	2020	2110	2270	2400	
1050	1220	1470	1610	1820	1960	2100	1680	2100	2100	2200	2310	2480	2620	
1130	1320	1590	1740	1960	2110	2270	1810	2270	2270	2380	2490	2680	2830	
1220	1430	1710	1870	2120	2280	2440	1960	2440	2440	2570	2690	2890	3060	
1400	1630	1950	2140	2420	2600	2790	2230	2790	2790	2930	3070	3300	3490	
1560	1820	2180	2390	2700	2910	3110	2490	3110	3110	3270	3430	3690	3890	10¾
1710	2000	2400	2630	2970	3200	3430	2740	3430	3430	3600	3770	4060	4280	
1920	2240	2690	2940	3330	3580	3840	3070	3840	3840	4030	4220	4540	4800	
2040	2380	2850	3120	3530	3800	4070	3260	4070	4070	4280	4480	4820	5090	
2440	2850	3420	3750	4240	4560	4890	3910	4890	4890	5130	5380	5790	6110	
2790	3260	3910	4280	4840	5210	5580	4470	5580	5580	5860	6140	6600	6980	
3140	3660	4390	4810	5440	—	—	5020	6270	6270	6590	6900	—	—	
3320	3870	4640	5080	5750	—	—	5300	6630	6630	6960	7290	—	—	

Fig. 3N-3 (Cont'd) (Courtesy U.S. Steel)

Line Pipe Data

Seamless, Electric Resistance Weld

Actual OD Inches	Wall Thickness Inches	Weight Lb/Ft Plain End	Weight Tons/Mile	Class	Mill Test Pressure*—psi						
					Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60
10.750	0.625	67.58	178.42		2620	2800	3000	3000	3000	—	—
	0.719	77.03	203.35		2800	2800	3000	3000	3000	—	—
	0.812	86.18	227.53		2800	2800	3000	3000	3000	—	—
	0.844	89.29	235.73		2800	2800	3000	3000	3000	—	—
	1.000	104.13	274.90		2800	2800	3000	3000	3000	—	—
	1.125	115.64	305.30		2800	2800	3000	3000	3000	—	—
	1.438	143.01	377.55		2800	2800	3000	3000	3000	—	—
12.750	0.141	18.99	50.13		500	580	790	860	980	—	—
	0.156	20.98	55.39		550	640	870	960	1080	1160	1250
	0.172	23.11	61.00		610	710	960	1050	1190	1280	1380
	0.188	25.22	66.59		660	770	1050	1150	1300	1400	1500
	0.203	27.20	71.81		720	840	1140	1250	1410	1520	1620
	0.219	29.31	77.38		770	900	1230	1340	1520	1640	1750
	0.250	33.38	88.11		880	1030	1400	1530	1730	1870	2000
	0.281	37.42	98.79		990	1160	1570	1720	1950	2100	2250
	0.312	41.45	109.42		1100	1280	1750	1910	2160	2330	2500
	0.330	43.77	115.56		1160	1360	1850	2020	2290	2460	2640
	0.344	45.58	120.33		1210	1420	1930	2110	2390	2570	2750
	0.375	49.56	130.84	Std.	1320	1540	2100	2300	2600	2800	3000
	0.406	53.52	141.30		1430	1670	2270	2490	2810	3000	3000
	0.438	57.59	152.05		1550	1800	2450	2690	3000	3000	3000
	0.500	65.42	172.70	XS	1760	2060	2800	3000	3000	3000	3000
	0.562	73.15	193.13		1980	2310	3000	3000	3000	—	—
	0.625	80.93	213.67		2210	2570	3000	3000	3000	—	—
	0.688	88.63	233.98		2430	2800	3000	3000	3000	—	—
	0.750	96.12	253.76		2650	2800	3000	3000	3000	—	—
	0.844	107.32	283.32		2800	2800	3000	3000	3000	—	—
	1.000	125.49	331.29		2800	2800	3000	3000	3000	—	—
	1.125	139.67	368.74		2800	2800	3000	3000	3000	—	—
	1.312	160.27	423.12		2800	2800	3000	3000	3000	—	—
	1.500	180.22	475.79		2800	2800	3000	3000	3000	—	—
	1.594	189.92	501.39		2800	2800	3000	3000	3000	—	—
	1.625	193.07	509.72		2800	2800	3000	3000	3000	—	—
14.000	0.188	27.73	73.21		600	700	960	1050	1190	1280	1370
	0.203	29.91	78.97		650	760	1040	1130	1280	1380	1480
	0.210	30.93	81.65		670	790	1070	1170	1330	1430	1530
	0.219	32.23	85.09		700	820	1120	1220	1380	1490	1600
	0.250	36.71	96.92		800	940	1280	1400	1580	1700	1820
	0.281	41.17	108.69		900	1050	1430	1570	1770	1910	2050
	0.312	45.61	120.41		1000	1170	1590	1740	1970	2120	2270
	0.344	50.17	132.45		1110	1290	1750	1920	2170	2340	2510
	0.375	54.57	144.06	Std.	1210	1410	1910	2090	2370	2550	2730
	0.406	58.94	155.61		1300	1520	2070	2270	2560	—	—
	0.438	63.44	167.48		1410	1640	2230	2450	2770	—	—
	0.469	67.78	178.93		1510	1760	2390	2620	2960	—	—
	0.500	72.09	190.32	XS	1610	1880	2550	2790	3000	—	—
	0.562	80.66	212.93		1810	2110	2870	3000	3000	—	—
	0.594	85.05	224.52		1910	2230	3000	3000	3000	—	—
	0.625	89.28	235.69		2010	2340	3000	3000	3000	—	—

Note: Some outside diameters, walls and grades are listed for information only and are not necessarily regular production items. Non-production sizes may be negotiated upon inquiry.

Some outside diameters, walls and grades are non-API.

*See notes at end of table on page 33.

Fig. 3N-3 (Cont'd) (Courtesy U.S. Steel)

Line Pipe Data

Seamless, Electric Resistance Weld

Internal Pressure at Minimum Yield Strength*—psi							Ultimate Bursting Pressure*—psi							Actual OD Inches
Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60	Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60	
3490	4070	4880	5350	6050	—	—	5580	6980	6980	7330	7670	—	—	
4010	4680	5620	6150	6960	—	—	6420	8030	8030	8430	8830	—	—	
4530	5290	6340	6950	7860	—	—	7250	9060	9060	9520	9970	—	—	
4710	5500	6590	7220	8170	—	—	7540	9420	9420	9890	10360	—	—	10¾
5580	6510	7810	8560	9670	—	—	8930	11160	11160	11720	12280	—	—	
6280	7330	8790	9630	10880	—	—	10050	12560	12560	13190	13810	—	—	
8030	9360	11240	12310	13910	—	—	12840	16050	16050	16850	17660	—	—	
660	770	930	1020	1150	—	—	1060	1330	1330	1390	1460	—	—	
730	860	1030	1130	1270	1370	1470	1170	1470	1470	1540	1620	1740	1840	
810	940	1130	1240	1400	1510	1620	1300	1620	1620	1700	1780	1920	2020	
880	1030	1240	1360	1530	1650	1770	1420	1770	1770	1860	1950	2090	2210	
960	1110	1340	1460	1660	1780	1910	1530	1910	1910	2010	2100	2260	2390	
1030	1200	1440	1580	1790	1920	2060	1650	2060	2060	2160	2270	2440	2580	
1180	1370	1650	1800	2040	2200	2350	1880	2350	2350	2470	2590	2780	2940	
1320	1540	1850	2030	2290	2470	2640	2120	2640	2640	2780	2910	3130	3310	
1470	1710	2060	2250	2540	2740	2940	2350	2940	2940	3080	3230	3470	3670	
1550	1810	2170	2380	2690	2900	3110	2480	3110	3110	3260	3420	3680	3880	
1620	1890	2270	2480	2810	3020	3240	2590	3240	3240	3400	3560	3830	4050	
1760	2060	2470	2710	3060	3290	3530	2820	3530	3530	3710	3880	4180	4410	
1910	2230	2670	2930	3310	3570	3820	3060	3820	3820	4010	4200	4520	4780	
2060	2400	2890	3160	3570	3850	4120	3300	4120	4120	4330	4530	4880	5150	12¾
2350	2750	3290	3610	4080	4390	4710	3760	4710	4710	4940	5180	5570	5880	
2640	3090	3700	4060	4580	—	—	4230	5290	5290	5550	5820	—	—	
2940	3430	4120	4510	5100	—	—	4710	5880	5880	6180	6470	—	—	
3240	3780	4530	4960	5610	—	—	5180	6480	6480	6800	7120	—	—	
3530	4120	4940	5410	6120	—	—	5650	7060	7060	7410	7760	—	—	
3970	4630	5560	6090	6880	—	—	6350	7940	7940	8340	8740	—	—	
4710	5490	6590	7220	8160	—	—	7530	9410	9410	9880	10350	—	—	
5290	6180	7410	8120	9180	—	—	8470	10590	10590	11120	11650	—	—	
6170	7200	8640	9470	10700	—	—	9880	12350	12350	12970	13580	—	—	
7060	8240	9880	10820	12240	—	—	11290	14120	14120	14820	15530	—	—	
7500	8750	10500	11500	13000	—	—	12000	15000	15000	15750	16500	—	—	
7650	8920	10710	11730	13250	—	—	12240	15290	15290	16060	16820	—	—	
810	940	1130	1240	1400	1500	1610	1290	1610	1610	1690	1770	1910	2010	
870	1010	1220	1330	1510	1620	1740	1390	1740	1740	1830	1910	2060	2170	
900	1050	1260	1380	1560	1680	1800	1440	1800	1800	1890	1980	2130	2250	
940	1090	1310	1440	1630	1750	1880	1500	1880	1880	1970	2060	2220	2350	
1070	1250	1500	1640	1860	2000	2140	1710	2140	2140	2250	2360	2540	2680	
1200	1400	1690	1850	2090	2250	2410	1930	2410	2410	2530	2650	2850	3010	
1340	1560	1870	2050	2320	2500	2670	2140	2670	2670	2810	2940	3160	3340	
1470	1720	2060	2260	2560	2750	2950	2360	2950	2950	3100	3240	3490	3690	
1610	1880	2250	2460	2790	3000	3210	2570	3210	3210	3380	3540	3800	4020	14
1740	2030	2440	2670	3020	—	—	2780	3480	3480	3650	3830	—	—	
1880	2190	2630	2880	3250	—	—	3000	3750	3750	3940	4130	—	—	
2010	2350	2810	3080	3480	—	—	3220	4020	4020	4220	4420	—	—	
2140	2500	3000	3290	3710	—	—	3430	4290	4290	4500	4710	—	—	
2410	2810	3370	3690	4170	—	—	3850	4820	4820	5060	5300	—	—	
2550	2970	3560	3900	4410	—	—	4070	5090	5090	5350	5600	—	—	
2680	3130	3750	4110	4640	—	—	4290	5360	5360	5630	5890	—	—	

Fig. 3N-3 (Cont'd) (Courtesy U.S. Steel)

Line Pipe Data

Seamless, Electric Resistance Weld

Actual OD Inches	Wall Thickness Inches	Weight Lb/Ft Plain End	Weight Tons/Mile	Class	Mill Test Pressure*—psi						
					Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60
14.000	0.688	97.81	258.23		2210	2580	3000	3000	3000	—	—
	0.750	106.13	280.19		2410	2800	3000	3000	3000	—	—
	0.812	114.37	301.93		2610	2800	3000	3000	3000	—	—
	0.938	130.85	345.45		2800	2800	3000	3000	3000	—	—
	1.094	150.79	398.09		2800	2800	3000	3000	3000	—	—
	1.250	170.21	449.36		2800	2800	3000	3000	3000	—	—
	1.406	189.11	499.26		2800	2800	3000	3000	3000	—	—
	1.656	218.32	576.36		2800	2800	3000	3000	3000	—	—
16.000	0.188	31.75	83.81		530	620	840	920	1040	1120	1200
	0.203	34.25	90.42		570	670	910	990	1120	1210	1290
	0.219	36.91	97.44		620	720	980	1070	1210	1300	1400
	0.250	42.05	111.02		700	820	1120	1220	1380	1490	1590
	0.281	47.17	124.54		790	920	1250	1370	1550	1670	1790
	0.312	52.27	138.01		880	1020	1390	1520	1720	1860	1990
	0.344	57.52	151.85		970	1130	1540	1680	1900	2050	2190
	0.375	62.58	165.21	Std.	1050	1230	1670	1830	2070	2230	2390
	0.406	67.62	178.51		1140	1330	1810	1980	2240	—	—
	0.438	72.80	192.18		1230	1440	1950	2140	2420	—	—
	0.469	77.79	205.37		1320	1540	2090	2290	2590	—	—
	0.500	82.77	218.51	XS	1410	1640	2230	2440	2760	—	—
	0.562	92.66	244.63		1580	1840	2510	2750	3000	—	—
	0.625	102.63	270.94		1760	2050	2790	3000	3000	—	—
	0.656	107.50	283.80		1840	2150	2930	3000	3000	—	—
	0.688	112.51	297.03		1940	2260	3000	3000	3000	—	—
	0.750	122.15	322.48		2110	2460	3000	3000	3000	—	—
	0.812	131.71	347.72		2280	2660	3000	3000	3000	—	—
	0.844	136.61	360.66		2370	2770	3000	3000	3000	—	—
	1.031	164.82	435.14		2800	2800	3000	3000	3000	—	—
	1.219	192.43	508.02		2800	2800	3000	3000	3000	—	—
	1.438	223.64	590.41		2800	2800	3000	3000	3000	—	—
	1.594	245.25	647.45		2800	2800	3000	3000	3000	—	—
18.000	0.219	41.59	109.79		550	640	870	950	1080	1160	1240
	0.250	47.39	125.12		620	730	990	1090	1230	1320	1420
	0.281	53.18	140.39		700	820	1110	1220	1380	1490	1590
	0.312	58.94	155.60		780	910	1240	1360	1530	1650	1770
	0.344	64.87	171.25		860	1000	1360	1490	1690	1820	1950
	0.375	70.59	186.35	Std.	940	1090	1490	1630	1840	1980	2130
	0.406	76.29	201.40		1010	1180	1610	1760	1990	—	—
	0.438	82.15	216.88		1100	1280	1740	1900	2150	—	—
	0.469	87.81	231.82		1170	1370	1860	2040	2300	—	—
	0.500	93.45	246.71	XS	1250	1460	1980	2170	2460	—	—
	0.562	104.67	276.32		1400	1640	2230	2440	2760	—	—
	0.625	115.98	306.18		1560	1820	2480	2720	3000	—	—
	0.688	127.21	335.82		1720	2010	2730	2990	3000	—	—
	0.750	138.17	364.78		1880	2190	2980	3000	3000	—	—

Note: Some outside diameters, walls and grades are listed for information only and are not necessarily regular production items. Non-production sizes may be negotiated upon inquiry.

Some outside diameters, walls and grades are non-API.

*See notes at end of table on page 33.

Fig. 3N-3 (Cont'd) (Courtesy U.S. Steel)

Line Pipe Data
Seamless, Electric Resistance Weld

Internal Pressure at Minimum Yield Strength*—psi							Ultimate Bursting Pressure*—psi							Actual OD Inches
Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60	Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60	
2950	3440	4130	4520	5110	—	—	4720	5900	5900	6190	6490	—	—	
3210	3750	4500	4930	5570	—	—	5140	6430	6430	6750	7070	—	—	
3480	4060	4870	5340	6030	—	—	5570	6960	6960	7310	7660	—	—	
4020	4690	5630	6160	6970	—	—	6430	8040	8040	8440	8840	—	—	
4690	5470	6560	7190	8130	—	—	7500	9380	9380	9850	10310	—	—	14
5360	6250	7500	8210	9290	—	—	8570	10710	10710	11250	11790	—	—	
6030	7030	8440	9240	10440	—	—	9640	12050	12050	12650	13260	—	—	
7100	8280	9940	10880	12300	—	—	11360	14190	14190	14900	15610	—	—	
700	820	990	1080	1220	1320	1410	1130	1410	1410	1480	1550	1670	1760	
760	890	1070	1170	1320	1420	1520	1220	1520	1520	1600	1670	1800	1900	
820	960	1150	1260	1420	1530	1640	1310	1640	1640	1720	1810	1940	2050	
940	1090	1310	1440	1630	1750	1880	1500	1880	1880	1970	2060	2220	2340	
1050	1230	1480	1620	1830	1970	2110	1690	2110	2110	2210	2320	2490	2630	
1170	1360	1640	1790	2030	2180	2340	1870	2340	2340	2460	2570	2770	2930	
1290	1500	1810	1980	2240	2410	2580	2060	2580	2580	2710	2840	3050	3230	
1410	1640	1970	2160	2440	2630	2810	2250	2810	2810	2950	3090	3330	3520	
1520	1780	2130	2330	2640	—	—	2440	3050	3050	3200	3350	—	—	
1640	1920	2300	2520	2850	—	—	2630	3290	3290	3450	3610	—	—	
1760	2050	2460	2700	3050	—	—	2810	3520	3520	3690	3870	—	—	
1880	2190	2630	2880	3250	—	—	3000	3750	3750	3940	4130	—	—	
2110	2460	2950	3230	3650	—	—	3370	4220	4220	4430	4640	—	—	16
2340	2730	3280	3590	4060	—	—	3750	4690	4690	4920	5160	—	—	
2460	2870	3440	3770	4260	—	—	3940	4920	4920	5170	5410	—	—	
2580	3010	3610	3960	4470	—	—	4130	5160	5160	5420	5680	—	—	
2810	3280	3940	4310	4880	—	—	4500	5630	5630	5910	6190	—	—	
3040	3550	4260	4670	5280	—	—	4870	6090	6090	6390	6700	—	—	
3160	3690	4430	4850	5490	—	—	5060	6330	6330	6650	6960	—	—	
3870	4510	5410	5930	6700	—	—	6190	7730	7730	8120	8510	—	—	
4570	5330	6400	7010	7920	—	—	7310	9140	9140	9600	10060	—	—	
5390	6290	7550	8270	9350	—	—	8630	10780	10780	11320	11860	—	—	
5980	6970	8370	9170	10360	—	—	9560	11950	11950	12550	13150	—	—	
730	850	1020	1120	1270	1360	1460	1170	1460	1460	1530	1610	1730	1830	
830	970	1170	1280	1440	1560	1670	1330	1670	1670	1750	1830	1970	2080	
940	1090	1310	1440	1620	1750	1870	1500	1870	1870	1970	2060	2220	2340	
1040	1210	1460	1590	1800	1940	2080	1660	2080	2080	2180	2290	2460	2600	
1150	1340	1610	1760	1990	2140	2290	1830	2290	2290	2410	2520	2710	2870	
1250	1460	1750	1920	2170	2330	2500	2000	2500	2500	2630	2750	2960	3130	
1350	1580	1890	2080	2350	—	—	2170	2710	2710	2840	2980	—	—	18
1460	1700	2040	2240	2530	—	—	2340	2920	2920	3070	3210	—	—	
1560	1820	2190	2400	2710	—	—	2500	3130	3130	3280	3440	—	—	
1670	1940	2330	2560	2890	—	—	2670	3330	3330	3500	3670	—	—	
1870	2190	2620	2870	3250	—	—	3000	3750	3750	3930	4120	—	—	
2080	2430	2920	3190	3610	—	—	3330	4170	4170	4380	4580	—	—	
2290	2680	3210	3520	3980	—	—	3670	4590	4590	4820	5050	—	—	
2500	2920	3500	3830	4330	—	—	4000	5000	5000	5250	5500	—	—	

Fig. 3N-3 (Cont'd) (Courtesy U.S. Steel)

Line Pipe Data

Seamless, Electric Resistance Weld, Double Submerged Arc Weld

Actual OD Inches	Wall Thickness Inches	Weight Lb./Ft. Plain End	Weight Tons/Mile	Class	Mill Test Pressure*—psi						
					Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60
18.000	0.812	149.06	393.51		2030	2370	3000	3000	3000	—	—
	0.938	170.92	451.24		2340	2740	3000	3000	3000	—	—
	1.156	207.96	549.01		2800	2800	3000	3000	3000	—	—
	1.375	244.14	644.52		2800	2800	3000	3000	3000	—	—
	1.562	274.22	723.94		2800	2800	3000	3000	3000	—	—
20.000	0.219	46.27	122.14		490	570	830	910	1020	1100	1180
	0.250	52.73	139.21		560	660	950	1040	1170	1260	1350
	0.281	59.18	156.23		630	740	1060	1160	1320	1420	1520
	0.312	65.60	173.19		700	820	1180	1290	1460	1570	1680
	0.344	72.21	190.65		770	900	1300	1420	1610	1730	1860
	0.375	78.60	207.50	Std.	840	980	1420	1550	1760	1890	2030
	0.406	84.96	224.30		910	1070	1530	1680	1900	—	—
	0.438	91.51	241.58		990	1150	1660	1810	2050	—	—
	0.469	97.83	258.27		1060	1230	1770	1940	2190	—	—
	0.500	104.13	274.90	XS	1120	1310	1890	2070	2340	—	—
	0.562	116.67	308.01		1260	1480	2120	2330	2630	—	—
	0.594	123.11	325.01		1340	1560	2250	2460	2750	—	—
	0.625	129.33	341.43		1410	1640	2360	2590	2750	—	—
	0.688	141.90	374.62		1550	1810	2600	2750	2750	—	—
	0.750	154.19	407.07		1690	1970	2750	2750	2750	—	—
	0.812	166.40	439.30		1830	2130	2750	2750	2750	—	—
	1.031	208.87	551.41		2320	2710	2750	2750	2750	—	—
	1.281	256.10	676.09		2750	2750	2750	2750	2750	—	—
22.000	0.250	58.07	153.31		510	600	860	940	1060	1150	1230
	0.281	65.18	172.08		570	670	970	1060	1200	1290	1380
	0.312	72.27	190.79		640	740	1070	1170	1330	1430	1530
	0.344	79.56	210.04		700	820	1180	1290	1460	1580	1690
	0.375	86.61	228.65		770	890	1290	1410	1600	1720	1840
	0.406	93.63	247.19		830	970	1400	1530	1730	—	—
	0.438	100.86	266.28		900	1050	1510	1650	1860	—	—
	0.469	107.85	284.72		960	1120	1610	1770	2000	—	—
	0.500	114.81	303.10		1020	1190	1720	1880	2130	—	—
	0.562	128.67	339.70		1150	1340	1930	2120	2390	—	—
	0.625	142.68	376.67		1280	1490	2150	2350	2500	—	—
	0.688	156.60	413.42		1410	1640	2360	2500	2500	—	—
	0.750	170.21	449.36		1530	1790	2500	2500	2500	—	—
	0.812	183.75	485.09		1660	1940	2500	2500	2500	—	—
	0.875	197.41	521.17		1790	2090	2500	2500	2500	—	—
	1.125	250.81	662.15		2300	2500	2500	2500	2500	—	—
	1.250	277.01	731.31		2500	2500	2500	2500	2500	—	—

Note: Some outside diameters, walls and grades are listed for information only and are not necessarily regular production items. Non-production sizes may be negotiated upon inquiry.

Some outside diameters, walls and grades are non-API.

*See notes at end of table on page 33.

Fig. 3N-3 (Cont'd) (Courtesy U.S. Steel)

Line Pipe Data

Seamless, Electric Resistance Weld, Double Submerged Arc Weld

Internal Pressure at Minimum Yield Strength*—psi								Ultimate Bursting Pressure*—psi								Actual OD Inches
Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60		Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60		
2710	3160	3790	4150	4690	—	—		4330	5410	5410	5680	5950	—	—		
3130	3650	4380	4790	5420	—	—		5000	6250	6250	6570	6880	—	—		
3850	4500	5390	5910	6680	—	—		6170	7710	7710	8090	8480	—	—		
4580	5350	6420	7030	7940	—	—		7330	9170	9170	9630	10080	—	—		18
5210	6070	7290	7980	9020	—	—		8330	10410	10410	10930	11450	—	—		
660	770	920	1010	1140	1230	1310		1050	1310	1310	1380	1580	1640	1710		
750	880	1050	1150	1300	1400	1500		1200	1500	1500	1580	1800	1880	1950		
840	980	1180	1290	1460	1570	1690		1350	1690	1690	1770	2020	2110	2190		
940	1090	1310	1440	1620	1750	1870		1500	1870	1870	1970	2250	2340	2430		
1030	1200	1440	1580	1790	1930	2060		1650	2060	2060	2170	2480	2580	2680		
1130	1310	1580	1730	1950	2100	2250		1800	2250	2250	2360	2700	2810	2930		
1220	1420	1710	1870	2110	—	—		1950	2440	2440	2560	2680	—	—		
1310	1530	1840	2010	2280	—	—		2100	2630	2630	2760	2890	—	—		
1410	1640	1970	2160	2440	—	—		2250	2810	2810	2950	3100	—	—		20
1500	1750	2100	2300	2600	—	—		2400	3000	3000	3150	3300	—	—		
1690	1970	2360	2590	2920	—	—		2700	3370	3370	3540	3710	—	—		
1780	2080	2490	2730	3090	—	—		2850	3560	3560	3740	3920	—	—		
1880	2190	2630	2880	3250	—	—		3000	3750	3750	3940	4130	—	—		
2060	2410	2890	3160	3580	—	—		3300	4130	4130	4330	4540	—	—		
2250	2630	3150	3450	3900	—	—		3600	4500	4500	4730	4950	—	—		
2440	2840	3410	3740	4220	—	—		3900	4870	4870	5120	5360	—	—		
3090	3610	4330	4740	5360	—	—		4950	6190	6190	6500	6800	—	—		
3840	4480	5380	5890	6660	—	—		6150	7690	7690	8070	8450	—	—		
680	800	950	1050	1180	1270	1360		1090	1360	1360	1430	1640	1700	1770		
770	890	1070	1180	1330	1430	1530		1230	1530	1530	1610	1840	1920	1990		
850	990	1190	1300	1470	1590	1700		1360	1700	1700	1790	2040	2130	2210		
940	1090	1310	1440	1630	1750	1880		1500	1880	1880	1970	2250	2350	2440		
1020	1190	1430	1570	1770	1910	2050		1640	2050	2050	2150	2450	2560	2660		
1110	1290	1550	1700	1920	—	—		1770	2210	2210	2330	2440	—	—		
1190	1390	1670	1830	2070	—	—		1910	2390	2390	2510	2630	—	—		
1280	1490	1790	1960	2220	—	—		2050	2560	2560	2690	2810	—	—		
1360	1590	1910	2090	2360	—	—		2180	2730	2730	2860	3000	—	—		22
1530	1790	2150	2350	2660	—	—		2450	3070	3070	3220	3370	—	—		
1700	1990	2390	2610	2950	—	—		2730	3410	3410	3580	3750	—	—		
1880	2190	2630	2880	3250	—	—		3000	3750	3750	3940	4130	—	—		
2050	2390	2860	3140	3550	—	—		3270	4090	4090	4300	4500	—	—		
2210	2580	3100	3400	3840	—	—		3540	4430	4430	4650	4870	—	—		
2390	2780	3340	3660	4140	—	—		3820	4770	4770	5010	5250	—	—		
3070	3580	4300	4700	5320	—	—		4910	6140	6140	6440	6750	—	—		
3410	3980	4770	5230	5910	—	—		5450	6820	6820	7160	7500	—	—		

Fig. 3N-3 (Cont'd) (Courtesy U.S. Steel)

Line Pipe Data

Seamless, Double Submerged Arc Weld

Actual OD Inches	Wall Thickness Inches	Weight Lb/Ft Plain End	Weight Tons/Mile	Class	Mill Test Pressure*—psi							
					Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60	Grade X-65
	0.250	63.41	167.41		470	550	790	860	980	1050	1130	1220
	0.281	71.18	187.92		530	610	890	970	1100	1180	1260	1370
	0.312	78.93	208.38		580	680	980	1080	1220	1310	1400	1520
	0.344	86.91	229.44		640	750	1080	1190	1340	1440	1550	1680
	0.375	94.62	249.79	Std.	700	820	1180	1290	1460	1580	1690	1830
	0.406	102.31	270.09		760	890	1280	1400	1580	1710	1830	—
	0.438	110.22	290.98		820	960	1380	1510	1710	1840	1970	—
	0.469	117.86	311.16		880	1030	1480	1620	1830	1970	2110	—
24.000	0.500	125.49	331.29	XS	940	1090	1580	1730	1950	2100	2250	—
	0.562	140.68	371.39		1050	1230	1770	1940	2190	—	—	—
	0.625	156.03	411.91		1170	1370	1970	2160	2300	—	—	—
	0.688	171.29	452.21		1290	1500	2170	2300	2300	—	—	—
	0.750	186.23	491.65		1410	1640	2300	2300	2300	—	—	—
	0.812	201.09	530.88		1520	1780	2300	2300	2300	—	—	—
	0.875	216.10	570.51		1640	1910	2300	2300	2300	—	—	—
	0.938	231.03	609.92		1760	2050	2300	2300	2300	—	—	—
	0.969	238.35	629.23		1820	2120	2300	2300	2300	—	—	—
	0.250	68.75	181.51		430	500	730	800	900	970	1040	1130
	0.281	77.18	203.77		490	570	820	890	1010	1090	1170	1260
	0.312	85.60	225.97		540	630	910	990	1120	1210	1300	1400
	0.344	94.26	248.84		600	690	1000	1100	1240	1330	1430	1550
	0.375	102.63	270.94	Std.	650	760	1090	1190	1350	1450	1560	1690
	0.406	110.98	292.98		700	820	1180	1290	1460	1570	1690	—
	0.438	119.57	315.68		760	880	1270	1390	1580	1700	1820	—
26.000	0.469	127.88	337.61		810	950	1360	1490	1690	1820	1950	—
	0.500	136.17	359.49	XS	870	1010	1450	1590	1800	1940	2000	—
	0.562	152.68	403.08		970	1130	1630	1790	2000	2000	2000	—
	0.625	169.38	447.16		1080	1260	1820	1990	2000	2000	2000	—
	0.656	177.56	468.76		1140	1320	1910	2000	2000	2000	2000	—
	0.688	185.99	491.01		1190	1390	2000	2000	2000	2000	2000	—
	0.750	202.25	533.95		1300	1510	2000	2000	2000	2000	2000	—
	0.250	79.43	209.70		370	440	630	690	780	840	900	980
	0.281	89.19	235.46		420	490	710	780	880	940	1010	1100
	0.312	98.93	261.16		470	550	790	860	970	1050	1120	1220
	0.344	108.95	287.64		520	600	870	950	1070	1160	1240	1340
	0.375	118.65	313.23		560	660	950	1040	1170	1260	1350	1460
	0.406	128.32	338.77		610	710	1020	1120	1270	1360	1460	—
	0.438	138.29	365.08		660	770	1100	1210	1370	1470	1580	—
30.000	0.469	147.92	390.50		700	820	1180	1290	1460	1580	1690	—
	0.500	157.53	415.88		750	880	1260	1380	1560	1680	1800	—
	0.562	176.69	466.47		840	980	1420	1550	1750	1890	2020	—
	0.625	196.08	517.65		940	1090	1580	1730	1950	2100	2250	—
	0.656	205.59	542.75		980	1150	1650	1810	2050	2200	2360	—
	0.688	215.38	568.60		1030	1200	1730	1900	2150	2310	2480	—
	0.750	234.29	618.53		1130	1310	1890	2070	2340	2520	2700	—

Note: Some outside diameters, walls and grades are listed for information only and are not necessarily regular production items. Non-production sizes may be negotiated upon inquiry.

Some outside diameters, walls and grades are non-API.

*See notes at end of table on page 33.

Fig. 3N-3 (Cont'd) (Courtesy U.S. Steel)

Line Pipe Data

Seamless, Double Submerged Arc Weld

Internal Pressure at Minimum Yield Strength*—psi								Ultimate Bursting Pressure*—psi								Actual OD Inches
Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60	Grade X-65	Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60	Grade X-65	
630	730	880	960	1080	1170	1250	1350	1000	1250	1250	1310	1500	1560	1630	1670	
700	820	980	1080	1220	1310	1400	1520	1120	1410	1410	1480	1690	1760	1830	1870	
780	910	1090	1200	1350	1460	1560	1690	1250	1560	1560	1640	1870	1950	2030	2080	
860	1000	1200	1320	1490	1610	1720	1860	1380	1720	1720	1810	2060	2150	2240	2290	
940	1090	1310	1440	1630	1750	1880	2030	1500	1880	1880	1970	2250	2340	2440	2500	
1010	1180	1420	1560	1760	1890	2030	—	1620	2030	2030	2130	2230	2400	2540	—	
1090	1280	1530	1680	1900	2040	2190	—	1750	2190	2190	2300	2410	2590	2740	—	
1170	1370	1640	1800	2030	2190	2350	—	1880	2350	2350	2460	2580	2770	2930	—	
1250	1460	1750	1920	2170	2330	2500	—	2000	2500	2500	2630	2750	2960	3130	—	24
1410	1640	1970	2150	2440	—	—	—	2250	2810	2810	2950	3090	—	—	—	
1560	1820	2190	2400	2710	—	—	—	2500	3130	3130	3280	3440	—	—	—	
1720	2010	2410	2640	2980	—	—	—	2750	3440	3440	3610	3780	—	—	—	
1880	2190	2630	2880	3250	—	—	—	3000	3750	3750	3940	4130	—	—	—	
2030	2370	2840	3110	3520	—	—	—	3250	4060	4060	4260	4470	—	—	—	
2190	2550	3060	3350	3790	—	—	—	3500	4380	4380	4590	4810	—	—	—	
2340	2740	3280	3600	4060	—	—	—	3750	4690	4690	4920	5160	—	—	—	
2420	2830	3390	3710	4200	—	—	—	3880	4850	4850	5090	5330	—	—	—	
580	670	810	880	1000	1080	1150	1250	920	1150	1150	1210	1380	1440	1500	1540	
650	760	910	990	1120	1210	1300	1400	1040	1300	1300	1360	1560	1620	1690	1730	
720	840	1010	1100	1250	1340	1440	1560	1150	1440	1440	1510	1730	1800	1870	1920	
790	930	1110	1220	1380	1480	1590	1720	1270	1590	1590	1670	1910	1980	2060	2120	
870	1010	1210	1330	1500	1620	1730	1880	1380	1730	1730	1820	2080	2160	2250	2310	
940	1090	1310	1440	1620	1750	1870	—	1500	1870	1870	1970	2060	2220	2340	—	
1010	1180	1420	1550	1750	1890	2020	—	1620	2020	2020	2120	2220	2390	2530	—	
1080	1260	1520	1660	1880	2020	2160	—	1730	2160	2160	2270	2380	2560	2710	—	26
1150	1350	1620	1770	2000	2150	2310	—	1850	2310	2310	2420	2540	2730	2880	—	
1300	1510	1820	1990	2250	2420	2590	—	2080	2590	2590	2720	2850	3070	3240	—	
1440	1680	2020	2210	2500	2690	2880	—	2310	2880	2880	3030	3170	3410	3610	—	
1510	1770	2120	2320	2620	2830	3030	—	2420	3030	3030	3180	3330	3580	3780	—	
1590	1850	2220	2430	2750	2960	3180	—	2540	3180	3180	3330	3490	3760	3970	—	
1730	2020	2420	2650	3000	3230	3460	—	2770	3460	3460	3630	3810	4100	4330	—	
500	580	700	770	870	930	1000	1080	800	1000	1000	1050	1200	1250	1300	1330	
560	660	790	860	970	1050	1120	1220	900	1120	1120	1180	1350	1400	1460	1500	
620	730	870	960	1080	1160	1250	1350	1000	1250	1250	1310	1500	1560	1620	1660	
690	800	960	1050	1190	1280	1380	1490	1100	1380	1380	1440	1650	1720	1790	1830	
750	880	1050	1150	1300	1400	1500	1630	1200	1500	1500	1580	1800	1880	1950	2000	
810	950	1140	1250	1410	1520	1620	—	1300	1620	1620	1710	1790	1920	2030	—	
880	1020	1230	1340	1520	1640	1750	—	1400	1750	1750	1840	1930	2070	2190	—	
940	1090	1310	1440	1630	1750	1880	—	1500	1880	1880	1970	2060	2220	2350	—	30
1000	1170	1400	1530	1730	1870	2000	—	1600	2000	2000	2100	2200	2370	2500	—	
1120	1310	1570	1720	1950	2100	2250	—	1800	2250	2250	2360	2470	2660	2810	—	
1250	1460	1750	1920	2170	2330	2500	—	2000	2500	2500	2630	2750	2960	3130	—	
1310	1530	1840	2010	2270	2450	2620	—	2100	2620	2620	2760	2890	3110	3280	—	
1380	1610	1930	2110	2390	2570	2750	—	2200	2750	2750	2890	3030	3260	3440	—	
1500	1750	2100	2300	2600	2800	3000	—	2400	3000	3000	3150	3300	3550	3750	—	

Fig. 3N-3 (Cont'd) (Courtesy U.S. Steel)

Line Pipe Data
Double Submerged Arc Weld

Actual OD Inches	Wall Thickness Inches	Weight Lb/Ft Plain End	Weight Tons/Mile	Class	Mill Test Pressure*—psi							
					Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60	Grade X-65
32.000	0.250	84.77	223.80		350	410	590	650	730	790	840	910
	0.281	95.19	251.30		400	460	660	730	820	890	950	1030
	0.312	105.59	278.76		440	510	740	810	910	980	1050	1140
	0.344	116.30	307.04		480	560	810	890	1010	1080	1160	1260
	0.375	126.66	334.38		530	620	890	970	1100	1180	1270	1370
	0.406	136.99	361.66		570	670	960	1050	1190	1280	1370	—
	0.438	147.64	389.77		620	720	1030	1130	1280	1380	1480	—
	0.469	157.94	416.95		660	770	1110	1210	1370	1480	1580	—
	0.500	168.21	444.07		700	820	1180	1290	1460	1580	1690	—
	0.562	188.70	498.16		790	920	1330	1450	1640	1770	1900	—
	0.625	209.43	552.89		880	1030	1480	1620	1830	1970	2110	—
	0.656	219.60	579.74		920	1080	1550	1700	1920	2070	2210	—
	0.688	230.08	607.40		970	1130	1630	1780	2010	2170	2320	—
	0.750	250.31	660.82		1050	1230	1770	1940	2190	2360	2530	—
34.000	0.250	90.11	237.90		330	390	560	610	690	740	790	860
	0.281	101.19	267.15		370	430	620	680	770	830	890	970
	0.312	112.25	296.35		410	480	690	760	860	920	990	1070
	0.344	123.65	326.43		460	530	760	840	950	1020	1090	1180
	0.375	134.67	355.52		500	580	830	910	1030	1110	1190	1290
	0.406	145.67	384.56		540	630	900	990	1120	1200	1290	—
	0.438	157.00	414.47		580	680	970	1070	1210	1300	1390	—
	0.469	167.95	443.40		620	720	1040	1140	1290	1390	1490	—
	0.500	178.89	472.27		660	770	1110	1220	1380	1480	1590	—
	0.562	200.70	529.85		740	870	1250	1370	1550	1670	1790	—
	0.625	222.78	588.13		830	970	1390	1520	1720	1850	1990	—
	0.656	233.61	616.73		870	1010	1460	1600	1810	1940	2080	—
	0.688	244.77	646.20		910	1060	1530	1680	1890	2040	2190	—
	0.750	266.33	703.12		990	1160	1670	1830	2060	2220	2380	—
36.000	0.250	95.45	251.99		310	360	530	580	650	700	750	810
	0.281	107.20	283.00		350	410	590	650	730	790	840	910
	0.312	118.92	313.94		390	450	660	720	810	870	940	1010
	0.344	131.00	345.83		430	500	720	790	890	960	1030	1120
	0.375	142.68	376.67		470	550	790	860	980	1050	1130	1220
	0.406	154.34	407.45		510	590	850	930	1060	1140	1220	—
	0.438	166.35	439.17		550	640	920	1010	1140	1230	1310	—
	0.469	177.97	469.85		590	680	980	1080	1220	1310	1410	—
	0.500	189.57	500.46		620	730	1050	1150	1300	1400	1500	—
	0.562	212.70	561.54		700	820	1180	1290	1460	1570	1690	—
	0.625	236.13	623.38		780	910	1310	1440	1630	1750	1880	—
	0.656	247.62	653.72		820	960	1380	1510	1710	1840	1970	—
	0.688	259.47	684.99		860	1000	1440	1580	1790	1930	2060	—
	0.750	282.35	745.41		940	1090	1580	1730	1950	2100	2250	—

Note: Some outside diameters, walls and grades are listed for information only and are not necessarily regular production items. Non-production sizes may be negotiated upon inquiry.

Some outside diameters, walls and grades are non-API.

*See notes at end of table on page 33.

Fig. 3N-3 (Cont'd) (Courtesy U.S. Steel)

Line Pipe Data
Double Submerged Arc Weld

Internal Pressure at Minimum Yield Strength*—psi								Ultimate Bursting Pressure*—psi								Actual OD Inches
Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60	Grade X-65	Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60	Grade X-65	
470	550	660	720	810	880	940	1020	750	940	940	980	1130	1170	1220	1250	
530	610	740	810	910	980	1050	1140	840	1050	1050	1110	1260	1320	1370	1410	
580	680	820	900	1010	1090	1170	1270	940	1170	1170	1230	1400	1460	1520	1560	
640	750	900	990	1120	1200	1290	1400	1030	1290	1290	1350	1550	1610	1680	1720	
700	820	980	1080	1220	1310	1410	1520	1130	1410	1410	1480	1690	1760	1830	1880	
760	890	1070	1170	1320	1420	1520	—	1220	1520	1520	1600	1670	1800	1900	—	
820	960	1150	1260	1420	1530	1640	—	1310	1640	1640	1720	1810	1940	2050	—	
880	1030	1230	1350	1520	1640	1760	—	1410	1760	1760	1850	1930	2080	2200	—	32
940	1090	1310	1440	1630	1750	1880	—	1500	1880	1880	1970	2060	2220	2340	—	
1050	1230	1480	1620	1830	1970	2110	—	1690	2110	2110	2210	2320	2490	2630	—	
1170	1370	1640	1800	2030	2190	2340	—	1880	2340	2340	2460	2580	2770	2930	—	
1230	1440	1720	1890	2130	2300	2460	—	1970	2460	2460	2580	2710	2910	3080	—	
1290	1500	1810	1980	2240	2410	2580	—	2060	2580	2580	2710	2840	3050	3230	—	
1410	1640	1970	2160	2440	2630	2810	—	2250	2810	2810	2950	3090	3330	3520	—	
440	510	620	680	760	820	880	960	710	880	880	930	1060	1100	1150	1180	
500	580	690	760	860	930	990	1070	790	990	990	1040	1190	1240	1290	1320	
550	640	770	840	950	1030	1100	1190	880	1100	1100	1160	1320	1380	1430	1470	
610	710	850	930	1050	1130	1210	1320	970	1210	1210	1270	1460	1520	1580	1620	
660	770	930	1010	1150	1240	1320	1430	1060	1320	1320	1390	1590	1650	1720	1760	
720	840	1000	1100	1240	1340	1430	—	1150	1430	1430	1500	1580	1700	1790	—	
770	900	1080	1190	1340	1440	1550	—	1240	1550	1550	1620	1700	1830	1930	—	
830	970	1160	1270	1430	1540	1660	—	1320	1660	1660	1740	1820	1960	2070	—	34
880	1030	1240	1350	1530	1650	1760	—	1410	1760	1760	1850	1940	2090	2210	—	
990	1160	1390	1520	1720	1850	1980	—	1590	1980	1980	2080	2180	2350	2480	—	
1100	1290	1540	1690	1910	2060	2210	—	1760	2210	2210	2320	2430	2610	2760	—	
1160	1350	1620	1780	2010	2160	2320	—	1850	2320	2320	2430	2550	2740	2890	—	
1210	1420	1700	1860	2100	2270	2430	—	1940	2430	2430	2550	2670	2870	3040	—	
1320	1540	1850	2030	2290	2470	2650	—	2120	2650	2650	2780	2910	3130	3310	—	
420	490	580	640	720	780	830	900	670	830	830	880	1000	1040	1080	1110	
470	550	660	720	810	870	940	1010	750	940	940	980	1120	1170	1220	1250	
520	610	730	800	900	970	1040	1130	830	1040	1040	1090	1250	1300	1350	1390	
570	670	800	880	990	1070	1150	1240	920	1150	1150	1200	1380	1430	1490	1530	
630	730	880	960	1080	1170	1250	1350	1000	1250	1250	1310	1500	1560	1630	1670	
680	790	950	1040	1170	1260	1350	—	1080	1350	1350	1420	1490	1600	1690	—	
730	850	1020	1120	1270	1360	1460	—	1170	1460	1460	1530	1610	1730	1830	—	
780	910	1090	1200	1350	1460	1560	—	1250	1560	1560	1640	1720	1850	1950	—	36
830	970	1170	1280	1440	1560	1670	—	1330	1670	1670	1750	1830	1970	2080	—	
940	1090	1310	1440	1620	1750	1870	—	1500	1870	1870	1970	2060	2220	2340	—	
1040	1220	1460	1600	1810	1940	2080	—	1670	2080	2080	2190	2290	2470	2600	—	
1090	1280	1530	1680	1900	2040	2190	—	1750	2190	2190	2300	2410	2590	2730	—	
1150	1340	1610	1760	1990	2140	2290	—	1830	2290	2290	2410	2520	2710	2870	—	
1250	1460	1750	1920	2170	2330	2500	—	2000	2500	2500	2630	2750	2960	3130	—	

Fig. 3N-3 (Cont'd) (Courtesy U.S. Steel)

Line Pipe Data

Double Submerged Arc Weld

Actual OD Inches	Wall Thickness Inches	Weight Lb / Ft Plain End	Weight Tons / Mile	Class	Mill Test Pressure*—psi							
					Grade A	Grade B	Grade X-42	Grade X-46	Grade X-52	Grade X-56	Grade X-60	Grade X-65
38.000	0.312	125.58	331.54		370	430	620	680	770	830	890	960
	0.344	138.35	365.23		410	480	680	750	850	910	980	1060
	0.375	150.69	397.82		440	520	750	820	920	990	1070	1150
	0.406	163.01	430.35		480	560	810	880	1000	1080	1150	—
	0.438	175.71	463.87		520	610	870	950	1080	1160	1240	—
	0.469	187.99	496.29		560	650	930	1020	1160	1240	1330	—
	0.500	200.25	528.66		590	690	990	1090	1230	1330	1420	—
	0.562	224.71	593.23		670	780	1120	1220	1380	1490	1600	—
	0.625	249.48	658.62		740	860	1240	1360	1540	1660	1780	—
	0.688	274.16	723.79		810	950	1370	1500	1690	1830	1960	—
	0.750	298.37	787.70		890	1040	1490	1630	1850	1990	2130	—
40.000	0.312	132.25	349.13		350	410	590	650	730	790	840	910
	0.344	145.69	384.63		390	450	650	710	800	870	930	1010
	0.375	158.70	418.96		420	490	710	780	880	940	1010	1100
	0.406	171.68	453.24		460	530	770	840	950	1020	1100	—
	0.438	185.06	488.57		490	570	830	910	1020	1100	1180	—
	0.469	198.01	522.74		530	620	890	970	1100	1180	1270	—
	0.500	210.93	556.86		560	660	950	1040	1170	1260	1350	—
	0.562	236.71	624.92		630	740	1060	1160	1320	1420	1520	—
	0.625	262.83	693.87		700	820	1180	1290	1460	1580	1690	—
	0.688	288.86	762.59		770	900	1300	1420	1610	1730	1860	—
	0.750	314.39	830.00		840	980	1420	1550	1760	1890	2030	—
42.000	0.312	138.91	366.73		330	390	560	620	700	750	800	870
	0.344	153.04	404.03		370	430	620	680	770	830	880	960
	0.375	166.71	440.11		400	470	680	740	840	900	960	1040
	0.406	180.35	476.14		430	510	730	800	900	970	1040	—
	0.438	194.42	513.27		470	550	790	860	980	1050	1130	—
	0.469	208.03	549.19		500	590	840	920	1050	1130	1210	—
	0.500	221.61	585.05		540	630	900	990	1110	1200	1290	—
	0.562	248.72	656.61		600	700	1010	1110	1250	1350	1450	—
	0.625	276.18	729.11		670	780	1130	1230	1390	1500	1610	—
	0.688	303.55	801.38		740	860	1240	1360	1530	1650	1770	—
	0.750	330.41	872.29		800	940	1350	1480	1670	1800	1930	—
44.000 (Subject to inquiry)	0.344	160.39	423.43		350	410	590	650	730	790	840	910
	0.375	174.72	461.26		380	450	640	710	800	860	920	1000
	0.406	189.03	499.03		420	480	700	760	860	930	1000	—
	0.438	203.78	537.97		450	520	750	820	930	1000	1080	—
	0.469	218.04	575.63		480	560	810	880	1000	1070	1150	—
	0.500	232.29	613.25		510	600	860	940	1060	1150	1230	—
	0.562	260.72	688.31		570	670	970	1060	1200	1290	1380	—
	0.625	289.53	764.35		640	750	1070	1180	1330	1430	1530	—
	0.688	318.25	840.18		700	820	1180	1290	1460	1580	1690	—
	0.750	346.43	914.58		770	890	1290	1410	1600	1720	1840	—

Note: Some outside diameters, walls and grades are listed for information only and are not necessarily regular production items. Non-production sizes may be negotiated upon inquiry.

Some outside diameters, walls and grades are non-API.

Fig. 3N-3 (Cont'd) (Courtesy U.S. Steel)

GAS LIFT RULES OF THUMB (for approximation only)

1. Pressure Needed

100 psi per 1000 ft of depth up to 1800 psi with a minimum of 300 psi

Example:

(a) Pressure needed for a 7000 ft well:

$$100 \times \frac{7000}{1000} = 700 \text{ psi needed}$$

(b) 10,000 ft well:

$$100 \times \frac{10{,}000}{1000} = 1000 \text{ psi}$$

Equipment life may be shortened above 1800 psi.

2. Gas Volumes Needed

(For no multiphase flow correlations available)

Continuous flow. 150–250 scf/B per 1000 ft of lift.

Example:

Given a well lifting from 4000 ft:

$$\text{Gas needed} = 200 \times \frac{4000}{1000} = 800 \text{ scf/B}$$

Intermittent flow. 250–400 scf/B per 1000 ft of lift. (See Appendix 3G)

Example:

Given an 8000 ft well:

$$\text{Gas needed} = 300 \times \frac{8000}{1000} = 2400 \text{ scf/B}$$

3. Allowable Continuous Flow Depth

$$\text{Depth} = \frac{p_c - p_t}{0.15}$$

where: p_c = casing pressure (psig)
p_t = tubing pressure (psig)

Example:

Operating casing pressure = 850 psig; flowing tubing pressure = 100 psig:

$$\text{Depth} = \frac{p_c - p_t}{0.15} = \frac{850 - 100}{0.15} = \frac{750}{0.15} = 5000 \text{ ft.}$$

4. Minimum Time Required for Intermittent Cycles

$$\text{Minutes} = 1.5 \times \frac{\text{depth}}{1000}$$

Example:

Depth of operating valve = 8000 ft:

$$\text{Minimum cycle time} = (1.5)\left(\frac{8000}{1000}\right) = 12 \text{ minutes}$$

5. Slug Velocity

For intermittent lift = 1000 ft/min. For a valve located at 8000 ft how long after gas is injected should the slug reach the surface?

$$\frac{8000}{1000} = 8 \text{ minutes}$$

6. Horsepower Required to Compress Gas

$$hp = 22\, q \frac{p_2}{p_1}$$

q = Volume of gas to be handled per day, mmcf
p_1 = Suction pressure
p_2 = Discharge pressure

Example: 5 MMcf/day $p_1 = 200$, $p_2 = 1000$

$$hp = (22)(5)\left(\frac{1000}{200}\right) = 550$$

7. Fuel Requirements for Compressor

(HP)(240) = scf/d of fuel required

Example:

For 1000 hp

(1000)(240) = 240,000 scf/d

8. PIPE CAPACITIES

For *Tubing*

$ft^3/100$ ft. = 0.545 (I.D.)^2

BBLS/100 ft. = 0.097 (I.D.)^2

For quick answer may use (nominal size)^2 = Bbls/1000'

Example 2" Tubing = $(2)^2$ = 4 BBLS/1000'

For *Annulus*

$ft^3/100$ ft. = 0.545 [(O.D.)^2 – (I.D.)^2]

BBLS/100 ft. = 0.097 [(O.D.)^2 – (I.D.)^2]

For *Annulus with two pipe strings*

$ft^3/100$ ft = 0.545 [(O.D.)^2 – $(I.D.)_1^2$ – $(I.D.)_2^2$]

B/100 ft = 0.097 [(O.D.)^2 – $(I.D.)_1^2$ – $(I.D.)_2^2$]

9. Temperature Drop Across a Regulator is 1°F per atmosphere (15 psi) of pressure drop.

Example: $p_u = 750$, $p_d = 600$ Find temperature drop

$750 - 600 = 150$

$\frac{150}{15} = 10$ atm.

(10) (1) = 10°F temp drop

10. Setting Weights for Hookwall Packers

Depth	Weight
2000 – 4000	Full Tubing Load
4000 – 6000	¾ Tubing Load
6000 – 8000	½ Tubing Load
8000 – 10000	¼ Tubing Load

11. Gas Volumes Used per Cycle on Intermittent Lift

$$G = \left(\frac{\text{drop in pressure on casing (psi)}}{15}\right) \times \left(\text{storage volume in casing [cu ft]}\right)$$

Given 5½ in. casing and 2 in. tubing with valve depth at 8000 ft. Casing builds to 800 psi and valves close at 700 psi on casing. Find gas used. The annular capacity for 5½ in. casing by 2 in. tubing is 0.10 cu ft/ft = 800 cu ft total.

$$G = \left(\frac{800 - 700}{15}\right)(800) = 5350 \text{ scf of gas.}$$

12. The Loss in Intermittent Lift is 7 percent per 1000 ft (For chambers use 5%)

Example:

Slug started from bottom = 6 bbls. Depth = 6000 ft.

$$\text{Loss} = 7\% \text{ of } \left(\frac{6000}{1000}\right) = 42\%$$

(0.42)(6) = 2.52 bbls

Recovery = (6 – 2.52) = 3.48 bbls

13. Approximate Maximum Rates Possible from Tubing Sizes under Normal Lift Conditions (Continuous Flow)

Tubing diam (in.)	Rate (B/D)
4	8000
3½	5000
3	4000
2½	3000
2	2500
1½	1000
1¼	600
1	350

14. Approximate Change in Production Rate from Intermittent Flow to Continuous Flow under Normal Conditions

Straight Tubing Installation	
Tubing size (in.)	Δ Rate (B/D)
1	+ 25–50
1¼	+ 50–75
1½	+ 75–125
2	+200
2½	+250
3	+300
chamber	+400–600

15. Pressure Exerted by Gas Column =

$$(2.5)\left(\frac{\text{casing pressure}}{100}\right)\left(\frac{\text{depth}}{1000}\right)$$

Example:

Surface operating pressure = 800 psig; depth = 9000 ft. Find the weight of the gas:

$$\text{Weight of gas} = (2.5)\left(\frac{800}{100}\right)\left(\frac{9000}{1000}\right) = 180 \text{ psig}$$

Pressure at depth of operating valve
= 800 + 180 = 980 psig

16. Dome Pressure Charged Unbalanced Gas Lift Valve

Valve set at 60°F will have a surface operating pressure very near its test rack opening pressure at 60°F. Gas column weight and tubing pressure effect is essentially offset by temp effect.

17. Normal Pressure Expected in New Well is 0.465 × depth

Example:

A new well is drilled to 10,000 ft. What BHP (bottom hole pressure) is expected?

$$(0.465)(10{,}000) = 4650 \text{ psig}$$

18. Normal Temperature Expected in a New Well along Texas and Louisiana Gulf Coast

Surface temp = 74°F; 1.6°F/100 ft gradient in a 10,000 ft well. What temp is expected at bottom?

$$74°\text{F} + 1.6°\text{F}\left(\frac{10{,}000}{100}\right) = 74 + 160 = 234°\text{F}$$

19. Productivity Index

$$J \equiv 0.6h\left[\frac{k_o}{\mu_o \beta_o}\right] \text{ for no water production}$$

where: J = productivity index
h = net pay thickness (ft)
k_o = permeability to oil (darcies)
β_o = oil formation volume factor
μ_o = oil viscosity (cp)

Example:

Given: h = 20 ft
k_o = 500 md = 0.5 darcies
β_o = 1.4
μ_o = 1 cp

$$J \equiv .6\,h\left[\frac{k_o}{\mu_o \beta_o}\right] = (.6)(20)\left[\frac{.5}{(1.4)(1)}\right] = 4.29$$

For water plus oil production:

$$J \equiv h\left[\frac{k_o}{\beta_o \mu_o} + \frac{k_w}{\beta_w \mu_w}\right](.6)$$

where: k_w = permeability to water
β_w = water formation volume factor
μ_w = water viscosity